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Revision Memo

12/99	The highlights of these revisions are:	
1.	Page 4	Updated Reference List
2.	Page 4	Updated Background information on Stacks
3.	Page 5	Updated Extended Surfaces information
4.	Page 6	Updated table to include stud spacing
5.	Page 13	Updated equations for Shield Tube Heat Density
6.	Page 16	Added table for equivalent lengths between studs and thick fins
7.	Page 16	Replaced equations for inside heat transfer coefficients with the Dittus Boelter equation
8.	Pages 20-23	Corrected tube metal temperature calculations
9.	Page 23	Corrected equation for equivalent tube conductivity
10.	Page 24	Updated information in Stack Linings section
11.	Page 25	Updated density equations to reflect effect of molecular weight
12.	Page 26	Added section on Flue Gas Pressure Variation and added graph for draft in a furnace
13.	Page 41	Updated frictional pressure drop formula to reflect changes in DP XIV , Fluid Flow
14.	Pages 32-59	Updated examples to reflect changes in equations
15.	Page 57-60	Updated units in Nomenclature
16.	Page 61	Added Computational Fluid Dynamics to Computer Programs listing
17.		Relocated TMT calculation from Appendix to Convection Section Design
18.		Deleted Figures 3A and 3B - Heat transfer coefficients for fluids in tubes
19.		Deleted Figure 4 - Inside film coefficients for fluids containing hydrogen (Colburn correlation)
20.	Page 77	Corrected Metric axes in Figure 14



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SCOPE

This section contains calculation procedures to be used for designing convection sections and stacks for all fired heaters. Where specialized procedures are required for the convection section design, they are noted.

The procedures outlined below are based on conventional single service convection section arrangements, with countercurrent flow of flue gas and process fluid. Flue gas flow is upward, and the process fluid flows downward from the top of the convection section to the bottom (see **SHIELD SECTION**). Although the basic equations are unchanged, other arrangements, as well as auxiliary services such as steam superheaters, will require modifications to these procedures.

REFERENCES

DESIGN PRACTICES

Other sections of Section VIII, Fired Heaters

[Section XIV](#) Fluid Flow

INTERNATIONAL PRACTICES

[IP 7-1-1](#), Fired Heaters

OTHER LITERATURE

ASHRAE Guide and Data Book, 1969.

General Electric Company, *Heat Transfer and Fluid Flow* (1970).

Grimison, E. D., *Cross Flow of Gases Over Tube Banks*, ASME Transactions PRO-59-8 (1937).

Tenner, A. R. and Golan, L. P., *Graphic Procedure for Calculating Heat Transfer Coefficient and Flow Regimes for Single and Two Phase Flow in Tubes*, ER&E Report No. EE.6ER.71 (March 1971).

BACKGROUND

CONVECTION SECTION

Flue gases leave the fired heater radiant section and pass through the convection section before entering the stack. At flue gas temperatures below 1500 to 1800°F (815 to 980°C), radiant heat transfer becomes relatively uneconomical, and if further heat recovery from the flue gas is desired, this is more economically accomplished by convective heat transfer. Typically, 35 to 40% of the total fired heater duty is obtained in modern extended surface convection sections. The convection section must be designed to transfer heat economically to the process fluid, and at the same time not to impede unduly the flue gas leaving the fired heater. Since the flue gas heat transfer coefficient on the outside tube surface is relatively poor, extended surface is usually used to increase heat transfer.

STACK

The purpose of the stack is to create the necessary draft for flue gases to flow properly through the fired heater system and to dispose of the flue gases. This must be done without imposing a positive pressure on any part of the fired heater box, since the construction box is not pressure-tight and flue gases leaking out can cause severe damage to the fired heater casing and structure. Also, flue gas leakage could seriously injure nearby personnel. Adequate draft is also necessary for natural draft burners, to inspire the required combustion air. The hot stack gases must also be discharged at a safe height with respect to surrounding equipment (towers and other structures).

Since flue gases contain pollutants such as SO_x, NO_x, and particulates, the stack must be designed to discharge these gases in a manner that avoids causing a local pollution problem. Stack height and flue gas exit velocity are important design variables that allow the stack to discharge and disperse pollutants for the purpose of meeting regulatory emission requirements. Turndown conditions (as well as design conditions) must be considered, since ground-level concentrations of pollutants from the flue gases are affected by the flue gas exit velocity.

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DEFINITIONS

See [Section VIII-A](#).

CONVECTION SECTION SURFACE

SHIELD SECTION

The first two rows of the convection section are exposed to radiation from the radiant section and are known as "shield tubes." The radiant heat transfer to these two rows is included in the total fired heater radiant duty (see [Section VIII-B](#)). Since the shield tubes also pick up some heat by convection, the total shield duty is the sum of these radiant and convection heat duties. Extended surface must not be used on the shield tubes, since the resulting heat pickup and heat densities would be excessive, causing high tube metal temperatures and/or vapor blanketing.

IP 7-1-1 requires the shield tubes to be of the same material as the radiant tubes. An exception can be made where the shield service is different from the radiant service, such as a steam generator coil in a process fired heater.

BARE TUBE SECTION

Although extended surface should be used as low in the convection section as possible after the shield section, this is usually not practical for the tubes immediately following the shield tubes. Flue gas temperatures may still be high and could cause excessively high heat densities and tube metal temperatures. Therefore, a few rows of bare tubes are usually required before extended surfaces can be used.

IP 7-1-1 requires at least one bare row after the shield in all cases.

EXTENDED SURFACES

▶ **Selection of Extended Surface Type** - The type of extended surface to be used is based on the fuel to be fired in the fired heater.

1. **Stud type and thick-fin type** extended surfaces are the only types that may be used if heavy liquid fuels [containing greater than 0.01% ash and 25° API or lower, $\rho = 56.4 \text{ lb/ft}^3$ (904 kg/m³) or greater] are to be fired. **Figure 1A** shows a studded tube.
2. **Thin-fin type** extended surface should be used only when fuel gas or light liquid fuels [containing less than 0.01% ash and above 25° API, ρ less than 56.4 lb/ft³ (904 kg/m³)] are to be fired. **Figure 1B** shows a finned tube.

In fired heaters where a number of fuels are to be fired, the heaviest fuel governs the selection of extended surface type. Note that in fired heaters where heavy liquid fuels are to be fired at any time in the future, studded or thick-finned tubes should be used. Conversion of a fired heater originally designed for *only gas firing* to permit heavy liquid fuel firing would require major revamping of the fired heater. The criteria for selecting thick-fins versus studs is usually based on cost and availability.

Studded Tubes

1. **Application** - Cylindrical studs are resistance welded to the tube in circular or helical rows. Carbon steel studs, 1/2-in. (12.7 mm) diameter, and spaced on rows 5/8 in. (15.9 mm) apart (19.2 rows per ft of tube or 63.0 rows per m of tube) are standard and should be so specified. Other types of studded surface, including 1/4 and 3/8 in. (6.4 and 9.5 mm) diameters and elliptical shaped studs, are available. However, these should not be used because of their disadvantages, such as reduced mechanical strength and poorer cleanability.
2. **Temperature** - Carbon steel studs should be limited to about 1050°F (565°C) maximum tip temperature, based on stud oxidation considerations. CS studs can be applied to any tube material. Stress relieving is not required. It is generally not economical to use alloy studs.
- ▶ 3. **Stud Arrangement** - Studs can be applied on the entire tube diameter, or on just the bottom half of the tube (the side facing the flow of flue gas). Half-studded tubes (studded along the bottom half of the tube) were used in the past but should no longer be used. Fully studded tubes are recommended.
4. **Stud Length** - Studs are available in length increments of 1/16 in. (1.6 mm), starting at 3/4 in. (19.1 mm) long (5/8 in. or 15.9 mm studs can also be obtained, but they must be cut to length by hand and should therefore not normally be used.) The maximum possible stud length is based on manufacturing capabilities, and depends on tube size, since the outside diameter of the tube and studs is limited to 11 1/8 in. (282.6 mm). Maximum stud lengths should usually be limited to about 2 in. (50 mm) because stud efficiency drops sharply with increasing length.

Selection of stud lengths depends upon convection section geometry and flue gas mass velocity, as discussed below. Short studs should be used as soon as possible in the lower convection section, as limited by stud tip, tube and film temperatures; and stud length should be increased as flue gas temperature decreases through the convection section.

CONVECTION SECTION SURFACE (Cont)

5. **Studs per Row** - The number of studs per circumferential row depends upon tube diameter. The number of studs per row should be limited to the maximum values listed below:

OUTSIDE DIAMETER OF PIPE OR TUBE, IN.	2.5	3.0	3.5	4.0	4.5	5.0	5.563	6.0	6.625	8.625
mm	63.5	76.2	88.9	101.6	114.3	127.0	141.3	152.4	168.3	219.1
Studs per circumferential row	8	10	12	12	14	16	18	20	22	28
BASED ON ROWS SPACED AT 5/8 in. (15.9 mm)										
Studs per foot	153.6	192.0	230.4	230.4	268.8	307.2	345.6	384.0	422.4	537.6
Studs per meter	504.0	630.0	756.0	756.0	882.0	1008.0	1134.0	1260.0	1386.0	1764.0

Finned Tubes

- Application** - Thin-fins are continuously wound around the tube in a spiral, with up to about 5 turns per in. (2 turns per cm). Thick-fins are also continuously wound in a spiral, with up to about 3 turns per in. (1.2 turns per cm). The fins should be continuously welded to the tube by a fillet or resistance weld. These welds are generally required over intermittent spot-welding attachments, because of their better heat transfer capabilities and their stronger bond to the tube (see [IP 7-5-1](#)).
- Fin Type** - Continuous solid fins ([Figure 1B](#)) are made by stretching a continuous strip of metal around the tube. As covered by [IP 7-5-1](#), solid fins are preferred over serrated fins, which are formed by partially splitting the metal strip before winding it around the tube. Solid fins have superior mechanical rigidity. Calculation procedures given in this section are based on solid fins. It may not be possible to obtain solid fins for all combinations of tube diameter, fin height, and fin material due to mechanical constraints imposed by the manufacturing process.
- Availability of Thin-Fins** - A variety of fins are available, as follows:
 - Fin height from 1/4 to 1-1/2 in. (from 6.4 to 38.1 mm).
 - Fin thicknesses from 0.02 to 0.06 in. (from 0.5 to 1.5 mm).
 - Fin spacing of up to 10 fins per in. of tube length (fins as close as 2.5 mm).

Not all combinations of these variables are available. For economical design, fins should be based on the following:

 - Fin height of 1/2, 3/4 or 1 in. (12.7, 19.1 or 25.4 mm).
 - Fin thickness of 0.05 in. (1.3 mm).
 - Spacing of 4 to 5 fins per in. (5.1 to 6.4 mm).
- Availability of Thick-Fins**
 - Fin height from 1/2 to 1-1/4 in. (12.7 to 30.6 mm).
 - Fin thicknesses of 0.105 in. (2.7 mm).
 - Fin spacing of 1 to 4 fins per in. (6.4 to 25.4 mm).

To minimize fouling potential, it is recommended that a maximum height of 3/4 in. (19 mm) and a spacing of 3 fins per in. (8.5 mm) be used.
- Material** - Almost any fin material can be attached to any tube material. Since fins are much thinner than studs, fin materials should be selected for zero oxidation at the maximum tip temperature. The higher the tip temperature, the better grade fin is required. The following fin materials are commonly used:

FIN MATERIAL	MAXIMUM FIN TIP TEMPERATURE	
	°F	°C
CS	850	455
11-13 Cr	1250	675
18/8 Cr/Ni	1450	790
25/20 Cr/Ni	1800	980

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CONVECTION SECTION LAYOUT

MECHANICAL CONSIDERATIONS

The convection section tube spacing and layout are based largely on process considerations, such as the required or optimum flue gas mass velocities, heat transfer rate, etc. However, the following mechanical considerations also limit or influence the convection section layout.

Length L_c - The effective length of the convection section (and shield) tubes depends on the particular type of fired heater being designed. In general, the maximum convection section length permitted by the radiant section design should be used. Convection section return bends should always be located in header boxes. Therefore, the inside length of the convection section box is the tube effective length.

Width W_c - Each horizontal row in the convection section should have at least 4 tubes or the number or passes in the fired heater, whichever is greater. (In very small fired heaters, 2 or 3 tubes per row may be used.)

If the convection section is long enough to require one or more intermediate tubesheets, it should not be made wider than about 14 ft (4.3 m), because of difficulties encountered in the design and fabrication of intermediate tubesheets larger than this. Short convection sections without intermediate tubesheets may be designed wider than 14 ft (4.3 m).

Number of Passes - In vaporizing services, the number of tube passes in the convection section should be the same as in the radiant section, each pass being connected with its own crossover. In all-vapor services, where distribution can be made by a natural split, a different number of passes may be used in the radiant and convection sections. Flows may be combined at the convection section outlet and redistributed at the radiant section inlet. See [Section VIII-J](#).

Return Bends - Whenever possible, the tube arrangements and layout should permit the use of standard short-radius 180° U-bends (center-to-center spacing = 2 times nominal diameter = 2 IPS). Long-radius U-bends (3 IPS) are used in unusual situations.

For other cases, where non-standard spacings are required, the following tables list minimum tube spacing for convection section tubes, based on clearances required for fabrication. Since return bends purchased for non-standard spacing are all custom made, any spacing above the minimum is obtainable.

PIPE SIZE		STANDARD SPACING CENTER-TO-CENTER		MINIMUM SPACING CENTER-TO-CENTER	
(IPS) in.	Nominal (mm)	in.	mm	in.	mm
2	50	4	102	4	102
3	80	6	152	6	152
4	100	8	203	6.75	171
5	125	10	254	8.25	210
6	150	12	305	9.5	241
8	200	16	406	12	305

TUBING SIZE		STANDARD SPACING CENTER-TO-CENTER		MINIMUM SPACING CENTER-TO-CENTER	
(O.D.) in.	(O.D.) (mm)	in.	mm	in.	mm
3	76.2	6	152	5	127
4	101.6	8	203	6	152
5	127.0	10	254	7.5	191
6	152.4	12	305	9	229

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CONVECTION SECTION LAYOUT (Cont)

Some current vendor designs employ plug headers in special applications, such as delayed coker service. Consult a manufacturer's catalog or fired heater design consultant for the minimum center-to-center spacing if less than the standard 2 times IPS spacing is desired.

Tube Arrangement and Spacing (see Figure 2) - Although convection section tubes may be placed on either a triangular or rectangular pitch, an equilateral or isosceles triangular pitch is *always* used for fired heater convection sections. (Heat transfer coefficients for bare tubes on rectangular pitch have been included with the heat transfer equations since a rectangular pitch is common in offsite boilers). If further information on rectangular pitch convection sections is required, consult the Heat Transfer Equipment Section of EETD.

Tubes are usually placed on **equilateral** triangular pitch. However, where the number of tubes per horizontal row equals the number of passes, the tubes may be located on an **isosceles** triangular pattern. The base (tube-to-tube spacing in the horizontal row) may be varied to give the desired flue gas mass velocity.

When extended surface convection tubes are used, the minimum spacing between fin or stud tips on adjacent tubes is 1-1/2 in. (38.1 mm). This is in any direction and is based on minimum requirements for mechanical design of tube supports. This requirement limits the maximum length of extended surface that can be used for a given tube-to-tube spacing.

A minimum spacing of 3/4 in. (19.1 mm) should be provided between the fin or stud tip and the convection section side wall.

Corbelling - Corbelling must be provided for each convection section row to prevent bypassing of flue gas up the side of the convection section. **Figure 2d** shows an approximate layout of the corbelling and convection section tubes. The normal maximum width of corbelling is 6 in. (152 mm). The same size corbelling is usually used throughout the convection section.

Corbelling reduces the open area for flow of flue gases, increasing the flue gas mass velocity and heat transfer rate. Therefore, corbelling should not be provided on the *bottom* shield row, since no increase in heat transfer is desired on this row.

Tube Supports - The standard maximum unsupported length for horizontal fired heater tubes is 35 times the tube O.D. (except for steam cracker shield tubes, which require more support). Convection sections longer than this will require alloy intermediate tube sheets to reduce the unsupported length. For some cases with very high tube metal temperatures (e.g., Steam Crackers) the 35:1 rule is recognized as being excessive and shorter tube spans are required. Likewise, for lower tube metal temperatures this criteria may be overly restrictive. Refer to W. B. Bedesem's ER&E memorandum 52870, of April 19, 1973, in order to perform more accurate calculations if necessary.

PIPE SIZE		MAXIMUM UNSUPPORTED LENGTH	
(IPS) in.	Nominal (mm)	ft	m
4	100	13.1	4.0
5	125	16.2	4.9
6	150	19.3	5.9

Convection Section Cleaning - The requirement for sootblowers and attendant erosion protection of convection section refractory walls (see [Section VIII-H](#)) has a significant effect on fired heater investment. The convection section layout should take into consideration the optimum arrangement of sootblowers, tubesheets and tubes.

Consideration should be given to widening the convection section as a means of minimizing the number of sootblower rows required. The wider convection section permits the use of more tube surface per row, resulting in fewer vertical rows to be covered by sootblowers. Even though the total number of tubes is increased (lower flue gas mass velocity and heat transfer coefficient), the reduction in sootblower investment would more than compensate for the increased number of tubes. However, reducing the number of sootblowers by this means is often not possible. Conventional flue gas mass velocities should then be used.

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CONVECTION SECTION LAYOUT (Cont)

FLUE GAS MASS VELOCITY G_c

One of the main variables in developing the convection section layout is the flue gas mass velocity.

Calculation Procedure

$$G_c = \frac{W_f}{A_f} \quad \text{Eq. (1)}$$

where: G_c = Flue gas mass velocity through convection section (for heat transfer calculations), lb/sec ft² (kg/s m²)
 W_f = Flue gas rate, lb/sec (kg/s)
 A_f = Free cross-sectional area for flue gas flow, ft² (m²)

$$W_f = \frac{F_g FG}{3600} \quad \text{(Customary)} \quad \text{Eq. (2)}$$

$$W_f = F_g FG \quad \text{(Metric)} \quad \text{Eq. (2)M}$$

where: F_g = Gross fuel required, lb/hr (kg/s)
 FG = Lb flue gas/lb fuel (kg flue gas/kg fuel) (see [Section VIII-M](#))

$$A_f = L_c W_c - R \quad \text{Eq. (3)}$$

where: L_c = Convection section length, ft (m)
 W_c = Convection section width ft (m)
 R = Restriction to flow area for flue gas due to tubes, corbelling, and extended surface, ft² (m²) (see below)

1. Shield Section Restriction R_{sh}

$$R_{sh} = \frac{D_o}{12} (L_c N_t) \quad \text{(Customary)} \quad \text{Eq. (4a)}$$

$$R_{sh} = \frac{D_o}{1000} (L_c N_t) \quad \text{(Metric)} \quad \text{Eq. (4a)M}$$

where: R_{sh} = Restriction in shield section, ft² (m²)
 D_o = Tube outside diameter, in. (mm)
 N_t = Number of tubes per row

Although not accounted for in this calculation, corbelling should be specified for the second shield row.

2. Bare Tube Section Restriction R_b

$$R_b = R_{sh} + \frac{C}{12} (L_c) \quad \text{(Customary)} \quad \text{Eq. (4b)}$$

$$R_b = R_{sh} + \frac{C}{1000} (L_c) \quad \text{(Metric)} \quad \text{Eq. (4b)M}$$

where: R_b = Restriction in bare section (from tubes and corbelling), ft² (m²)
 C = Corbelling width, in. (mm)

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CONVECTION SECTION LAYOUT (Cont)

3. Studded Section Restriction R_s

$$R_s = R_b + \frac{h_s}{15} (L_c N_t) \quad (\text{Customary}) \quad \text{Eq. (4c)}$$

$$R_s = R_b + \frac{h_s}{1250} (L_c N_t) \quad (\text{Metric}) \quad \text{Eq. (4c)M}$$

where: R_s = Restriction in studded section, ft^2 (m^2)
 h_s = Stud length, in. (mm)

Note: Since studs in adjacent rows on the tube are staggered, the restriction of studded tubes to flue gas flow is based on only every other stud in the plane of the tube row acting as a restriction to flow. **Eq. (4c)** is based on 1/2 in. (12.7 mm) diameter studs on 5/8 in. (15.9 mm) pitch. A more general form of the equation is:

$$R_s = R_b + \left(\frac{h_s d_s}{12 p_s} \right) (L_c N_t) \quad (\text{Customary}) \quad \text{Eq. (4c1)}$$

$$R_s = R_b + \left(\frac{h_s d_s}{1000 p_s} \right) (L_c N_t) \quad (\text{Metric}) \quad \text{Eq. (4c1)M}$$

where: d_s = Stud diameter, in. (mm)
 p_s = Stud row pitch, in. (mm)

4. Finned Section Restriction R_f

$$R_f = R_b + \frac{h_f}{6} (t_f n_f L_c N_t) \quad (\text{Customary}) \quad \text{Eq. (4d)}$$

$$R_f = R_b + \frac{h_f}{500} \frac{(h_f L_c N_t)}{n_f} \quad (\text{Metric}) \quad \text{Eq. (4d)M}$$

where: R_f = Restriction in finned section, ft^2 (m^2)
 h_f = Fin height, in. (mm)
 n_f = Number of fins per in. (fin spacing, mm)
 t_f = Fin thickness, in. (mm)

Design Flue Gas Mass Velocity

- Where stack heights are based on *draft requirements*, recommended maximum flue gas mass velocities are found in the following tabulation. For combination firing applications, the heaviest liquid fuel is controlling.

FUEL	RECOMMENDED MAXIMUM FLUE GAS MASS VELOCITIES, G_c	
	lb/sec ft^2	kg/s m^2
Clean or Natural Gas	0.5 to 0.6	2.5 to 3.0
Low Metals Fuel Oil and Sour Fuel Gas	0.4 to 0.5	2.0 to 2.5
High Metals Fuel Oil (Pitch)	0.3 to 0.4	1.5 to 2.0

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CONVECTION SECTION LAYOUT (Cont)

These recommendations are for the portion of the convection section with the highest mass velocity (i.e., the longest extended surface). Mass velocities in sections of the convection section with bare tubes or shorter extended surface may fall below the recommended ranges.

These ranges of maximum mass velocity are considered optimum for convection section design and should be used if practical. Under certain conditions, as discussed below, higher mass velocities should be used. In other cases, lower mass velocities may be advantageous, such as to minimize the number of sootblowers by making the convection section wider, as previously discussed. However, design flue gas mass velocities below about 0.2 lb/sec ft² (1.0 kg/s m²) should be avoided, because of poor flue gas flow characteristics.

- If the flue gas mass velocity calculated above is higher than the recommended range for the particular fuel, the convection section layout should be revised. More tubes may be added to each convection section row or the tube to tube spacing in the row may be increased. In either case, the convection section width (Wc) will also increase. Preferentially, the number of tubes per row should be increased. However, the total number of tubes per row should be a multiple of the number of passes, i.e., a two-pass fired heater should have 4, 6, or 8 tubes per row in the convection section.

Under certain conditions, the number of tubes per row can deviate from this principle. For example, a 4-pass fired heater could have 6 tubes per row (**Figure 2c**). One pass would have two tubes in one row and one tube in the next row. The next pass would have one tube in the first row then two tubes in the next row, etc. However, care must be taken so that the total heat input to each pass is equal. Also note that this arrangement requires an equilateral triangular tube pitch.

Where the number of convection tubes per row equals the number of passes, as previously discussed, the tube spacing in the row may be varied to give the desired mass velocity. This spacing should be used for the entire convection section.

- Where stack heights are set by *pollution* considerations rather than draft requirements, flue gas mass velocities higher than those shown above are acceptable. However, in these cases the pressure drop across the convection section should never be more than one half the total draft provided by the stack.
- Fired heaters equipped with *induced draft fans* may be designed with higher flue gas mass velocities. However, the necessity for bypassing the fan during fan outages and sending flue gas directly to the stack should be considered. The optimum velocity should be based on an economic study of fan cost (investment and operating cost) versus convection section investment savings. In general, with induced draft fans, a mass velocity of 0.7 to 0.9 lb/sec ft² (3.5 to 4.4 kg/s m²) is optimum.
- In convection sections where flue gas flow is forced by a positive pressure, the mass velocity should also be higher. CO boilers are typical of this situation. Flue gas mass velocities in the range of about 0.8 to 1.2 lb/sec ft² (3.9 to 5.9 kg/s m²) should be considered. The convection section in these cases must be completely sealed to prevent hot flue gases leaking out through any openings.

CONVECTION SECTION DESIGN

CONVECTION SECTION DUTY

This equals the *total fired heater duty minus the radiant section duty*. Remember that the shield tubes (bottom two rows in convection section) have received part of their heat through radiation, in addition to the convection duty. The flue gas leaving the radiant section is considered to have given up its radiant heat to both the radiant tubes and shield tubes.

CALCULATION PROCEDURE

The procedure outlined below is used to calculate the required convection section surface. This procedure consists of a series of trial and error solutions. The Fired Heater Simulation Program 3558 may be used for these calculations.

The necessary equations for design of the convection section are presented following this calculation procedure.

Step 1 - Divide the convection section into smaller increments of several tube rows each. Start calculations at the shield section and work upward to coil inlet.

Step 2 - By a heat balance around the radiant section, the crossover temperature (oil side temperature out of shield section) can be determined. The shield radiant duty must be calculated and separated from the radiant section duty. The flue gas temperature into the shield section will be the bridgewall temperature previously determined.

Step 3 - For each new section being designed, calculate the maximum oil film temperature, tube metal temperature, and stud/fin tip temperature to insure they are not excessive. This should be done for the first row of each new material, extended surface, etc. used. These calculations should all be based on *entering flue gas and leaving oil temperatures* for that row.

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CONVECTION SECTION DESIGN (Cont)

Excessive temperatures cause the following:

1. Excessive oil film temperature: coking or product degradation.
2. Excessive stud/fin tip temperature: oxidation.
3. Excessive tube metal temperature: oxidation, corrosion, loss of strength.

Step 4 - If oil film, tube metal, or stud/fin tip temperatures are excessive, heat transfer should be reduced as follows:

1. For bare tubes, reduce flue gas mass velocity and/or increase inside film coefficient by reducing tube size and/or increasing tube spacing.
2. For extended surface tubes, reduce stud or fin height, reduce fin pitch, or use bare tubes. It is often necessary to use a number of bare tube rows above the shield rows to prevent excessive temperatures in the lowest row of extended surface tubes.

Step 5 - For each new section being designed, determine the inlet and outlet flue gas and oil side temperatures. The inlet flue gas and outlet oil side temperature are known from the previous section. Assume an oil side temperature entering the new section, and by heat balance, calculate the flue gas temperature leaving this section. (Alternatively, assume leaving flue gas temperature or heat absorbed, and calculate the others by heat balance.)

Plots of heat absorbed vs. fluid temperature and flue gas temperature are useful for this purpose.

Step 6 - Calculate the heat absorbed for this section, using the heat transfer equations given below based on average flue gas temperature, and check against the assumed heat absorbed or oil inlet temperature.

Step 7 - By trial and error, recalculate until the assumed heat absorbed or oil inlet temperature checks the calculated value.

Step 8 - Proceed to the next section of the convection section and repeat Steps 2 through 7 above. It is usually economical to use two or three different stud lengths in a single convection section, to maintain reasonable oil film and metal temperatures. For finned tubes, a number of combinations of fin height, fin density and/or materials can be used.

Step 9 - Proceed as above until sufficient surface has been provided to transfer the entire convection section duty.

Step 10 - Check the convection and radiant layout to see that piping to the convection section and from the radiant section are best suited for the overall unit plot plan.

HEAT TRANSFER EQUATIONS

Fired Heater Equations

1. Total Fired Heater Duty

$$Q_a = Q_r + Q_{sr} + Q_c \quad \text{Eq. (5)}$$

where: Q_a = Total fired heater heat duty, Btu/hr (W)
 Q_r = Heat absorbed by radiant section surface, Btu/hr (W)
 Q_{sr} = Shield section radiant heat duty, Btu/hr (W)
 Q_c = Total heat absorbed in convection section, excluding shield section radiant duty, Btu/hr (W)

2. Total Radiant Duty

$$Q_{tr} = Q_r + Q_{sr} \quad \text{Eq. (6)}$$

where: Q_{tr} = Total fired heater radiant duty, Btu/hr (W) (Note that bridgewall temperature (T_{bw}) is based on Q_{tr})

3. Total Convection Section Duty

$$Q_{tc} = Q_c + Q_{sr} \quad \text{Eq. (7)}$$

where: Q_{tc} = Total convection section heat duty, Btu/hr (W). (Note that Q_{tc} is the basis of the crossover temperature from the convection section to the radiant section.) Also note that Q_{tr} and Q_{tc} overlap, since both include the shield radiant duty, Q_{sr} .

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CONVECTION SECTION DESIGN (Cont)

Shield Section (Bottom Two Rows of Convection Section)

1. Total Heat Transfer

$$Q_s = Q_{sr} + Q_{sc} \quad \text{Eq. (8)}$$

where: Q_s = Total shield section duty Btu/hr (W)
 Q_{sc} = Heat transferred in shield section by convection, Btu/hr (W)

2. **Shield Section Radiant Duty Q_{sr}** - The shield tubes are directly exposed to the radiant section and absorb a portion of the total radiant duty. *This duty is included in the heat absorbed in cooling the flue gases down to the bridgewall temperature.*

$$Q_{sr} = L_c W_c \phi_s \quad \text{Eq. (9)}$$

where: L_c = Convection section length, ft (m)
 W_c = Convection section width, ft (m)
 ϕ_s = Radiant heat density on front face of shield tubes, Btu/hr ft² (W/m²)

For most process fired heaters, ϕ_s can be considered equal to the radiant section front face heat density:

3. Shield Tube Front Face Heat Density ϕ_s

$$\phi_s = C_1 \phi_{1\text{-side}} \quad \text{Eq. (10)}$$

where: $\phi_{1\text{-side}}$ = Average radiant section heat density based on one-side firing, Btu/hr ft² (W/m²)

$$\phi_s = \left[\frac{(T_{bw} + 460)^4 - (T_{mas} + 460)^4}{(T_{bw} + 655)^4 - (T_{ma} + 460)^4} \right] \phi_{la} F_{BL} \quad \text{(Customary)} \quad \text{Eq. (10a)}$$

$$\phi_s = \left[\frac{(T_{bw} + 273)^4 - (T_{mas} + 273)^4}{(T_{bw} + 381)^4 - (T_{ma} + 273)^4} \right] \phi_{la} F_{BL} \quad \text{(Metric)} \quad \text{Eq. (10a)M}$$

where: T_{bw} = Bridgewall temperature, °F (°C)
 T_{ma} = Length average radiant tube metal temperature, °F (°C)
 For steam reforming fired heaters, refer to [Section VIII-E](#) for T_{ma} . For process fired heaters, T_{ma} should be calculated on the basis of ϕ_r and the average bulk fluid temperature in the radiant section (uncoked), and assuming no maldistribution of heat input ($C_1 \cdot C_2 \cdot C_3 = 1.0$)
 T_{mas} = Length average shield tube metal temperature (on front 180 of all shield tubes), °F (°C)
 $= 0.866 T_{ms} - 0.266 T_{b1} + 0.4 T_{b2}$ Eq. (55)
 T_{ms} = Maximum shield tube metal temperature, °F (°C) (see below)
 T_{b1} = Process fluid temperature leaving the shield section, °F (°C) (crossover temperature)
 T_{b2} = Process fluid temperature entering the shield section, °F (°C)
 ϕ_{la} = Radiant section front face length average heat density, Btu/hr ft² (W/m²)
 $= C_1 \phi_r$
 ϕ_r = Radiant section average heat density, Btu/hr ft² (W/m²)
 C_1 = Peripheral maldistribution of heat around tube. See [Section VIII-B](#).
 C_2 = Vertical maldistribution of heat around tube. See [Section VIII-B](#).
 C_3 = Luminosity factor. See [Section VIII-B](#).
 F_{BL} = Mean beam length correction factor (Use 1.5)

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CONVECTION SECTION DESIGN (Cont)

Overall Heat Transfer Equation

Heat transfer in the convection section (except for shield radiant heat duty) is determined by the following equation:

$$q_c = U A (\text{LMTD}) \quad \text{Eq. (11)}$$

where: q_c = Heat transferred in the section being designed, Btu/hr (W)
 U = Overall heat transfer coefficient, Btu/hr ft² °F (W/m² °C)
 A = Total outside surface area for the section being designed, ft² (m²)
LMTD = Logarithmic mean temperature difference between flue gas and process fluid, °F (°C)

$$\text{LMTD} = \frac{(T_{g1} - T_{b1}) - (T_{g2} - T_{b2})}{\ln \left(\frac{T_{g1} - T_{b1}}{T_{g2} - T_{b2}} \right)} \quad \text{Eq. (12)}$$

where: T_{g1} = Flue gas temperature entering the section, °F (°C) (see **Figure 2a**)
 T_{b1} = Process fluid temperature leaving the section, °F (°C)
 T_{g2} = Flue gas temperature leaving the section, °F (°C)
 T_{b2} = Process fluid temperature entering the section, °F (°C)

Overall Heat Transfer Coefficient U

This coefficient takes into consideration the resistances of the fluid side film coefficient, the tube wall and the flue gas side coefficient.

$$\frac{1}{U} = \frac{1}{h_o} + \left(\frac{A_o}{A_i} \right) \left(\frac{1}{h_i} \right) + \left(\frac{A_o}{A_m} \right) \left(\frac{t_a}{k_m} \right) \quad \text{(Customary)} \quad \text{Eq. (13)}$$

$$\frac{1}{U} = \frac{1}{h_o} + \left(\frac{A_o}{A_i} \right) \left(\frac{1}{h_i} \right) + \left(\frac{A_o}{A_m} \right) \left(\frac{t_a}{k_m} \right) \times 10^{-3} \quad \text{(Metric)} \quad \text{Eq. (13)M}$$

where: U = Overall coefficient, Btu/hr ft² °F (W/m² °C)
 h_o = Outside film coefficient, Btu/hr ft² °F (W/m² °C)
 A_o = Total outside surface area, (includes extended surface, if any) ft²/ft (m²/m)
 A_i = Inside tube surface area, ft²/ft (m²/m)
 h_i = Inside film coefficient, Btu/hr ft² °F (W/m² °C)
 A_m = Area at mean tube diameter ft²/ft = 0.262 ($D_o - t_a$)
or metric [m²/m = 3.14 × 10⁻³ ($D_o - t_a$)] Eq. (14)

t_a = Average thickness of tube wall, in. (mm)

k_m = Thermal conductivity of tube wall, $\frac{\text{Btu/hr ft}^2}{\text{°F/in.}}$ (W/m °C)

(see [Section VIII-B](#))

D_o = Tube outside diameter, in. (mm)

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CONVECTION SECTION DESIGN (Cont)

AREA CALCULATIONS

$$A_o = A_e + A_b \quad \text{Eq. (15)}$$

where: A_o = Area of total outside surface, ft²/ft (m²/m)
 A_e = Area of extended surface on tube, ft²/ft (m²/m)
 A_b = Area of exposed bare tube surface, ft²/ft (m²/m)

1. Studded Tubes

$$A_e = 0.209 h_s n_s + 0.0262 n_s \quad (\text{Customary}) \quad \text{Eq. (16)}$$

$$A_e = 2.51 \times 10^{-3} h_s n_s + 7.98 \times 10^{-3} n_s \quad (\text{Metric}) \quad \text{Eq. (16)M}$$

$$A_b = 0.262 D_o - 0.0262 n_s \quad (\text{Customary}) \quad \text{Eq. (17)}$$

$$A_b = 3.14 \times 10^{-3} D_o - 7.98 \times 10^{-3} n_s \quad (\text{Metric}) \quad \text{Eq. (17)M}$$

where: D_o = Tube outside diameter, in. (mm)
 h_s = Stud length, in. (mm)
 n_s = Number of studs per circular row

Note: **Eqs. (16) and (17)** are valid for 1/2 in. (12.7 mm) diameter studs on circular rows with 5/8 in. (15.9 mm) pitch. General equations are:

$$A_e = 0.262 \left(\frac{d_s}{p_s} \right) h_s n_s + 0.0654 \left(\frac{d_s^2}{p_s} \right) n_s \quad (\text{Customary}) \quad \text{Eq. (16a)}$$

$$A_e = 3.14 \times 10^{-3} \left(\frac{d_s}{p_s} \right) h_s n_s + 7.85 \times 10^{-4} \left(\frac{d_s^2}{p_s} \right) n_s \quad (\text{Metric}) \quad \text{Eq. (16a)M}$$

$$A_b = 0.262 D_o - 0.0654 \left(\frac{d_s^2}{p_s} \right) n_s \quad (\text{Customary}) \quad \text{Eq. (17a)}$$

$$A_b = 3.14 \times 10^{-3} D_o - 7.85 \times 10^{-4} \left(\frac{d_s^2}{p_s} \right) n_s \quad (\text{Metric}) \quad \text{Eq. (17a)M}$$

where: d_s = Stud diameter, in. (mm)
 p_s = Stud row pitch, in. (mm)

CONVECTION SECTION DESIGN (Cont)

2. Finned Tubes

$$A_e = 0.131 n_f (D_f^2 - D_o^2 + 2 D_f t_f) \quad (\text{Customary}) \quad \text{Eq. (18)}$$

$$A_e = \left(1.57 \times 10^{-3} / n_f \right) \cdot (D_f^2 - D_o^2 + 2 D_f t_f) \quad (\text{Metric}) \quad \text{Eq. (18)M}$$

$$A_b = 0.262 D_o (1 - t_f n_f) \quad (\text{Customary}) \quad \text{Eq. (19)}$$

$$A_b = 3.14 \times 10^{-3} D_o (1 - t_f / n_f) \quad (\text{Metric}) \quad \text{Eq. (19)M}$$

where: n_f = Number of fins per in. (fin spacing, mm)
 D_f = Fin outside diameter, in. (mm)
 D_o = Tube outside diameter, in. (mm)
 t_f = Fin thickness, in. (mm)

$$D_f = D_o + 2h_f \quad \text{Eq. (20)}$$

where: h_f = Fin height, in. (mm)

▶ The table below shows the equivalent extended surface heights between thick fins and studs for various pipe sizes. The following standard dimensions are used for thick fins: height of 3/4 in. (19 mm), spacing of 3 fins per in. (8.5 mm), and a thickness of 0.105 in. (2.7 mm). Half-inch (12.7 mm) stud diameters are assumed having the standard 5/8 in. (15.9 mm) stud pitch. This table should be applied to convection section geometries in oil-fired heaters, where thick fins and studs are typically used.

NOM. / OD		THICK FINS (A_o)		STUDS: EQUIVALENT h_s (@ n_s)	
(in.)	(mm)	ft ² / ft	sq m / m	(in.)	(mm)
3 / 3.500	76.2 / 88.9	6.047	1.844	2.041 (12)	51.9 (12)
4 / 4.500	101.6 / 114.3	7.487	2.283	2.152 (14)	54.7 (14)
5 / 5.563	127.0 / 141.3	9.017	2.749	2.006 (18)	50.9 (18)
6 / 6.625	152.4 / 168.3	10.547	3.215	1.913 (22)	48.6 (22)
8 / 8.625	203.2 / 219.1	13.426	4.093	1.904 (28)	48.4 (28)

INSIDE FILM COEFFICIENT h_i

▶ 1. Single Phase

For single-phase turbulent flow heat transfer calculations, the Dittus Boelter correlation is used. The equation is rearranged and grouped into two terms that describe the process data and the geometric data.

$$h_i = 0.023 \text{Re}_D^{0.8} \text{Pr}^{0.4} \frac{k}{D} \quad \text{Eq. (21)}$$

$$h_i = \phi_1 \phi_2 \quad \text{Eq. (21a)}$$

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CONVECTION SECTION DESIGN (Cont)

$$\phi_1 = 18.6 \frac{G^{0.8}}{D^{0.2}} \quad \text{Eq. (21b)}$$

$$\phi_1 = 23.0 \frac{G^{0.8}}{D^{0.2}} \quad \text{Eq. (21b)M}$$

$$\phi_2 = k^{0.6} \left(\frac{C_p}{\mu} \right)^{0.4} \quad \text{Eq. (21c)}$$

where: Re_D = Reynolds number based on the inside diameter, D, dimensionless
 Pr = Prandtl number, dimensionless
 G = Mass velocity of process fluid, lb/s-ft² (kg/s m²)
 D = Inside diameter of tube, in. (mm)
 k = Thermal conductivity of process fluid, Btu/hr ft °F (W/m °C)
 C_p = Specific heat of process fluid, Btu/lb °F (kJ/kg °C)
 μ = Dynamic viscosity of process fluid, cP (mPa s)

3. Two-Phase

For two-phase flow, the heat transfer coefficients are found by taking the mass average of the single phase coefficients.

- Determine the all-liquid coefficient, assuming that the total bulk fluid is liquid. Use the single-phase equation using the liquid properties of the process fluid.
- Determine the all-vapor coefficient, assuming that the total bulk fluid is vapor. Use the single-phase equation using the vapor properties of the process fluid.
- Calculate the two-phase coefficient, as a weighted average of (a) and (b), using weight fractions of vapor and liquid:

$$h_{tp} = X h_v + (1 - X) h_\ell \quad \text{Eq. (21d)}$$

where: h_{tp} = Two-phase inside heat transfer coefficient, Btu/hr ft² °F (W/m² °C)
 X = Weight fraction of vapor in the fluid
 h_ℓ = Liquid film coefficient, based on no vaporization at bulk temperature, Btu/hr ft² °F (W/m² °C)
 h_v = Vapor film coefficient, based on 100% vaporization at bulk temperature, Btu/hr ft² °F (W/m² °C)

Note: This method is recommended for calculating *design* coefficients for two-phase flow. Assuming satisfactory fluid flow patterns, the actual coefficient at any given point *may* be significantly higher than either the all-liquid or all-vapor coefficient. However, it is not practical to use these anticipated higher coefficients for design. For special cases requiring better definition of coefficients, consult Report No. EE.6ER.71 or the Heat Transfer Equipment Section of EETD.

OUTSIDE (FLUE GAS SIDE) FILM COEFFICIENT h_o

This coefficient takes into account both radiation and convection.

1. For Bare Tubes

$$h_o = h_c + h_r \quad \text{Eq. (22)}$$

where: h_o = Outside film coefficient, Btu/hr ft² °F (W/m² °C)
 h_c = Heat transfer coefficient for convection, Btu/hr ft² °F (W/m² °C) [see [Eq. \(23\)](#)]
 h_r = Heat transfer coefficient for radiation, Btu/hr ft² °F (W/m² °C) [see [Eqs. \(24a\)](#) and [\(24b\)](#)]

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CONVECTION SECTION DESIGN (Cont)

For *triangular pitch*

$$h_c = 5.4 \left[\frac{G_c^{0.67}}{D_o^{0.33}} \right] \left[\frac{T_g + 460}{100} \right]^{0.40} \quad (\text{Customary}) \quad \text{Eq. (23)}$$

$$h_c = 30.7 \left[\frac{G_c^{0.067}}{D_o^{0.33}} \right] \left[\frac{T_g + 273}{55.6} \right]^{0.40} \quad (\text{Metric}) \quad \text{Eq. (23)M}$$

where: G_c = Flue gas mass velocity, lb/sec ft² (kg/s m²)
 D_o = Tube outside diameter, in. (mm)
 T_g = Flue gas temperature, °F (°C)
 (Use *entering* flue gas temperature for calculating maximum metal and oil film temperature, *average* flue gas temperature for calculating heat absorbed in the section.)

For *shield section*,

$$h_r = 0 \quad \text{Eq. (24a)}$$

(Accounted for in shield section radiant heat duty Q_{sr} .)

For *convection section*,

$$h_r = F_1 F_2 \quad \text{Eq. (24b)}$$

where: F_1 = Temperature Factor, based on average flue gas and tube metal temperatures in the bank. (Assume average tube metal temperature is 50°F (28 °C) hotter than fluid temperature.) See **Figure 3**.

F_2 = Convection Section Characteristic Factor, a function of tube geometry and flue gas properties (L_b) ($P_{CO_2+H_2O}$). See **Figures 4 and 5**.

$P_{CO_2+H_2O}$ = Partial pressure of $CO_2 + H_2O$ in the flue gas Eq. (25)

For *triangular pitch*,

$$L_b = 0.0903 \left(\frac{S_R S_T}{D_o} \right) - 0.0708 D_o = 0.0903 \frac{S_T \sqrt{S_L^2 - (S_T/2)^2}}{D_o} - 0.0708 D_o \quad (\text{Customary}) \quad \text{Eq. (26)}$$

$$L_b = 1.08 \times 10^{-3} \left(\frac{S_R S_T}{D_o} \right) - 8.50 \times 10^{-4} D_o = 1.08 \times 10^{-3} \frac{S_T \sqrt{S_L^2 - (S_T/2)^2}}{D_o} - 0.0708 D_o \quad (\text{Metric}) \quad \text{Eq. (26)M}$$



CONVECTION SECTION DESIGN (Cont)

where: L_b = Mean beam length of convection bank, ft (m)
 S_R = Distance between centers of tube rows, in. (mm)
 S_T = Distance between tube centers in horizontal row, in. (mm)
 S_L = Distance between tube centers in adjacent rows, in. (mm)
For tubes on 2 IPS equilateral triangular pitch, L_b is as follows:

D_o		$S_L = S_T$		L_b	
in.	mm	in.	mm	ft	m
4.5	114.3	8	203	0.79	0.241
5.563	141.3	10	254	0.96	0.293
6.625	168.3	12	305	1.16	0.354
8.625	219.1	16	406	1.62	0.494

For rectangular pitch, use

$$h_c = 4.3 \left[\frac{G_c^{0.67}}{D_o^{0.33}} \right] \left[\frac{T_g + 460}{100} \right]^{0.40} \quad \text{(Customary)} \quad \text{Eq. (23a)}$$

$$h_c = 24.4 \left[\frac{G_c^{0.67}}{D_o^{0.33}} \right] \left[\frac{T_g + 273}{55.6} \right]^{0.40} \quad \text{(Metric)} \quad \text{Eq. (23a)M}$$

For rectangular pitch, use

$$L_b = 0.108 \left(\frac{S_R S_T}{D_o} \right) - 0.085 D_o \quad \text{(Customary)} \quad \text{Eq. (26a)}$$

$$L_b = 1.30 \times 10^{-3} \left(\frac{S_R S_T}{D_o} \right) - 1.02 \times 10^{-3} D_o \quad \text{(Metric)} \quad \text{Eq. (26a)M}$$

2. For Extended Surface

$$h_o = h_e \left[\frac{EA_e + A_b}{A_o} \right] \quad \text{Eq. (27)}$$

where: h_o = Outside film coefficient for tube plus its extended surface, Btu/hr ft² °F (W/m² °C)
 h_e = Film coefficient for extended surface [see **Eqs. (28a), (28b), and (28c)**], Btu/hr ft² °F (W/m² °C)
 E = Extended surface efficiency factor. See **Figure 6** or **7a** for studded tubes, or **Figure 8** for finned tubes.

For fully-studded tubes,

$$h_e = 7 \left[\frac{G_c^{0.55}}{D_o^{0.45}} \right] \left[\frac{T_g + 460}{100} \right]^{0.48} \quad \text{(Customary)} \quad \text{Eq. (28a)}$$

$$h_e = 71.3 \left[\frac{G_c^{0.55}}{D_o^{0.45}} \right] \left[\frac{T_g + 273}{55.6} \right]^{0.48} \quad \text{(Metric)} \quad \text{Eq. (28a)M}$$

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CONVECTION SECTION DESIGN (Cont)

For *half-studded* tubes,

$$h_e = 8.5 \left[\frac{G_c^{0.55}}{D_o^{0.45}} \right] \left[\frac{T_g + 460}{100} \right]^{0.48} \quad (\text{Customary}) \quad \text{Eq. (28b)}$$

$$h_e = 86.5 \left[\frac{G_c^{0.55}}{D_o^{0.45}} \right] \left[\frac{T_g + 273}{55.6} \right]^{0.48} \quad (\text{Metric}) \quad \text{Eq. (28b)M}$$

For *finned* tubes,

$$h_e = 4.0 \left[\frac{G_c^{0.55}}{D_o^{0.45}} \right] \left[\frac{T_g + 460}{100} \right]^{0.48} \quad (\text{Customary}) \quad \text{Eq. (28c)}$$

$$h_e = 40.70 \left[\frac{G_c^{0.55}}{D_o^{0.45}} \right] \left[\frac{T_g + 273}{55.6} \right]^{0.48} \quad (\text{Metric}) \quad \text{Eq. (28c)M}$$

CALCULATION OF MAXIMUM TUBE METAL TEMPERATURE

The following equations for calculating tube metal temperatures include factors to account for maldistribution of heat transfer around the tubes.

$$T_m = T_{b1} + \Delta T_f + \Delta T_m \quad \text{Eq. (29)}$$

where: T_m = Maximum tube metal temperature, °F (°C)
 T_{b1} = Oil temperature leaving section (i.e., the maximum bulk fluid temperature), °F (°C)
 ΔT_f = Temperature rise across inside oil film, °F (°C)
 ΔT_m = Temperature rise across tube wall, °F (°C)

1. Shield Tubes

$$\Delta T_f = [U' (T_{bw} - T_{b1}) + \phi_s] \frac{A_o}{A_i} \frac{1}{h_i} \quad \text{Eq. (30)}$$

$$\Delta T_m = [U' (T_{bw} - T_{b1}) + \phi_s] \frac{A_o}{A_m} \frac{t_a}{k_m} \quad (\text{Customary}) \quad \text{Eq. (31)}$$

$$\Delta T_m = [U' (T_{bw} - T_{b1}) + \phi_s] \frac{A_o}{A_m} \frac{t_a \times 10^{-3}}{k_m} \quad (\text{Metric}) \quad \text{Eq. (31)M}$$

$$\frac{1}{U'} = \frac{1}{1.5 h_c} + \frac{A_o}{A_i} \frac{1}{h_i} + \frac{A_o}{A_m} \frac{t_a}{k_m} + f_e \quad (\text{Customary}) \quad \text{Eq. (31a)}$$

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$$\frac{1}{U'} = \frac{1}{1.5 h_c} + \frac{A_o}{A_i} \frac{1}{h_i} + \frac{A_o}{A_m} \frac{t_a \times 10^{-3}}{k_m} + f_e \quad (\text{Metric}) \quad \text{Eq. (31a)M}$$

where: T_{bw} = Bridgewall temperature, °F (°C)
 U' = Local coefficient for shield tubes, based on T_{bw} and T_{b1} , Btu/hr ft² °F (W/m² °C). A maldistribution factor of 1.5 is used to account for nonuniform heat transfer around the tubes. It is applied only to the outside heat transfer coefficient, where this behavior occurs.
 ϕ_s = Shield tube front face radiant heat density, as previously calculated, Btu/hr ft² (W/m²)
 f_e = external resistance due to fouling, ft² °F/Btu/hr (m² °C/W)

2. Bare Tubes

$$\Delta T_f = U' (T_{g1} - T_{b1}) \frac{A_o}{A_i} \frac{1}{h_i} \quad \text{Eq. (32)}$$

$$\Delta T_m = U' (T_{g1} - T_{b1}) \frac{A_o}{A_m} \frac{t_a}{k_m} \quad (\text{Customary}) \quad \text{Eq. (33)}$$

$$\Delta T_m = U' (T_{g1} - T_{b1}) \frac{A_o}{A_m} \frac{t_a}{k_m} \times 10^{-3} \quad (\text{Metric}) \quad \text{Eq. (33)M}$$

where: T_{g1} = Flue gas temperature entering section, °F (°C)
 U' = Local coefficient for bare tubes being investigated, based on T_{g1} , Btu/hr ft² °F (W/m² °C). For all practical purposes, it has the same form as the U previously calculated.

3. Extended Surface Tubes

$$\Delta T_f = U' (T_{g1} - T_{b1}) \frac{A_o}{A_i} \frac{1}{h_i} \quad \text{Eq. (34)}$$

$$\Delta T_m = U' (T_{g1} - T_{b1}) \frac{A_o}{A_m} \frac{t_a}{k_m} \quad (\text{Customary}) \quad \text{Eq. (35)}$$

$$\Delta T_m = U' (T_{g1} - T_{b1}) \frac{A_o}{A_m} \frac{t_a}{k_m} \times 10^{-3} \quad (\text{Metric}) \quad \text{Eq. (35)M}$$

where: U' = Local overall coefficient, Btu/hr ft² °F (W/m² °C)

a. Local Overall Coefficient U'

$$\frac{1}{U'} = \frac{1}{h_o'} + \frac{A_o}{A_i} \frac{1}{h_i} + \frac{A_o}{A_m} \frac{t_a}{k_m} \quad (\text{Customary}) \quad \text{Eq. (36)}$$

$$\frac{1}{U'} = \frac{1}{h_o'} + \frac{A_o}{A_i} \frac{1}{h_i} + \frac{A_o}{A_m} \frac{t_a}{k_m} \times 10^{-3} \quad (\text{Metric}) \quad \text{Eq. (36)M}$$

where: h_o' = Local extended surface coefficient, Btu/hr ft² °F (W/m² °C)

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b. Local Extended Surface Coefficient h_o'

$$h_o' = h_e' \left[\frac{E' A_e + A_b}{A_o} \right] \quad \text{Eq. (37)}$$

where: h_e' = Local extended surface film coefficient (based on T_{g1})

$$h_e' = f_m h_e \quad \text{Eq. (38)}$$

where: E' = Local extended surface efficiency factor, based on h_e' .

See **Figure 6** or **7A** for studs, **Figure 8** for fins.

f_m = Extended surface maldistribution factor.

Recommended values: fins $f_m = 1.5$, studs $f_m = 1.9$, half-studs $f_m = 6$.

CALCULATION OF STUD OR FIN TIP TEMPERATURE

$$T_t = T_{g1} - E'' (T_{g1} - T_m) \quad \text{Eq. (39)}$$

where: T_t = Maximum stud or fin tip temperature, °F (°C)

E'' = Local extended surface temperature factor, based on h_e' .

See **Figures 6** or **7B** for studs, **Figure 9** for fins.

ALLOWANCE FOR INTERNAL COKING AND EXTERNAL FOULING

An allowance for coking is not normally included in convection section heat transfer calculations, even in coking services. Likewise, for new designs, an allowance for external fouling of convection section tubes is not normally included in convection section calculations. It is assumed that for instances where fouling fuels are used that the normally provided sootblowers will keep the external surfaces clean.

However, on occasion, the design must include provision for a layer of internal coke or external fouling. Examples include all-convection fired heaters, such as CO heaters; or severe high-temperature coking services, such as thermal crackers. Additional terms for the coke layer and the external fouling layer must be included in the overall heat transfer coefficient as well as in tube metal temperature calculations:

$$\frac{1}{U_c} = \frac{1}{h_o} + \frac{A_o}{A_i'} \frac{1}{h_i} + \frac{A_o}{A_c} \frac{t_c}{k_c} + \frac{A_o}{A_m} \frac{t_a}{k_m} + f_e \quad (\text{Customary}) \quad \text{Eq. (40)}$$

$$\frac{1}{U_c} = \frac{1}{h_o} + \frac{A_o}{A_i'} \frac{1}{h_i} + \frac{A_o}{A_c} \frac{t_c \times 10^{-3}}{k_c} + \frac{A_o}{A_m} \frac{t_a}{k_m} \times 10^{-3} + f_e \quad (\text{Metric}) \quad \text{Eq. (40)M}$$

where: U_c = Overall coefficient in coked and/or fouled condition, Btu/hr ft² °F (W/m² °C)

A_i' = Area at inside diameter of coke layer, ft²/ft (m²/m)

$$= 0.262 (D_i - 2t_c) \quad (\text{Customary}) \quad \text{Eq. (41a)}$$

$$= \left[3.14 \times 10^{-3} (D_i - 2t_c) \right] \quad (\text{Metric}) \quad \text{Eq. (41a)M}$$

A_c = Area at mean diameter of coke layer, ft²/ft (m²/m)

$$= 0.262 (D_i - 2t_c) \quad (\text{Customary}) \quad \text{Eq. (41b)}$$

$$= \left[3.14 \times 10^{-3} (D_i - t_c) \right] \quad (\text{Metric}) \quad \text{Eq. (41b)M}$$



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- D_i = Inside tube diameter, based on average wall thickness, in. (mm)
 f_e = External fouling factor, hr ft² °F/Btu (m² °C/W)
 k_c = Thermal conductivity of coke, $\frac{\text{Btu/hr ft}^2}{^\circ\text{F/in.}}$ (W/m °C) (see [Section VIII-B](#))
 t_c = Design coke thickness, in. (mm)

Note: The latest release of the Fired Heater Simulation Program 3558 allows one to enter the coke resistance as a separate item for convection section tubes. The program calculates an equivalent thermal conductivity and equivalent tube thickness that accounts for coke deposits in the convection section. It effectively "lumps" the coke deposit and tube wall into an equivalent tube thickness.

The value of f_e is normally less than 0.1 hr ft² °F/Btu (0.0176 m² °C/W). The default value is 0.0 for new designs.

$$\frac{1}{k_m'} = \frac{A_m'}{A_c} \frac{t_c}{t_a'} \frac{1}{k_c} + \frac{A_m'}{A_m} \frac{t_a}{t_a'} \frac{1}{k_m} \quad \text{Eq. (41c)}$$

where: k_m' = "Equivalent" tube conductivity, $\frac{\text{Btu/hr ft}^2}{^\circ\text{F/in.}}$ (W/m °C)

$$t_a' = t_a + t_c \quad \text{Eq. (41d)}$$

$$A_m' = 0.262 (D_o - t_a') \quad (\text{Customary}) \quad \text{Eq. (41e)}$$

$$\left[A_m' = 3.14 \times 10^{-3} (D_o - t_a') \right] \quad (\text{Metric}) \quad \text{Eq. (41e)M}$$

where: t_a' = "Equivalent" tube wall thickness, in. (mm)
 A_m' = Area at "equivalent" mean tube diameter, ft²/ft (m²/m)

STACK DESIGN CONSIDERATIONS

TYPES OF CONSTRUCTION AND ARRANGEMENT

Stacks used for process fired heaters are usually made of steel or concrete. Brick stacks are rarely used, although many concrete stacks have brick liners. Stacks may be placed on the ground beside the fired heater (ground-supported stacks) or may be mounted on top of the fired heater itself (fired heater-supported stacks).

Ground-Supported Stacks - Below a height of about 250 ft (75 m), these stacks are usually made of steel; taller stacks are usually of concrete.

Ground-supported stacks are often used to serve several fired heaters. If the stack height is based on draft requirements only, the fired heater system with the highest pressure drop will determine the stack height.

If a fired heater tied into a common stack is to be shut down and work carried out inside the fired heater while other fired heaters using the stack continue to operate, provisions such as blinds or guillotine dampers in the ducting should be made to isolate this fired heater from the operating fired heaters.

Since additional pressure losses are encountered in getting the flue gases from the fired heater exit over to the stack, ground-supported stacks must be taller than fired heater-supported stacks, if stack height is determined by draft requirements only.

Fired Heater-Supported Stacks - These are always made of steel. The maximum economical height for fired heater-supported stacks is 150 to 200 ft (45 to 60 m) above grade. Above 200 ft (60 m), a ground-supported stack should be specified.

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Stack Take-Off - One fired heater exit is required for each 40 ft (12.2 m) of convection section tube length. For longer convection sections, additional take-offs are required. Multiple fired heater-supported stacks may be used, or the take-offs may be ducted to a common stack which can be either fired heater-supported or ground-supported.

STACK DIAMETER

The diameter is usually a function of the quantity of flue gas flowing. Stacks should usually be designed for a velocity of 25 ft/sec (7.6 m/s). Unless details of the stack are known, for design purposes the diameter should be considered uniform for the entire length. The design specification should state the inside diameter required at the stack exit. The following considerations may also affect the stack diameter:

Fired Heater Supported Stacks - Where flue gases pass directly into the stack, the diameter should be no greater than the outside width (about 12 in. or 300 mm greater than the inside width) of the convection section. Where flue gases enter the stack from ducts (**Figure 12A**) the stack may have a diameter somewhat greater than the outside width of the convection section.

Pollution Control Requirements - When relatively high stacks are required by pollution control considerations or by local regulations, stack gas velocities higher than 25 ft/sec (7.6 m/s) may be economical, because the extra draft to compensate for the extra pressure drop is available anyway. High velocities (*instead of or in addition to* tall stacks) may also be required on occasion for pollution control considerations. When a restricted stack exit (choke cone) is used to obtain this increased velocity, the pressure drop through the choke cone may place the top portion of the stack under positive internal pressure, if the exit velocity is much above 25 ft/sec (7.6 m/s). This can cause damage to concrete or brick stacks, and a steel liner would then probably be required. When the exit velocity is a nominal 25 ft/sec (7.6 m/s) or less, the exit loss is relatively small, and the effect of a choke cone on the stack can be ignored.

Minimum Stack Gas Velocity - At very low flue gas velocities in the stack, much lower than 10 ft/sec (3 m/s), inversion can occur, with cold air spilling into the stack and running down one side. This recirculation could possibly reduce the available draft and should be avoided. Where extended operations of the fired heater at extremely low throughputs are expected, the following measures should be considered:

1. **Operate at Higher Excess Air Rate** - During operation at low throughput, the percent excess air can be increased as necessary, to avoid recirculation if this actually becomes a problem. This solution is usually preferable to increasing the *design* stack gas velocity, and is particularly recommended for cases where stack height is based on draft requirements. In such cases, increasing the design stack gas velocity would require increasing the stack height. Another reason for accepting higher than design percent excess air during operation at low throughput is that, due to control difficulties and air leakage into the fired heater, it is probably not practical to operate at low excess air rates under these conditions.
2. **Removable Choke Plate** - A removable choke cone on the stack can be considered, as a means of increasing stack exit gas velocity during initial operation at low throughput. The cone would be removed when the fired heater throughput is increased.

STACK LININGS

Refractory linings are not required in a stack when the flue gas contains less than 5 vppm of SO₃ or if the fuel contains less than 0.25% by weight of sulfur. In these circumstances, a bare steel stack could be constructed even if its metal temperature is expected to be below the acid dewpoint of the flue gas. Its metal temperature must, however, be above the water dewpoint. In cold climates, ambient temperatures may cause the metal temperature to drop below the water dewpoint. The formation of carbonic acid and rust could damage the steel shell. For this reason, bare steel stacks are usually found in refineries having warmer climates.

When the flue gas contains more than 5 vppm of SO₃, carbon steel stacks and steel liners in concrete chimneys may be at risk from acid corrosion. Corrosion will occur if the temperature of the metal in contact with the flue gases falls below the acid dewpoint of the flue gas. As a rule-of-thumb, the metal temperature of a steel stack (uninsulated and with no lining) is midway between the flue gas temperature and the ambient temperature. All carbon steel stacks require linings as required to maintain the shell temperature below 900°F (480°C). A layer of 3-inch (76.2 mm) medium weight castable refractory is common.

External insulation could also be used to keep metal temperatures above the acid dewpoint. It is generally not used because of the increased thermal stresses/growth in the steel shell, potential problems with insulation maintenance, and under-insulation corrosion. Furthermore, external insulation does not provide any protection from an internal fire.

Special consideration should be taken when designing stacks that will operate under steam-air decoking conditions, where the flue gas into the stack is approximately 1000°F (538°C). As the metal temperature increases, there is a reduction of strength in the stack material (typically carbon steel) and a thicker wall may be required. So it may be preferable to internally line the stack with refractory, which will reduce the metal temperature and the required wall thickness.

To protect nearby personnel, furnace-supported bare stacks should be lined internally, up to the elevation where personnel access instrumentation (stack damper elevation). Bare stacks may also have a protective screen that prevents personnel from coming into contact with the potentially hot wall.

