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Revision Memo

- 12/00** The highlights of this revision are:
Editorial and typographical corrections. Deleted outdated references. Corrected equation for Heat Release Factor (HRF). Eliminated Table 1 stress values for micro alloyed 25 Cr - 35 Ni-Nb.

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SCOPE

This section provides the procedures to be used in designing an Exxon-type steam reformer heater. The design considerations and limits and the mechanical details to be included in the Design Specification are summarized. Instructions for using the Steam Reformer Design Program 3676 are also included.

Although the calculation procedures are set up specifically for an Exxon-type reformer, with modifications to specific constants, they can be applied to other reformer designs as well. For more information, contact the Heat Transfer Equipment of EMRE Technology Department.

The Exxon reformer design, presented in this Design Practice, is project specific and, in certain applications, may provide cost advantages over designs offered by contractors. However, the economics of each reformer must be evaluated before specific designs can be adopted. To protect this proprietary return of the ExxonMobil technology, care must be taken to ensure that contractors and fired heater vendors are given information on the ExxonMobil reforming heater only after they have signed appropriate confidentiality agreements.

REFERENCES

DESIGN PRACTICES (BESIDES OTHER SECTIONS OF SECTION VIII)

Section XII	Instrumentation
Section XV	Safety Facilities

INTERNATIONAL PRACTICES

IP 3-4-1	Piping for Fired Equipment
IP 7-1-1	Fired Heaters
IP 7-4-1	Burners for Fired Heaters
IP 7-5-1	Coils for Fired Heaters and Boilers
IP 15-1-1	Instrumentation for Fired Heaters
IP 18-3-2	Statically Cast Steel and Alloy Pressure Containing Parts, and Tube Supports, for Fired Heaters
IP 18-3-3	Centrifugally Cast High Alloy Tubes for Fired Heaters
IP 19-1-1	Paint and Protective Coatings
IP 19-3-3	Linings for Fired Heaters

OTHER REFERENCES

EMRE, *Refinery Construction Materials Manual*, Manual No. EETD 028
 ASME Standard B31.3, *Process Piping*
Hydrogen Plant Operating Guide, ER&E Report EE.43E.84, May 1984

BACKGROUND

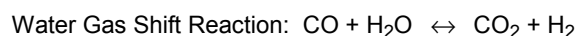
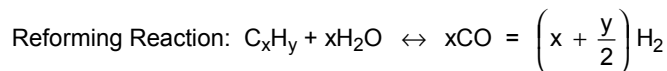
Steam reformers are used in refineries to produce 95 - 97% purity hydrogen for consumption in hydrotreating units. They are also used in chemical plants in a variety of hydrogen producing capacities. Their design and operation are critical because heater operating conditions are at the limit of the best available materials both for radiant tubes and outlet connection components.

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BACKGROUND (Cont)

STEAM REFORMING PROCESS

In the catalytic reforming process, a mixture of steam and hydrocarbon is partially converted to hydrogen and carbon oxides by the reforming reaction and the water gas shift reaction.



In the dominant reaction, the reforming reaction, hydrocarbon reacts with the steam to form carbon monoxide and hydrogen. This reaction is highly endothermic. In the water gas shift reaction, the carbon monoxide reacts with the steam to form more hydrogen and carbon dioxide. This reaction is exothermic. Under the conditions normally selected for reforming, the unreacted hydrocarbon is mainly methane. The overall reaction is highly endothermic.

With present-day technology and existing catalysts, it is normally economical to steam reform at temperatures of 1350°F (730°C) to 1650°F (900°C), pressures of 300 psig (2050 kPa) to 500 psig (3500 kPa), and at steam to carbon ratios ranging from 3:1 to 9:1. High temperature, low pressure and high steam to carbon ratios favor hydrogen production, while low temperature, high pressure and low steam to carbon ratios favor methane production. The operating conditions that are best suited for a particular plant must be determined on the basis of economics, which weigh operating costs against investments.

The feed to a reformer must be of high purity to avoid catalyst deactivation leading to coking and irreversible catalyst deterioration. It should contain less than 0.5% olefins, less than 0.2 wppm sulfur, less than 0.2 wppm (mg/kg) chlorides or other halogens, and less than 0.1 wppm (mg/kg) heavy metals.

Coking can also result when the carbon formation reactions proceed faster than the reforming and shift reactions. Carbon can be formed either by methane cracking to form carbon and hydrogen or by carbon monoxide reverting to form carbon and carbon dioxide. This usually results from operation at low steam to carbon ratios, high film temperatures, or high concentrations of carbon monoxide in the feed.

REFORMING HEATERS

The ExxonMobil reformer design is advantageous over some of the designs provided by contractors. As mentioned previously, careful consideration must be given to the design requirements on a project-by-project basis. **Figure 1** represents the various heater arrangements commonly observed throughout the ExxonMobil circuit.

Reformer heaters are typically described by the position of the burners within the radiant section. Wall-fired reformers, with a large number of small burners (**Figure 1A**), provides a large degree of flexibility in setting the radiant heat flux profile. However, a considerable amount of operator attention is required to maintain the burners and obtain the optimum firing pattern. Terrace wall furnaces, (**Figures 1B and 1C**) attempt to maintain an even heat flux profile along the length of the radiant tubes with less burners. Down-fired units, **Figure 1D**, are becoming more prevalent, as certain process streams require the maximum heat input to be provided at the radiant tube inlets.

The ExxonMobil design features floor firing with a relatively small number of long flame burners (see **Figure 2**). Floor firing permits the use of two rows of tubes per radiant box and the comparatively small number of burners, mostly located at grade, reduces the amount of operator attention required.

DESIGN CONSIDERATIONS

FEEDSTOCK AND CATALYST TYPE

Natural gas and refinery fuel gases have been the feedstocks to the majority of ExxonMobil hydrogen plants. The reforming catalysts for these feeds are well proven and readily available. For naphtha feedstocks proprietary catalysts and an operating license from the vendor may be required.

Wall-fired reformer heaters have been used in the past for all types of feedstock due to the large amount of flexibility in setting the heat flux profile. However, due to the large number of burners and associated operator attention, this type of heater is no longer specified for new units. Up and terrace wall-fired heaters are typically chosen for lighter feedstocks, such as natural gas and refinery fuel gas. Use of these types of heaters limit the amount of heat input at the radiant tube inlet and reduces the possibility of carbon formation due to methane cracking. Also, these feedstocks tend to have a large proportion of H_2 at the radiant inlet, adding heat at this point would be somewhat ineffective. Down-fired units have mostly been applied to feedstocks such as LPG or naphtha. These units provide more heat at the radiant tube inlet to compensate for the lower inlet temperatures of the mixed feedstock and are required to sustain the reaction.



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DESIGN CONSIDERATIONS (Cont)**S COIL OUTLET TEMPERATURE**

The coil outlet temperature is set as high as practical based on the materials available and the economics of the reformer heater design. Most of the reformers specified for ExxonMobil hydrogen plants have had coil outlet temperature of 1450°F (790°C) to 1550°F (843°C). With higher coil outlet temperatures, there are savings in operating costs in downstream equipment and a lower steam to carbon ratio can be used in the reformer.

An outlet temperature of 1550°F (843°C) is near the operating limit of the Incoloy 800 HT and Hastelloy S materials typically used in outlet pigtails and pigtail connections. These materials may be substituted by Haynes Alloy 230, for heaters requiring higher coil outlet temperatures. However, if a coil outlet temperature above 1550°F (843°C) is being considered, the Materials Engineering Section of EMRE should be contacted regarding the latest industry experience in this area.

Any Design Basis Specification or Duty Specification shall have a risk assessment performed whenever the Coil Outlet Temperature of a Steam Reformer Furnace is specified in excess of 1550°F (843°C). Following this review, a formal NPQC review of materials, design details, and configuration for the steam reformer outlet pigtails, outlet header, and waste heat boiler attachment details shall be conducted.

COIL OUTLET PRESSURE

Most hydrogen consumed in Exxon refineries is used at a pressure of 900 psig (6200 kPa) or higher. Consequently, there is an incentive to manufacture hydrogen at the highest practical pressure in order to minimize compression costs.

On the other hand, higher pressures also increase the stresses on the radiant tubes. Because of the high operating temperature, even a small increase in the stress on the tubes can result in a large increase in the wall thickness required.

The coil outlet pressure used is an economic balance between compression costs and reformer tube costs. For most ExxonMobil hydrogen plants built to date, a coil outlet pressure of 300 - 400 psig (2140 - 2740 kPa) has proven economical.

INLET STEAM / HYDROCARBON MIXTURE TEMPERATURE

It is desirable to design for the highest permissible inlet temperature to the catalyst tubes. This minimizes the radiant section heat duty and, therefore, the amount of radiant tube surface required. Upper limits are specified to minimize olefin formation which can lead to catalyst coking. The following are recommended design inlet temperatures for the steam / hydrocarbon mix.

SERVICE	COMMON PREHEAT COIL FOR STEAM PLUS HYDROCARBON	DEDICATED HTSH ⁽¹⁾ COIL ONLY FOR PROCESS STEAM
Natural Gas	1000°F (540°C)	1150°F (620°C)
Refinery Gas (negligible C ₃ & heavier)	925°F (500°C)	1050°F (565°C)
C ₃ or C ₄ , LPG	850°F (455°C) ⁽²⁾	950°F (510°C)
Naphtha	850°F (455°C) ⁽²⁾	850°F (455°C)

Notes:

(1) HTSH; High Temperature Super Heater.

(2) Hydrocarbon feed is bypassed around feed/steam preheat coil during startup.

The coil inlet temperature can be maximized if the increased temperature is provided only by the process steam, i.e., if there is no preheating of the hydrocarbon stream in the convection section downstream of the feed desulfurization facilities. This requires the convection section to have a split superheater coil. A split superheater coil, with the high temperature superheater coil occupying the lower portion of the convection section, is preferable to designs which have a single superheater coil and the hydrocarbon stream in the convection section.

CATALYST VOLUME

The minimum catalyst volume required in the fired zone is set based on the space velocity of the reformer feed through the catalyst at the inlet to the reformer. The space velocity is based both on the feed inlet flow rate and composition. This limit is set by the process designers in accordance with input from the catalyst vendors in reaction kinetics.

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DESIGN CONSIDERATIONS (Cont)

OPTIMUM HEATER DESIGN

The optimum heater design is an economic balance between the volume of radiant tube metal, the number of radiant tubes and connection components, and the tube spacing. To a large extent, the optimum design is the one which meets the catalyst volume and pressure drop requirements yet minimizes the pounds of radiant tube material. The other variables are significant, however, and should not be ignored. The designer should do an economic evaluation of the relative effect of each on his particular design.

PRESSURE DROP

The pressure drop through the catalyst packed tubes should be designed for approx. 1 psig/ft (23 kPa/m) of catalyst tube length as this provides a margin over the limit of 2 psi/ft (45 kPa/m). This is to prevent crushing of the catalyst rings with subsequent blockage and tube overheating resulting from process gas maldistribution among the radiant tubes. Although there are indications that this limit could be increased, more data and test work are required to confirm this. For designs where the pressure drop is limiting, the Heat Transfer Equipment Section of EETD should be contacted for the latest information regarding pressure drop limits.

RADIANT HEAT DENSITY

The maximum demonstrated heat densities at the inside tube wall surface must not be exceeded. These densities are:

Gas Catalysts:	Peak	25,000 - 26,000 Btu/hr-ft ²	(78 - 82 kW/m ²)
	Average	20,000 - 22,000 Btu/hr-ft ²	(63 - 69 kW/m ²)
Ceramic Based Catalysts:	Peak	23,000 - 25,000 Btu/hr-ft ²	(72 - 79 kW/m ²)
	Average	20,000 Btu/hr-ft ²	(63 kW/m ²)

RADIANT TUBE LENGTH

Longer radiant tubes are normally economical in a steam reformer because this reduces the number of tubes required and therefore reduces the length of the heater. Because of the smaller number of tubes, the mass velocity is also higher, thereby increasing the inside heat transfer coefficient and decreasing the tube metal temperature. The pressure drop, however, will be higher. With the ExxonMobil design, the maximum fired length is around 35 ft (10.7 m), as limited by the heat release pattern from the floor-mounted burners. Thirty-five ft (10.7 m) is normally specified for an ExxonMobil reformer, however, shorter tubes to a minimum of 31 ft (9.5 m), may be used if economical. More recently, vendors have provided tube lengths up to 41 ft (12.5 m), however, these furnaces have been designed with terrace wall burners to provide a uniform heat flux along the tube length.

Although higher axial dead load stresses occur with longer heater tubes, these stresses are highest in the colder top region of the tube and are small compared to pressure stresses. For the ExxonMobil design, dead load stresses should not influence the selection of tube length.

RADIANT TUBE MATERIAL

In the past, centrifugally cast HK-40 was used for radiant section tubes. Since about 1980, a niobium modified HP-45 has been used. This material contains more alloy and on a per-pound basis is more expensive than HK-40; however, because of its superior creep strength it can be used in a thinner wall design making its cost competitive. From a process standpoint, the thinner wall design provides more volume for catalyst for a given outside diameter. This can improve conversion and also result in lower tube metal temperatures. More recently, an improved micro-alloyed version of the niobium modified HP-45 material has been used. This alloy contains small additions of elements such as Ti, W, B and others that enhance the creep strength. Each foundry has developed specific micro-alloyed additions in order to enhance creep strength. **Table 1** provides a list of allowable stresses for centrifugally cast radiant section tubes. It should be noted that the allowable stresses for the niobium modified HP-45 and the micro-alloyed niobium modified HP-45 are based on an assessment of heat of material supplied by several recognized foundries. For individual foundries, the supplied alloy may provide a higher or lower strength. It is important to verify the strength by requesting results of tests from representative heats from and individual foundry. It is also possible to use higher allowable stresses if a foundry can substantiate it with creep data from representative heats. In all cases, foundries supplying tubes must be qualified by EMRE to ensure proper quality control procedures are in place and the supplied tubes will meet allowable stress requirements.

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DESIGN CONSIDERATIONS (Cont)

TUBE DIAMETER

Generally, the economic inside tube diameter is the smallest that requires a wall thickness for stress which is greater than the minimum required for casting integrity. This minimizes the total volume of radiant tube metal required and, since the tubes are purchased per pound, minimizes the total tube cost. Smaller diameter tubes require more tubes with no decrease in wall thickness permitted. Larger diameter tubes usually require a rapidly increasing wall thickness to satisfy stress requirements.

Since the cost of installing the tubes and the cost of the inlet and outlet fittings is significant, a somewhat larger tube diameter can sometimes be justified. The designer must weigh the cost of a thicker wall for the larger tubes against the savings possible for the reduction in the number of tubes required.

The inside diameter of the tube should be no less than 3.5 in. (90 mm) or five times the effective catalyst particle diameter. This is to ensure uniform catalyst packing. It is essential that the catalyst tubes be packed uniformly since voids or bridging in the catalyst will cause local overheating of the tube and rapid failure. The effective catalyst diameter, d_p , is listed in **Figure 12** for the commonly used catalyst sizes.

TUBE WALL THICKNESS

The sound tube wall thickness is calculated to limit the pressure stress to the minimum stress that would produce rupture in 100,000 hours. The specified tube wall includes the minimum sound wall thickness plus allowances on the inside and outside diameters for casting tolerance and unsound metal. The minimum "as cast" wall thickness permitted is 0.40 in. (10 mm) as required to obtain reasonable casting integrity. For tubes which are machined internally, this would result in a minimum final tube wall thickness of 0.335 in. (8.5 mm). (Also see **IP 18-3-3**.) The maximum total wall thickness permitted is 0.80 in. (20 mm) as required to prevent excessive temperature gradients and thermal stresses through the tube wall.

We recommend the inside surface of the radiant tubes be machined as experience suggests there is if no significant surcharge for machined tubes. Machined tubes are preferable since they are thinner for a given sound wall thickness, they facilitate good catalyst loading, and provide more volume for the catalyst per foot of tube length. Machined tubes are also conducive to the non-destructive inspection techniques used to determine remaining tube life. They also permit better quality assurance that the specified sound wall thickness is met. For HK-40 tubes, the reduction in tube metal temperature which results from machining often permits a further reduction in wall thickness which is large enough to compensate for all or part of the machining cost.

TUBE SPACING

Tube spacing is an economic balance between thicker tube walls for a small spacing and a longer heater for a large spacing. As the tubes are spaced closer together, they partially shield each other. This causes a greater temperature maldistribution around the tube causing a higher maximum tube metal temperature for the same heat absorbed. For the higher tube metal temperature, a thicker tube wall is required.

Past studies have indicated a 1.8 diameter spacing is optimum for HK-40 tubes. For 25 Cr - 35 Ni-Nb alloy, a closer spacing is more economic. However, since in the past HK-40 tubes have been used exclusively in ExxonMobil reformers, there is limited ExxonMobil experience with closer tube spacings. For the present, a minimum tube spacing of 1.6 diameters or 7.5 in. (190 mm) should be used, whichever is larger. The 7.5 in. (190 mm) spacing represents a practical limit on the spacing of the hemispherical caps on the outlet header.

RADIANT SECTION TUBE DESIGN

As discussed above, the radiant section design centers on the selection of the radiant tubes. The accurate calculation of tube metal temperature profile and required tube metal thickness is a lengthy double trial-and-error procedure, in which both the radiating temperature and process gas composition along the tube must be determined. For this reason, Computer Program 3676 is used for the design of the radiant section of our reformer heaters.

COMPUTER CALCULATIONS

The input to Program 3676 consists of an assumed heater geometry and desired process conditions. Procedures are presented below for estimating tube wall thickness and for calculating the radiant exchange factor, both of which are needed input.

The program computes gas composition, pressure drop, heat density, tube stress, and actual and allowable tube metal temperatures along the tube. The heater geometry can then be modified as necessary to meet tube metal temperature and pressure drop criteria.

Since Program 3676 is easy and inexpensive to use, several logical combinations of tube diameter and number of tubes can be checked. The guidelines presented above under Design Considerations should be followed in determining which combinations to check.

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RADIANT SECTION TUBE DESIGN (Cont)

The following variables are suggested as a starting point for designing a reforming heater. The other variables can then be calculated, including number of radiant tubes, approximate tube metal temperature, and tube wall thickness.

where: L = Radiant Tube Length = 35 ft (10.7 m)
 D_i = Tube inside diameter = 3.5 in. (90 mm)
 (assuming $D_i / d_p > 5$)
 ϕ_{ri} = Average radiant heat density on inside tube surface = 20,000 Btu/hr ft² (63 kW/m²)

Customary	Metric
$\phi_r = 20,000 \frac{D_i}{D_o}$	$\phi_r = 63 \frac{D_i}{D_o}$

F = Radiant exchange factor = 0.53

The number of radiant tubes per box should be divisible by four, two rows of tubes with two tubes per tube support. More recent designs, offered by vendors allow four tubes to be held by a single support. In this case, the number of radiant tubes per box should be divisible by six.

CALCULATION PROCEDURE

The following stepwise procedure should be used for radiant tube designs. The first 4 steps must be followed to develop the input to Program 3676. The remainder of the procedure is consistent with that used by Program 3676. The necessary equations are presented following this procedure.

- The required process conditions include feed rate and composition, coil inlet and outlet temperature, coil outlet pressure, outlet reforming approach temperature and outlet shift approach temperature. The temperature at which reforming begins must also be specified (usually 1150°F) (620°C).
- Determine the radiant section geometry. This includes the number of tubes, tube diameter and length, tube spacing, and radiant exchange factor.
- Divide the tube length into the desired number of sections. Ten sections are assumed in Program 3676.
- Select the radiating temperature profile. This depends upon the heater configuration. **Figure 4** can be used for the ExxonMobil reformer heater (or consult the Heat Transfer Equipment Section of EETD). Program 3676 will default to the profile for an ExxonMobil reformer if no other is specified.
- Determine the composition and enthalpy of the process stream as a function of temperature. The outlet composition is determined by the outlet temperature, pressure, steam to carbon ratio, and approach to reforming equilibrium. The inlet composition is known up to the point at which reforming is assumed to begin, 1150°F (620°C). A pseudo approach to reforming equilibrium is calculated for 1150°F (620°C) as described below. A third order curve is then fitted between the calculated approach at 1150°F (620°C) and the approach specified at the coil outlet. Intermediate reforming approach temperatures are determined from this curve and are used to calculate the composition for the corresponding temperature. The pseudo approach temperature at 1150°F (620°C) is determined by trial and error. An approach temperature is assumed. The corresponding composition of CO, CO₂, CH₄, and H₂ is then calculated. If the enthalpy of the pseudo composition matches the enthalpy of the feed, then the assumed approach is correct. If not, a new approach must be assumed. This is a long and tedious hand calculation, therefore, Program 3676 is normally used even when hand calculations are to follow.
- Prepare a graph of enthalpy vs. temperature.
- Assume average front face tube metal temperature T_{ff1} in Section 1.
- Using the assumed radiating gas temperature T_g (Step 4) and tube temperature T_{ff1} , calculate the front face radiant heat density ϕ_{ff1} in Section 1 [**Eq. (1a)**].
- Calculate heat input Q_{r1} in Section 1 [**Eq. (3)**].
- Determine bulk fluid outlet temperature t_{b1} from Section 1 using the enthalpy graph from Step 6.
- Calculate average bulk fluid temperature T_1 in Section 1 (average of inlet and outlet temperatures).
- Calculate inside film coefficient h_{i1} in Section 1 based on T_1 [**Eq. (4)**].
- Using the assumed T_{ff1} (Step 7), T_1 (Step 11) and h_{i1} (Step 12), calculate ϕ_{ff1} .
- Compare ϕ_{ff1} from Steps 8 and 13. Repeat Steps 7 to 13 until ϕ_{ff1} checks.
- Repeat Steps 7 to 14 for Section 2, and so on for each section.



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RADIANT SECTION TUBE DESIGN (Cont)

16. After calculation of all sections, if the coil outlet temperature equals the required outlet temperature, the temperature profile is correct. If the calculated outlet temperature does not equal the required outlet temperature, raise or lower the entire radiating gas temperature profile (Step 4) by approximately 20°F (10°C) and repeat the calculations starting with Step 7 until the outlet temperatures agree. For process streams containing 60% H₂ or more, the radiating gas profile, provided in **Figure 4**, may be shifted up or down by as much as 100 - 200°F (38 - 90°C).
17. Calculate a profile of maximum tube metal temperature T_m in each section. This must be done to determine where the combination of temperature and pressure (tube stress) is limiting [Eq. (5)].

RADIANT SECTION DESIGN EQUATIONS

RADIANT SECTION FRONT FACE HEAT DENSITY

$$\phi_{ff1} = \sigma F \left[(T_g + 460)^4 - (T_{ff} + 460)^4 \right] \quad (\text{Customary}) \quad \text{Eq. (1a)}$$

$$\phi_{ff1} = \sigma F \left[(T_g + 273)^4 - (T_{ff} + 273)^4 \right] \quad (\text{Metric}) \quad \text{Eq. (1a)M}$$

where: ϕ_{ff1} = Radiant heat density on the front face of the tube, Btu/hr ft² (kW/m²)
 σ = Stefan Boltzman constant
= 0.1714 x 10⁻⁸ Btu/hr ft² °R⁴ (Customary) [56.7 x 10⁻¹² kW/m² K⁴ (Metric)]
F = Radiant exchange factor
T_g = Radiating gas temperature, °F (°C)
T_{ff} = Front face tube metal temperature, °F (°C)

$$\phi_{ff} = \frac{(T_{ff} - T)}{\left(\frac{t}{k_m} \right) \left(\frac{2D_o}{D_i + D_o} \right) + \frac{1}{h_i} \left(\frac{D_o}{D_i} \right)} \quad (\text{Customary}) \quad \text{Eq. (1b)}$$

$$\phi_{ff} = \frac{(T_{ff} - T)}{\left(\frac{t}{k_m} \right) \left(\frac{2D_o}{D_i + D_o} \right) + \frac{1}{h_i} \left(\frac{D_o}{D_i} \right)} \frac{1}{1000} \quad (\text{Metric}) \quad \text{Eq. (1b)M}$$

where: T_{ff} = Front face tube metal temperature, °F (°C)
T = Average bulk fluid temperature, °F (°C)
D_o = Outside tube diameter, in. (mm)
D_i = Inside tube diameter, in. (mm = D_o - 2t)
t = Total tube wall thickness, in. (mm)
h_i = Tube inside heat transfer coefficient, Btu/hr ft² °F (W/m²°C)
k_m = Thermal conductivity of tube wall, $\frac{\text{Btu/hr ft}^2}{^\circ\text{F/in.}} \left(\frac{\text{W/m}^2}{^\circ\text{C/mm}} \right)$

See **Figure 16** in **Section VIII-B**.



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RADIANT SECTION DESIGN EQUATIONS (Cont)

RADIANT SECTION AVERAGE HEAT DENSITY

$$\phi_r = \frac{\phi_{ff}}{C_1} \quad \text{(Customary)} \quad \text{Eq. (2)}$$

$$\phi_r = \frac{\phi_{ff}}{C_1} \quad \text{(Metric)} \quad \text{Eq. (2)M}$$

where: ϕ_r = Radiant section average heat density, Btu/hr ft² (kW/m²)
 C_1 = Dimensionless correction factor for peripheral maldistribution of heat input around the tube. See **Figure 3**.

RADIANT SECTION HEAT DUTY Q_r

$$Q_r = \phi_r A_r \quad \text{(Customary)} \quad \text{Eq. (3)}$$

$$Q_r = \frac{\phi_r A_r}{1000} \quad \text{(Metric)} \quad \text{Eq. (3)M}$$

where: Q_r = Heat absorbed by the radiant section surface, Btu/hr (MW)
 A_r = Radiant section outside area, ft² (m²)

TUBE INSIDE HEAT TRANSFER COEFFICIENT h_i

$$h_i = \frac{C k_f}{D_i} \left(\frac{G d_p}{\mu} \right)^{0.62} \left(\frac{C_p \mu}{k_f} \right)^{0.30} \left(\frac{1}{\exp \left(\frac{4 d_p}{D_i} \right)} \right) \left(\frac{1}{\exp (\Sigma)} \right) \quad \text{Eq. (4)}$$

where: h_i = Inside heat transfer coefficient, Btu/hr ft² °F (W/m² °C)
 C = Constant, for Customary units = 160.127 (Metric units = 2350)

k_f = Gas thermal conductivity, $\frac{\text{Btu/hr ft}^2}{\text{°F ft}}$ (W/m °C)

See **Figure 5** for approximate value

μ = Gas viscosity, cP (mPa•s). See **Figure 6** for approximate value.

$\frac{C_p \mu}{k_f}$ = Prandtl No. See **Figure 7** for approximate value.

Note: **Figures 5, 6, and 7** provide typical reformer gas properties and should be used for hand calculations of inside coefficients. Deviations from actual properties will not have a significant effect on the calculation of inside coefficient.

G = Mass velocity, based on empty tube cross-sectional area, lb/sec ft² (kg/s m²)

d_p = Effective catalyst particle diameter, in. (mm). See **Figure 12**.

D_i = Tube inside diameter, in. (mm)

Σ = Reaction factor. This is an empirical factor to account for the effect of endothermic reactions on heat transfer coefficient. Use $\Sigma = 1$ up to the temperature at which reforming begins [usually 1150°F (620°C)]. The value of Σ at the coil outlet temperature is given in **Figure 8**. Between the temperature at which reforming begins and coil outlet, Σ is linear. In some cases, the catalyst has been activated to allow it to reform the feed at temperatures below 1150°F (620°C). For these cases, consult the Heat Transfer Equipment Section of EETD for the correct value of Σ .

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RADIANT SECTION DESIGN EQUATIONS (Cont)

S MAXIMUM TUBE METAL TEMPERATURE T_m

$$T_m = T_b + \Delta T_f + \Delta T_m \quad \text{Eq. (5)}$$

$$\Delta T_f = \left(\frac{1}{h_i} \right) \left(\frac{D_o}{D_i} \right) (\phi_r)_{\max} \quad \text{Eq. (5a)}$$

$$\Delta T_m = \left(\frac{t}{k_m} \right) \left(\frac{2 D_o}{D_o + D_i} \right) (\phi_r)_{\max} \quad \text{Eq. (5b)}$$

- where:**
- T_m = Maximum tube metal temperature, °F (°C) (no margin should be added)
 - T_b = Temperature of bulk fluid at the point of calculation, °F (°C)
 - ΔT_f = Temperature rise between bulk fluid and tube wall, °F (°C)
 - ΔT_m = Temperature rise across tube wall, °F (°C)
 - D_o = Outside tube diameter, in. (mm)
 - D_i = Inside tube diameter, in. (mm) = $D_o - 2t$
 - t = Actual tube wall thickness, (sound wall plus applicable casting allowances) in. (mm)
 - h_i = Tube inside heat transfer coefficient, Btu/hr ft² °F (W/m² °C)
 - k_m = Thermal conductivity of tube wall, $\frac{\text{Btu/hr ft}^2}{\text{°F/in.}} \left(\frac{\text{W/m}^2}{\text{°C/mm}} \right)$

See **Figure 16** in **Section VIII-B**.

- $(\phi_r)_{\max}$ = Maximum point heat density, based on outside tube diameter and corrected for longitudinal and peripheral maldistribution of heat, Btu/hr ft² (kW/m²)

$$(\phi_r)_{\max} = C_1 C_2 \phi_r \quad \text{Eq. (5c)}$$

OR

$$= C_2 \phi_{ff}$$

- where:**
- C_1 = Dimensionless correction factor for peripheral maldistribution around the tube. C_1 varies with tube spacing, diameter and thickness. See **Figure 3**.
 - C_2 = Dimensionless correction factor for vertical maldistribution. This is determined by the detailed tube calculations (Program 3676). See below for value to be used in determining the approximate tube metal temperature.
 - ϕ_r = Radiant section average heat density, Btu/hr ft² (kW/m²)
 - ϕ_{ff} = Radiant section front face heat density, Btu/hr ft² (kW/m²) = $C_1 \phi_r$

APPROXIMATE TUBE METAL TEMPERATURE T_m

For a quick estimate of the maximum tube metal temperature, the following constants may be used in **Eq. (5)**. **Note:** Design calculations should not be based on this estimate, since a small increase in the maximum tube metal temperature can result in a large decrease in tube life.

- T_b = Coil outlet temperature, °F (°C)
- ϕ_r = Radiant section average heat density, Btu/hr ft² (kW/m²)
- C_2 = 0.86
- h_i = Film coefficient should be based on outlet properties and temperature

This approximate maximum tube temperature can be used as the basis for determining the initial estimate of tube thickness for Program 3676.

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RADIANT SECTION DESIGN EQUATIONS (Cont)

TUBE WALL THICKNESS t

Radiant tubes are centrifugally cast and purchased to the specifications of [IP 18-3-3](#). The tube material will be either HK-40, 25 Cr - 35 Ni-Nb, or microalloyed 25 Cr - 35 Ni-Nb depending on the economics of the particular design. Although the 25 Cr - 35 Ni-Nb alloys are more expensive per pound than HK-40, at high-reforming severity the higher alloys are more economical because thinner tubes can be used.

The “as cast” tube wall thickness is calculated as follows (per [IP 7-1-1](#)):

$$t_{AC} = t_{sw} + a + c \quad \text{Eq. (6)}$$

where: t_{AC} = “As cast” tube wall thickness, in.
= 0.40 in. (10 mm) minimum

$$t_{sw} = \frac{PD_o}{2.0 S_a + P}$$

P = Internal pressure, psig, at the point on the tube where the actual stress is closest to the allowable stress. This is calculated by Program 3676 and is usually about 8 ft (2.5 m) from the outlet for ExxonMobil reformers. If this is not known, or for initial input to Program 3676, use a pressure 10 psig [70 kPa (g)] greater than the expected coil outlet pressure.

D_o = Tube outside diameter, in. (mm)

S_a = Allowable stress, psi (kPa). The allowable stress is the minimum stress required to produce rupture in 100,000 hrs. See [Table 1](#) for allowable stresses. This basis is expected to give a tube life of about 10 years.

a = Allowance for unsound metal and casting tolerance, equals 0.10 in. (2.54 mm) (0.065 in. (1.65 mm) on the inside of the tube and 0.035 in. (0.89 mm) on the outside)

c = Corrosion allowance, if required, in. (mm). Usually $c = 0$.

RADIANT EXCHANGE FACTOR F

The radiant exchange factor determines what flue gas temperature is required in the firebox to achieve a desired radiant heat density. Its value depends on the ratio of reradiating refractory surface to cold tube surface. For the ExxonMobil reformer, its value is usually between 0.52 and 0.55 depending on the size of the heater.

The exchange factor used in the design of the ExxonMobil reformer is the average of the exchange factors for the three heater zones defined in [Figure 9](#).

$$F = \frac{F_1 + 2F_2 + F_3}{4} \quad \text{Eq. (7)}$$

where: F = Overall radiant exchange factor

F_1, F_2, F_3 = Radiant exchange factors for Zones 1, 2 and 3, respectively



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RADIANT SECTION DESIGN EQUATIONS (Cont)

The radiant exchange factor for each zone is determined from **Figure 11** as a function of the following:

B = Mean beam length, ft (m). The mean beam length for Zones 1 and 3 is equal to the distance from the tube row center-line to the side wall. The mean beam length for Zone 2 is equal to the distance between the two tube rows. See **Figure 9**.

Excess air. Use 15% excess air for radiant section design and determine ε for use in **Figure 11**.

$$\varepsilon = \frac{\text{Refractory surface area}}{\text{Equivalent cold plane area}} = \frac{A_w}{\alpha A_{cpr} + A_{cps}} \quad \text{Eq. (8)}$$

where: A_w = Exposed refractory surface area, ft² (m²). This includes the exposed refractory surface on the floor, arch and end walls, and for Zones 1 and 3 the side wall also.
 αA_{cpr} = Equivalent cold plane area of the radiant tubes, ft² (m²)
 α = Effectiveness factor of the tube plane, from **Figure 10**
N = Number of radiant tubes in the zone (do not include the shield tubes in determining **N** for Zone 2)
c = Center-to-center spacing of adjacent tubes in a row, ft (mm)
L = Exposed length of the tubes, ft (m) ($\alpha A_{cpr}{}_1$ is usually equal to $(\alpha A_{cpr})_3$
 $(\alpha A_{cpr})_2$ usually equals 2 times $(\alpha A_{cpr})_1$)
 A_{cps} = Equivalent cold plane area of the shield tubes, ft² (m²)
= 0 for Zones 1 and 3
= $L_c W_c$ for Zone 2 (0 for offset convection sections)
 L_c = Convection section length, ft (m)
 W_c = Convection section width, ft (m)

Radiant Catalyst Tube Pressure Drop ΔP

The pressure drop per foot of catalyst in the packed radiant tubes is determined by **Eq. (9)**. Inlet and outlet losses are not included. The pressure drop should be calculated for each tube section used in the design of the radiant tube (usually 10 sections), with the total pressure drop equal to the sum of the pressure drops in individual sections. Program 3676 calculates pressure drop in this manner. If a shaped catalyst is used within the tubes, the total pressure drop in the tube is = 0.9 ΔP .

$$\frac{\Delta P}{L} = \frac{0.00848 G^{1.9} \mu^{0.1} J}{\rho} \quad \text{(Customary)} \quad \text{Eq. (9)}$$

$$\frac{\Delta P}{L} = \frac{0.151 G^{1.9} \mu^{0.1} J}{\rho} \quad \text{(Metric)} \quad \text{Eq. (9)M}$$

where: ΔP = Pressure drop across the tube, psi (kPa)
L = Radiant tube length, ft (m)
G = Fluid mass velocity based on empty tube cross-sectional area, lb/sec ft² (kg/s m²)
 μ = Gas viscosity, cP (mPa•s) (from **Figure 6**)
 ρ = Fluid density, lb/ft³ (kg/m³)
J = A geometrical term dependent on catalyst size and tube diameter

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RADIANT SECTION DESIGN EQUATIONS (Cont)

It can be determined from **Figure 12** for six common catalyst sizes, or it can be calculated by:

$$J = \frac{\lambda^{1.1}(1 - \omega)}{(d_p)^{1.1} \omega^3} \quad \text{Eq. (10)}$$

where: d_p = Effective catalyst particle diameter, in. (mm). The diameter of a sphere having the same volume as the catalyst ring (see **Figure 12**).

$$= 1.145 \left[(d_o^2 - d_i^2) S \right]^{1/3} \quad \text{(Customary)} \quad \text{Eq. (10a)}$$

$$= 29.08 \left[(d_o^2 - d_i^2) S \right]^{1/3} \quad \text{(Metric)} \quad \text{Eq. (10a)M}$$

$$\lambda = \text{Particle shape factor}$$

$$0.205 AV^{-0.667} \quad \text{Eq. (10b)}$$

ω = Void fraction of packed tube

$$= \omega_c + \frac{(1 - \omega_c) d_i^2}{d_o^2} \quad \text{Eq. (10c)}$$

A = Total surface of catalyst ring including surface area of hole, sq in. (mm²)

d_i = Catalyst ring inside diameter, in. (mm)

d_o = Catalyst ring outside diameter, in. (mm)

d_s = Equivalent sphere diameter, in. (mm). This is the diameter of the sphere having the volume of a solid cylinder of the same length and outside diameter as the catalyst ring (see **Figure 12**).

$$= 1.145 d_o^{2/3} S^{1/3} \quad \text{(Customary)} \quad \text{Eq. (10d)}$$

$$= 29.08 d_o^{2/3} S^{1/3} \quad \text{(Metric)} \quad \text{Eq. (10d)M}$$

D_i = Inside diameter of catalyst tube, in. (mm)

S = Length of catalyst ring, in. (mm)

V = Volume of ring, in.³ (mm³)

$$0.785 (d_o^2 - d_i^2) S \quad \text{(Customary)} \quad \text{Eq. (10e)}$$

$$12900 (d_o^2 - d_i^2) S \quad \text{(Metric)} \quad \text{Eq. (10e)M}$$

ω_c = Solid cylinder void fraction, from **Figure 13**

Approximate Catalyst Tube Pressure Drop ΔP

A simplified procedure for calculating the approximate pressure drop of a catalyst packed tube is to take the sum of the following:

1. Calculate the pressure drop per foot of tube at inlet conditions [by **Eq. (9)**] and multiply by 1/3 the radiant tube length.
2. Calculate the pressure drop per foot of tube at outlet conditions and multiply by 2/3 the radiant tube length.

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HEATER DESIGN

RADIANT SECTION LAYOUT

The following dimensions should be used in determining the overall furnace configuration (also see **Figure 2**):

1. The distance from the center of the floor burners to the tube center-line should be at least 36 in. (915 mm). From the center of floor burners to the side walls should be at least 24 in. (610 mm).
2. The distance between the end radiant tubes and the end wall should be at least 18 in. (460 mm) to give proper balance of refractory surface and radiant tube surface at the ends of the heater.
3. Burner location will not affect the overall dimensions of the furnace, but is critical to proper operation of the heater. The end burner in a row of floor burners should be approximately adjacent to the second from the end tube in a tube row (i.e., about 27 in. (685 mm) from the end wall).
4. The centerline of the radiant tube row should be at least 33 in. (840 mm) from the inside (hot) face of the convection wall to allow sufficient room for radiant tube hangers.

FUEL

A relatively high grade fuel is required in steam-reforming heaters. Gas fuels are almost always used. The maximum sulfur content of the fuel should be 1.5 wt%. See the ExxonMobil *Refinery Construction Materials* for further discussion of fuels and reformer operation.

BURNERS

Each row of catalyst filled tubes is fired on two sides with floor mounted, John Zink PM-12-4H, burners [John Zink PM 14 - 5H burners are provided for low Btu (low joule) gas firing]. These natural draft burners have been specially developed to have a long, narrow, stable flame. This results in a uniform heat density over the length of the tube and permits the ExxonMobil reformer to use only floor firing in most cases. A note should be included in the Design Specification requiring final approval of the burner sizing by the Owner's Engineer. To date, applications of low NO_x burners technology in this area have been limited. If low NO_x burners are required for steam reformers, the designer should contact the Heat Transfer Equipment Section of EMRE Technology Department.

Four rows of burners should be provided, two side rows (one between each tube row and the adjacent side wall) and two center rows (between the two tube rows). Two center rows of burners permits independent balancing of the firing rate to each tube row. Each row should contain the same number of burners and have a separate fuel manifold.

The normal burner heat release is determined by the required heat fired divided by the total number of burners. The total number of burners, in turn, is determined by the length of the radiant box. For good heat distribution the burner center to center spacing should be about 3 1/2 ft (1 m). (See **Section VIII-F** for burner dimensions.) This should result in a normal heat release in the range of 2 1/4 to 4 MBtu/hr (0.67 to 1.2 MW) per burner. The maximum burner heat release should be 1.10 times the normal heat release rate.

One row of wall burners should be provided only for LPG or naphtha feeds on each side of the radiant box near the arch. They are required in ExxonMobil reformers with LPG or naphtha feeds to better match the reforming heat demand at the tube inlet. They have also been used for additional flexibility during startup and in balancing the reformer load at maximum throughput in units with all types of feed. Except for unit with naphtha and LPG feeds, the wall burners have typically not been used on a continuous basis and have been removed at locations processing lighter feeds to improve the heater thermal efficiency.

The wall burners should be located as shown in **Figure 2**. Each burner provides about 15% of the total heat fired, and is sized for a heat release of 0.85 - 1.0 MBtu/hr (0.25 - 0.30 MW) each.

SHIELD AND CONVECTION SECTION DESIGN AND LAYOUT

The shield and convection section design for a steam reformer follows the procedures outlined in **Section VIII-C**, with the following exceptions and additions.

Shield Duty - If the shield is exposed directly to the radiant section, the duty and tube metal temperature should be calculated according to the optional method given in the **Appendix** of **Section VIII-C**. In this case, the shield radiant duty is included as part of the total radiant duty as given in **Eq. (13a)**.

If the convection section is offset (as will be the case for down fired heaters), the shield radiant duty is not included as part of the total radiant duty. In addition, neither of the methods given in **Section VIII-C** for calculating the shield radiant duty is applicable. For this case, contact the Heat Transfer Equipment Section of EMRE Technology Department.

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HEATER DESIGN (Cont)

Excess Air Rate - While the radiant section heat transfer surface and tube metal temperatures and convection section heat transfer surface are calculated at 15% excess air, convection section materials and design tube metal temperatures are determined for 30% excess air. This requires a new bridgewall temperature and fuel gas rate for the 30% excess air operation. The increase in bridgewall temperature over design at 15% excess air can be estimated by using the difference in T'_{bw} from **Figures 14** (for 15% excess air) and **15** (for 30% excess air) for the same radiant heat density. The bridgewall temperature for 30% excess air is usually 15 to 20°F higher than for the 15% excess air case.

The convection section can be readily designed using Computer Program 3558.

Tube Surface - The convection section should be designed making maximum use of extended surface tubes. However, a minimum of one bare tube row should be used above the two shield tube rows before fin tubes are considered, regardless of the calculated flue gas temperatures. This is because fins on the third row could be exposed to a small amount of radiation from the radiant section.

Design of independent convection section coils, particularly steam superheaters, should take into account variations in heat transfer that can occur over the ranges of heater operating conditions. These variations can cause higher convection section coil outlet and metal temperatures than normally anticipated.

Steam Drum - The steam drum should be completely specified according to the procedures presented in **Section V-A**. These procedures were developed because of a high incidence of water carryover with contractor designed drums. High quality steam and sufficient holdup are essential for reliable reformer operation.

REFORMER HEAT BALANCE

DESIGN EXCESS AIR RATE

As discussed above, 15% excess air rate should be used for the reformer heat balance.

BRIDGEWALL TEMPERATURE T_{bw}

The bridgewall temperature is the temperature of the flue gas as it leaves the radiant section. For convection sections exposed directly to the firebox, the bridgewall temperature is the temperature of the flue gas after having transferred radiant heat to the shield tubes. In heaters with an offset convection section, it is before radiant heat transfer to the shield.

For an ExxonMobil reforming heater,

$$T_{bw} = T_G - 200^{\circ}\text{F} + \text{HRF} \quad (\text{Customary}) \quad \text{Eq. (11)}$$

$$T_{bw} = T_G - 110^{\circ}\text{C} + \text{HRF} \quad (\text{Metric}) \quad \text{Eq. (11)M}$$

where:

$$T_G = \left\{ \frac{0.55}{F} \left[(T'_G + 460)^4 - (T_T + 460)^4 \right] + (T_T + 460)^4 \right\}^{1/4} - 460 \quad (\text{Customary}) \quad \text{Eq. (11a)}$$

$$T_G = \left\{ \frac{0.55}{F} \left[(T'_G + 273)^4 - (T_T + 273)^4 \right] + (T_T + 273)^4 \right\}^{1/4} - 273 \quad (\text{Metric}) \quad \text{Eq. (11a)M}$$



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REFORMER HEAT BALANCE (Cont)

This follows from **Eq. (11b)** and corrects the radiating gas temperature for radiant exchange factors other than 0.55.

$$F \left[(T_G + 460)^4 - (T_T + 460)^4 \right] = 0.55 \left[(T'_G + 460)^4 - (T_T + 460)^4 \right] \quad (\text{Customary}) \quad \text{Eq. (11b)}$$

$$F \left[(T_G + 273)^4 - (T_T + 273)^4 \right] = 0.55 \left[(T'_G + 273)^4 - (T_T + 273)^4 \right] \quad (\text{Metric}) \quad \text{Eq. (11b)M}$$

- where:**
- T_{bw} = Bridgewall temperature, °F (°C)
 - T_G = Effective radiating gas temperature, °F (°C)
 - T'_{bw} = Bridgewall temperature factor, °F (°C). Determine from **Figure 14**, using T_T and ϕ_{ff} .
 - T'_G = Effective gas temperature factor, °F (°C)
 - = $T'_{bw} + 200^\circ\text{F}$ $T'_{bw} + 110^\circ\text{C}$
 - T_T = Effective cold plane temperature, °F (°C)
 - = $T_{La} + 25^\circ\text{F}$ $T_{La} + 14^\circ\text{C}$
 - T_{La} = Length average tube metal temperature on the front face of the tubes, °F (°C)
(Program 3676 calculates this value)
 - ϕ_{ff} = Radiant section average front face heat density, Btu/hr ft² kW/m² (°C), from Program 3676
 - HRF = Heat release factor, °F (°C)
 - = Customary (40-h)⁸; Metric (12.2-h)^{4.4} for Exxon reformer heaters. For other heater designs, contact the Heat Transfer Equipment Section of EMRE Technology Department.
 - h = Distance from the floor to the center line of the bottom row of shield tubes, ft (m) [usually 37 ft (11.3 m) for 35 ft (10.7 m) tubes]

RADIANT SECTION HEAT DUTY Q_r

The radiant section heat duty (reforming process duty) may be calculated from enthalpy information when the inlet and outlet compositions and conditions have been defined. This duty is also calculated by Computer Program 3676.

SHIELD RADIANT HEAT DUTY Q_{sr}

When the shield tubes are directly exposed to the radiant section, they absorb a portion of the total radiant duty.

$$Q_{sr} = \phi_s L_c W_c \quad (\text{Customary}) \quad \text{Eq. (12)}$$

$$Q_{sr} = \frac{\phi_s L_c W_c}{1000} \quad (\text{Metric}) \quad \text{Eq. (12)M}$$

- where:**
- Q_{sr} = Shield section radiant duty, Btu/hr (MW)
 - ϕ_s = Shield section front face radiant heat density, Btu/hr ft² (kW/m²) (use the optional procedure in the **Appendix** of **Section VIII-C** for calculation of ϕ_s)
 - L_c = Convection section length, ft (m)
 - W_c = Convection section width, ft (m)

If the convection section is offset, contact the Heat Transfer Equipment Section of EETD for an appropriate method of calculating the radiant shield duty.

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REFORMER HEAT BALANCE (Cont)

TOTAL RADIANT HEAT DUTY Q_{tr}

If the shield is directly exposed to the radiant section,

$$Q_{tr} = Q_r + Q_{sr} \quad \text{Eq. (13a)}$$

If the convection section is offset,

$$Q_{tr} = Q_r \quad \text{Eq. (13b)}$$

where: Q_{tr} = Total radiant section heat duty (including shield section radiant duty), Btu/hr (MW)
 Q_r = Heat absorbed by the radiant section surface, Btu/hr (MW)

NET FUEL REQUIREMENT F_n

$$F_n = \frac{Q_{tr}}{(HA)_{bw}} \quad \text{(Customary)} \quad \text{Eq. (14)}$$

$$F_n = \frac{Q_{tr} \times 1000}{(HA)_{bw}} \quad \text{(Metric)} \quad \text{Eq. (14)M}$$

where: F_n = Net fuel required, lb/hr (kg/s)
 $(HA)_{bw}$ = Heat available at bridgewall temperature, Btu/lb (kJ/kg) (see [Figures 7-10](#) of [Section VIII-M](#))

Net fuel is used for all flue gas enthalpy calculations in the heater.

GROSS FUEL REQUIREMENT F_g

$$F_g = 1.02 F_n \text{ for heaters larger than 100 MBtu/hr (30 MW) heat absorbed} \quad \text{Eq. (15a)}$$

$$= 1.03 F_n \text{ for heaters between 15 and 100 MBtu/hr (4 and 30 MW)} \quad \text{Eq. (15b)}$$

$$= 1.04 F_n \text{ for heaters smaller than 15 MBtu/hr (4 MW)} \quad \text{Eq. (15c)}$$

where: F_g = Gross fuel requirement, lb/hr (kg/s)

Gross fuel is used to determine total heat fired, efficiency, combustion air requirements, stack gas velocity, and flue gas mass velocity and pressure drop. The difference between net fuel and gross fuel accounts for radiation and other heat losses from the heater (excluding stack losses).

HEAT FIRED Q_f

$$Q_f = F_g (\text{LHV}) \quad \text{(Customary)} \quad \text{Eq. (16)}$$

$$Q_f = F_g (\text{LHV}) \times 10^{-6} \quad \text{(Metric)} \quad \text{Eq. (16)M}$$

where: Q_f = Heat fired, Btu/hr (MW)
LHV = Fuel lower heating value, Btu/lb (kJ/kg) ([Figures 7-10](#) of [Section VIII-M](#))

By convention, lower heating value (LHV) is always used in fired heater and burner calculations.



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REFORMER HEAT BALANCE (Cont)

STACK TEMPERATURE T_s

The economical stack temperature depends on a balance of fuel costs and investment costs, including convection section tube surface, return bends, refractory walls and stack. Procedures for determining economical stack temperature are given in [Section VIII-A](#).

TOTAL HEAT DUTY Q_a

$$Q_a = F_n (HA)_s \quad (\text{Customary}) \quad \text{Eq. (17)}$$

$$Q_a = F_n (HA)_s \times 10^{-6} \quad (\text{Metric}) \quad \text{Eq. (17)M}$$

where: Q_a = Total heat absorbed in the heater, Btu/hr (MW)
 F_n = Net fuel required, lb/hr (kg/s) [from [Eq. \(14\)](#)]
 $(HA)_s$ = Heat available at stack temperature, Btu/lb (kJ/kg)

HEATER EFFICIENCY E_{LHV}

$$E_{LHV} = \frac{100 Q_a}{Q_f} \quad \text{Eq. (18)}$$

where: E_{LHV} = Thermal efficiency of heater based on lower heating value, percent

TOTAL CONVECTION SECTION DUTY Q_{tc}

The convection section of a reformer heater is usually used for process preheat and/or steam generation and superheat.

$$Q_{tc} = Q_a - Q_r \quad \text{Eq. (19)}$$

where: Q_{tc} = Total convection section duty, Btu/hr (MW)

Note that Q_{tr} and Q_{tc} overlap, since both include the shield radiant duty, Q_{sr} .

MECHANICAL SPECIFICATION

In addition to the items normally covered in the Design Specification (see [Section VIII-A, Table 1](#)), the mechanical details specific to the ExxonMobil reformer design must be included. These details are shown in [Figures 16-20](#). The Heat Transfer Equipment Section of EMRE Technology Department should be contacted for a complete, up-to-date specification of these details.

CATALYST TUBE INLET PIPING

The feed inlet headers, which run along the arch on each side of the heater should be sized to limit flow maldistribution among the tubes to 2%. This is accomplished by limiting the velocity head in the header to a maximum of 5% of the catalyst tube pressure drop. Also, the feed should enter the inlet header on the same end of the box as the feed discharges through the outlet header and the first pigtail takeoff should be a minimum of five diameters downstream of any bends. Adequate run length should be provided to ensure the steam and hydrocarbon are well mixed.

Feed is transferred from the inlet headers to the catalyst tubes by means of flexible pigtails. They are usually fabricated from 1 in. (25 mm) NPS pipe. Materials range from 1 1/4 Cr - 1/2 Mo to Type 304 stainless steel depending on the temperature, pressure, flexibility stresses, and requirements for hydrogen attack. Two and 1/4 Cr - 1 Mo is the most commonly used material for the manufacturer of inlet pigtails. If calculations suggest the use of stainless steel, the Materials Engineering Section of EMRE shall be contacted before the final specification as severe problems with stress corrosion cracking have been observed in recent years. The pigtails must be designed to absorb the movement from:

- Design thermal expansion of the radiant tubes.
- Thermal expansion of the inlet header from the anchored midpoint at design temperature.

The stresses must be limited to the ANSI B31.3, Refinery Piping Code, allowable stress. Stress intensification factors should be included to account for the socket weld connection of the pigtail to the radiant tube.