

Fired process heaters

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1. Introduction

Furnaces are a versatile class of equipment where heat is liberated and transferred directly or indirectly to a solid or fluid mass for the purpose of effecting a physical or chemical change. In industrial practice many and varied types of furnaces are used which may differ in function, overall shape or mode of firing, and furnaces may be classified accordingly on the basis of their function such as smelting or roasting, their shape such as crucibles, shafts and hearths or they may be classified according to their mode of firing into electrical, nuclear, solar, and combustion furnaces. Combustion furnaces are of two general types: fired heaters and converters. A converter is a type of furnace in which heat is liberated by the oxidation of impurities or other parts of the material to be heated. Fired heaters, on the other hand, are furnaces that produce heat as a result of the combustion of fuel. The heat liberated is transferred to the material to be heated directly (in internally-heated furnaces) or indirectly (in externally-heated furnaces). Examples of internally-heated furnaces include submerged heaters and blast furnaces where a solid mass is heated by a blast of hot gases. Externally-heated furnaces include ovens, fire-tube boilers and tubular heaters.

In tubular or pipe-still heaters process fluids flowing inside tubes mounted inside the furnace are heated by gases produced by the combustion of a liquid or gaseous fuel. Such heaters were first introduced in the oil industry by M. J. Trumble. Two advantages were thereby gained, viz. the achievement of continuous operation and the reduction of foam formation. These heaters are widely used for heating purposes in petroleum refining, petrochemical plants and other chemical process industries. The calculation and design of fired heaters in petroleum refineries in particular remains one of the most important applications of heat transfer.

Tubular fired heaters are generally built with two distinct heating sections: a radiant section, variously called a combustion chamber or firebox, and a convection section followed by the stack. The hot flue gases arising in the radiation section flow next into the convection section where they circulate at high speed through a tube bundle before leaving the furnace through the stack. A third section, known as a shield or shock section, separates the two major heating sections. It contains those tubes close to the radiation section that shield the remaining convection section tubes from direct radiation. The shield section normally consists of two to three rows of bare tubes that are directly exposed to the hot gasses and

flame in the radiant section, but the arrangement varies widely for the many different heater designs. In certain fired heater designs, known as the all-radiant type, there is no separate convection section.

The functions served by fired heaters in chemical plants are many ranging from simple heating or providing sensible heat and raising the temperature of the charge to heating and partial evaporation of the charge, where equilibrium is established between the unvaporised liquid and the vapour. The charge leaves the furnace in the form of a partially evaporated liquid in equilibrium. Fired heaters may also be used to provide the heat required for cracking or reforming reactions. In this case the furnace would be divided into two parts: a heater, where the temperature of the charge is raised, and a soaker, where heat is provided in order to maintain a constant temperature. The soaker may be a part of either the radiation or convection sections. Radiation soakers are generally to be preferred to convection soakers for better control of heat input.

Fired heaters are usually classified as vertical cylindrical or box-type heaters depending on the geometrical configuration of the radiant section.

In box-type heaters, the radiant section has generally a square cross section (as in older furnaces, where the reduced height is compensated for by a larger construction site) or a rectangular cross section (with its height equal to 1.5 to 2.5 times its width) as in cabin-type heaters. The configuration of the cabin type heater with horizontal tubes may have a so-called hip, or the radiant section may be just a rectangular box.

The tubes in the radiant section may be arranged horizontally or vertically along the heater walls including the hip and the burners are located on the floor or on the lower part of the longest side wall where there are no tubes. A fire wall is often built down the centre of the combustion chamber in heaters fired from the sides.

Box-type furnaces are best suited for large capacities and large heat duties. This makes such furnaces particularly suitable for topping units where it is possible to increase tube length in both the radiation and convection sections reducing thereby the required number of headers or return bends. It is possible in such furnaces to open the headers for cleaning, something which makes cleaning the tubes in the radiation section easier; this is especially significant in furnaces used to heat easily-coking oils.

In the cylindrical-type furnace, the radiation section is in the shape of a cylinder with a vertical axis, and the burners are located on the floor at the base of the cylinder. The heat exchange area covers the vertical walls and therefore exhibits circular symmetry with respect to the heating assembly. In the radiant section, the tubes may be in a circular pattern around the walls of the fire box or they may be in a cross or octagonal design which will expose them to firing from both sides. Older designs have radiating cones in the upper part of the radiant section as well as longitudinal fins on the upper parts of tubes. The shield and convection tubes are normally horizontal.

Cylindrical heaters with vertical tubes (Fig. 2) are commonly used in hot oil services and other processes where the duties are usually small, but larger units, 100 million kJ/hr and higher, are not uncommon.

Cylindrical heaters are often preferred to box-type heaters. This is mainly due to the more uniform heating rate in cylindrical heaters and higher thermal efficiency. Furthermore, cylindrical heaters require smaller foundations and construction areas and their construction cost is less. High chimneys are not essential in cylindrical furnaces because they normally produce sufficient draught.

2. Analysis and evaluation of heat transfer

In the usual practice, the process fluid is first heated in the convection section preheat coil which is followed by further heating in the radiant section. In both sections heat is transferred by both mechanisms of heat transfer, viz. radiation and convection, where radiation is the dominant mode of heat transfer at the high temperatures prevalent in the radiant section and convection predominates in the convection section where the average temperature is much lower. Most of the heat transferred to the process fluid takes place in the radiant section while the convection section serves only to make up the difference between the heat duty of the furnace and the part absorbed in the radiant section. Roughly-speaking, about 45-55% of the total heat release in the furnace is transferred to the process fluid in the radiant section, leaving about 25-45% of the total heat release to be either transferred to the process fluid in the convection section or carried by the flue gases through the stack and is lost.

Generally speaking, the chief mechanism by which heat is transferred to the process fluid is radiative heat transfer. In the radiant section the process fluids are mainly heated by direct radiation from the flame. About 70-90 % of the heat absorbed by the tubes in the radiant and shield sections of a furnace is transferred by radiation. This clearly indicates that heat transfer by radiation is of particular significance for the thermal analysis and design of fired heaters in general. Furthermore, the fundamental understanding of heat transfer by radiation is essential for a better appraisal of the expected effects of any modifications of existing heaters or suggestions for possible future developments. Heat transfer by radiation can not however be considered in isolation, but it must be considered in conjunction with other mechanisms of heat transfer such as conduction and convection, in addition to heat liberation by such chemical reactions as may take place inside heater tubes.

The heat-absorbing surface in both sections of the heater is the outside wall of the tubes mounted inside the heater. The overall thermal efficiency of the fired heater is dependent to a large extent on the effectiveness of the recovery of heat from the flue gases, which depends in turn on the size of the heat exchange surface area in the furnace. In order to increase the heat transfer area and thus further increase the overall efficiency of the heater, finned or studded tubes are normally used in the convection section. By this means it is often possible to attain heat flux in the convection section comparable to that in the radiation section.

Different methods are available for the calculations involved in the thermal evaluation and design of fired heaters and some of these methods may be based in theory on fundamental

considerations, but more usually the calculation of heat transfer in heaters is based on empirical methods and the construction of such units precedes in general theoretical developments. The validity and significance of a method is determined to a large extent by its general application for different furnace designs. Such a method would take into consideration a large number of variables that affect the process of heat transfer.

While some of the empirical methods available are simplified methods that can be used for an overall evaluation of the heater in a relatively simple and rapid manner, other methods are more rigorous where it is possible to obtain a better approximation, but the application of such methods requires more information and longer time, and would in fact require the use of a computer. Using computer programmes greatly simplifies the procedure for the computations involved in fired heaters' evaluation and analysis and makes it possible to use more rigorous methods giving better and more accurate results. Most computer programmes available in the literature are written in Basic, but there are also some programmes written in other more advanced computer languages such as Matlab, a programming language which is widely used in all scientific fields and in engineering sciences in particular.

The use of computer simulation and modelling for the optimization of fired heaters operation and design can be a powerful tool indispensable to combustion researchers, furnace designers and manufacturers in their efforts to develop and design advanced heaters with higher performance and efficiency and lower pollutant emissions. Modelling in most cases, however, involves the use of numerical methods and simplifying assumptions which may lead if not approached with care to serious errors, given in particular the complexity of the problems involved.

In most methods available in the literature for the modelling and design of fired heaters, the heater is divided into surface and volume (gas) zones of different heights where each gas zone is treated as well-stirred and completely mixed and temperatures may be calculated at the inlet and outlet to each zone. For a rough analysis of heat transfer by radiation, the surface zones may be assumed to be black bodies and the volume zones grey gases, but for more accurate work the surface zones may have to be considered as grey and the volume zones as real gases.

In the stirred furnace model, which is an early design method, three zones are assumed: two surface zones (the sink and the refractory) and a gas zone comprising the combustion products. This is a simple and approximate method which may be further refined for greater accuracy by increasing the number of zones, particularly where significant changes in gas temperature and composition and in the surface temperature and emissivity are expected. Dividing the furnace into a series of well stirred zones approximates to the long furnace model where plug flow and radial radiation are assumed, with no axial radiation and negligible back mixing. In the Monte Carlo model, on the other hand, the zone model may also be refined by the appropriate choice of the shape of the zones so that they fit the heater geometry. In this method radiative exchange between zones is calculated by tracing "parcels" of radiation moving along random paths.

The basic objection to the zone model in the considered opinion of many designers is that it assumes discontinuous changes from one homogenous gas zone to the next. This makes this model not a very realistic one. The flux model which allows for variation in gas properties as a smooth function throughout space would be a more realistic model than the zone model. The flux method should be used when accuracy demands that continuous changes in gas temperature or optical properties cannot be represented by discrete homogeneous gas zones. In the flux method radiative transfer through gases is considered as beams of photons which are absorbed and scattered according to the laws of astrophysics.

Modelling of fired heaters is generally based on the two heater sections, namely the radiation and the convection sections. Within each section three heat transfer elements need to be considered. These are the flue gas, the process fluid and the heat exchange surfaces including the tubes and the refractory walls of the heater. In addition to the calculation of heat exchange areas and heat transfer rates to the process fluid, other key variables serve as a basis for the determination of heater performance. These include:

- (1) Process fluid and flue gas temperatures.
- (2) Flame and tube skin or tube wall temperatures.
- (3) Process fluid flow rate.
- (4) Fuel flow rate and composition.
- (5) Process fluid pressure drop and the pressure profile in the heater and stack.

For most furnace designers and operators, however, the thermal efficiency remains the most important single performance factor. This is particularly so in view of the current trend of rising fuel costs and environmental concerns. The successful design and operation of a fired heater must aim first and foremost at the highest possible thermal efficiency with due regards to other performance and pollution considerations.

3. Heat transfer in the radiant section

Heat transfer by radiation is governed in theory by the Stefan-Boltzman law for black body radiation, $Q_R = \sigma T^4$. In practice, the mathematical solution of radiative heat transfer is more complicated however as it involves the calculation of heat exchange factors as a function of the furnace geometry and the calculation of the absorptivity and emissivity of the combustion gases. There are also other factors to be considered such as the emissivities of the surface and the effects of re-radiation of the tubes.

There are two primary sources of heat input to the radiant section, the combustion heat of fuel, Q_{rls} , and the sensible heat of the combustion air, Q_{air} , fuel, Q_{fuel} and the fuel atomization fluid (for liquid fuel when applicable), Q_{fluid} . Part of this heat input liberated in the radiant section is absorbed by the tubes in the radiant Q_R and shield Q_{shld} sections, while the remaining heat is either carried as sensible heat by the exiting flue gas, $Q_{flue\ gases}$ or is lost by radiation through the furnace walls or casing, Q_{losses} . The temperature of the flue gas can then be calculated by setting up a heat balance equation.

Radiation heat losses through furnace walls depend on the size of the furnace, where greater heat losses are to be expected in small furnaces as the ratio between the walls area and the

volume of the radiation section decreases with furnace size increase. For furnaces of 10 MW or more, they range from 2 to 5% of the net calorific value. Apart from furnace size, however, radiation heat losses depend also on the material composing the insulating refractory lining and its thickness, and these losses may be low for an economically optimum insulating lining.

For the radiant section, the heat balance equations are:

$$Q_{in} = Q_{rls} + Q_{air} + Q_{fuel} + Q_{fluid} \quad (1)$$

$$Q_{rls} = m_{fuel} \times NCV \quad (2)$$

$$Q_{air} = m_{air} \times C_{p,air} \times (T_{air} - T_{datum}) \quad (3)$$

$$Q_{fuel} = m_{fuel} \times C_{p,fuel} \times (T_{fuel} - T_{datum}) \quad (4)$$

$$Q_{fluid} = C_{p,fluid} \times (T_{fluid} - T_{datum}) \quad (5)$$

$$Q_{out} = Q_R + Q_{shld} + Q_{losses} + Q_{flue\ gases} \quad (6)$$

$$Q_R = Q_r + Q_{conv.} \quad (7)$$

$$Q_r = \sigma \times (\alpha \times A_{cp}) \times F \times (T_g^4 - T_w^4) = \text{radiant heat transfer} \quad (8)$$

$$Q_{conv} = h_{conv} \times A_t \times (T_g - T_w) \quad (9)$$

= convective heat transfer in the radiant section

$$Q_{shld} = \sigma \times (\alpha \times A_{cp})_{shld} \times F \times (T_g^4 - T_w^4) \quad (10)$$

$$Q_{losses} = (2-5)\% \times m_{fuel} \times NCV = \text{Radiation heat losses} \quad (11)$$

$$Q_{flue\ gases} = m_{flue\ gases} \times C_{p,flue\ gases} \times (T_g - T_{datum}) \quad (12)$$

The heat balance equation is:

$$Q_{rls} + Q_{air} + Q_{fuel} + Q_{fluid} = Q_R + Q_{shld} + Q_{losses} + Q_{flue\ gases} \quad (13)$$

4. Calculation of Flame and Effective Gas Temperatures.

Flame and effective gas temperatures are key variables that need to be accurately determined before analysis of the heat transfer in the radiant section of fired heaters can be meaningfully undertaken.

Flame temperature is the temperature attained by the combustion of a fuel. This temperature depends essentially on the calorific value of the fuel. A theoretical or ideal flame temperature may be calculated assuming complete combustion of the fuel and perfect mixing. But even when complete combustion is assumed, the actual flame temperature would always be lower than the theoretical temperature. There are several reasons for this, chiefly the dissociation reactions of the combustion products at higher temperatures (greater than about 1370°C), which are highly endothermic and absorb an enormous amount of heat, and heat losses by radiation and convection to the walls of the combustion chamber.

Some work has been done on the calculation of flame temperature, including work by Stehlik and others who studied furnace combustion and drew furnace temperature and

enthalpy profiles. Vancini wrote a programme in assembly language for the calculation of the average flame temperature, taking into account dissociation at higher temperatures.

A simple heat balance normally serves as the basis for calculating the flame temperature. The increase in enthalpy between the unburned and burned mixtures is assumed to be equal to the heat produced by the combustion. When the fuel is fired, the heat liberated raises the temperature of the combustion products from t_1 to t_2 so that the following relationship is satisfied:

$$Q_{combustion} = \sum_{i=1}^5 W_i \times \int_{t_1}^{t_2} C_{p_i} \cdot dt \quad (14)$$

Where:

$Q_{combustion}$ = Heat of combustion of fuel based on the gross calorific value.

W_i = Mass of a flue gas component. The five components considered are CO_2 , N_2 , O_2 , SO_2 and water vapour.

C_{p_i} = Molar heat of a flue gas component.

t_1 and t_2 = Initial and final temperatures.

The use of Equation (14) allows the calculation of the flame temperature by iteration using a programmable calculator. The variation of C_{p_i} with temperature can be approximated by a polynomial, having the obvious advantage of being integrated easily. Using a third-degree polynomial, C_{p_i} can be written as:

$$C_{p_i} = a_i + b_i \times t + c_i \times t^2 + d_i \times t^3 \quad (15)$$

Where, a_i , b_i , c_i and d_i are constants dependent on the nature of the gas. Assuming t_1 to be negligible ($= 0$), Equation (14) thus becomes:

$$Q_{combustion} = \sum_{i=1}^5 W_i \times \int_0^t (a_i + b_i \times t + c_i \times t^2 + d_i \times t^3) \cdot dt \quad (16)$$

Integrating:

$$Q_{combustion} = \sum_{i=1}^5 W_i \times \left(a_i + \frac{b_i \times t}{2} + \frac{c_i \times t^2}{3} + \frac{d_i \times t^3}{4} \right) \times t \quad (17)$$

It is customary to call the parenthetic term in Equation (17) the mean molar heat:

$$C_{p_{m,i}} = a_i + \frac{b_i \times t}{2} + \frac{c_i \times t^2}{3} + \frac{d_i \times t^3}{4} \quad (18)$$

By taking mean molar heats instead of true molar heats, the integration of Equation (14) may be dispensed with. The molar heats at constant pressure for air and flue gases are given in Table (1).

For heat transfer at constant pressure:

$$Q_{\text{combustion}} = \sum_{i=1}^5 W_i \times C_{p_{m,i}} \times (t_2 - t_1) \quad (19)$$

Equation (19) allows the calculation of the theoretical flame temperature, t_2 , by iteration via a programmable calculator. In order to compensate for the factors that tend to lower the theoretical flame temperature, the heat of combustion is usually multiplied by an empirical coefficient. The values normally used for this coefficient are only estimates; this is why the temperature calculated with any method can only approximate actual values. For an accurate calculation of the actual flame temperature, account must be taken of heat losses through the casing by setting up heat balance equation for fuel gas as follows:

$$Q_{\text{combustion}} - Q_{\text{losses}} = \sum_{i=1}^5 W_i \times C_{p_{m,i}} \times (t_2 - t_1) \quad (20)$$

Where:

$$Q_{\text{combustion}} = M_{\text{fuel}} \times \text{GCV} \quad (21)$$

$$Q_{\text{losses}} = 5\% \times Q_{\text{combustion}} \quad (22)$$

The Newton-Raphson method is used to solve the heat balance equation and determine the actual flame temperature, for which two Matlab programmes are written (Appendices 1 and 2).

Gas	Molar heat (kJ/ kmol.K)	Temp. Range
Air	$33.915 + 1.214 \times 10^{-3} \times T$	50-1500 K
CO ₂	$43.2936 + 0.01147 \times T - 818558.5/T^2$	273-1200 K
N ₂	$27.2155 + 4.187 \times 10^{-3} \times T$	300-3000 K
O ₂	$34.63 + 1.0802 \times 10^{-3} \times T - 785900/T^2$	300-5000 K
SO ₂	$32.24 + 0.0222 \times T - 3.475 \times 10^{-6} \times T^2$	300-2500 K
*H ₂ O _(g)	$34.42 + 6.281 \times 10^{-4} \times T + 5.611 \times 10^{-6} \times T^2$	300-2500 K

*H₂O_(g) is gas phase

Table 1. Molar heats at constant pressure for air and flue gases

The effective gas temperature is the temperature controlling radiant transfer in the heater radiant section. For most applications complete flue gas mixing in the radiant section is normally assumed and the effective gas temperature is taken to be equal to the bridgewall temperature, i.e. the exit temperature of the flue gases leaving the radiant section. This is normally assumed in most methods used for the estimation of the effective gas and other radiant section temperatures, including the widely-used Lobo-Evans method. In general, this is an acceptable assumption with the notable exception of high temperature heaters with tall narrow fireboxes and wall firing where longitudinal and transverse temperature gradients may not be ignored and the effective gas temperature may be 95 to 150°C higher than the bridgewall temperature. In this and other cases where the two temperatures differ widely and an adjustment may be necessary, the use of a more accurate gas temperature

may have to be considered. The usual approach is to divide the radiant section into zones for the energy balance calculations, or temperature gradients in the fired heater may be evaluated in order to predict the bridgewall and effective gas temperatures.

The average tube wall temperature is given by:

$$T_w = 100 + 0.5 \left(\frac{T_{in} + T_{out}}{2} \right) \quad (23)$$

The Newton-Raphson method is then used to solve the heat balance equation (Eq. 13) and determine the effective gas temperature, for which two Matlab programmes are written (Appendices 2 and 3).

To illustrate the use of the above programmes, an example is worked out for an actual crude oil heater used in an atmospheric topping unit at the Homs Oil Refinery (Cabin 43-5-16/21 N). In this example, fuel gas is fired with 25% excess air. Ambient temperature = 15°C, exit gas temperature = 400°C. Table (2) shows the geometrical characteristics for the heater, and Table (3) shows its Process data sheet and the characteristics of the fuel (natural gas), flue gas, process fluid and air.

External Dimensions of heater (m)	20.000×4.800×19.650
Total Number of tubes	100
Weight of heater (kg)	307000
Weight of refractory (kg)	298000
Geometrical Characteristics of Radiant Section	
Number of passes	2
Number of tubes	60
Overall tube length (m)	20.824
Effective tube length (m)	20.024
Tube spacing, centre-to-centre (mm)	394
Tube spacing, centre-to-furnace wall (mm)	220
Outside diameter of tube (mm)	219
Wall thickness of tube (mm)	8
Tube materials	Stainless steel 18 Cr-8 Ni, Type AISI 304
Geometrical Characteristics of Convection Section	
Total number of tubes	40
Number of passes	2
Number of shield tubes	8
Overall tube length (m)	20.824
Effective tube length (m)	20.024
Tube spacing, centre-to-centre (mm)	250
Outside diameter of tube (mm)	168
Wall thickness of tube (mm)	8
Tube materials	Stainless steel 18 Cr-8 Ni, Type AISI 304

Table 2. Geometrical Characteristics of Box-Type Fired Heater.

Design thermal load (Duty) (kJ/h)	8.482×10 ⁷
Unit process conditions	
Process fluid	Heavy Syrian Crude oil
Fluid flow rate (kg/h)	225700
Specific gravity at 15 °C	0.9148
UOP K	12.1
Molecular weight	105.183
Specific heat (kJ/kg. K)	2.5744
Inlet conditions	
Temperature (°C)	210
Pressure (bar)	Max 19.5
Liquid density kg/m ³	914.8
Liquid viscosity (cst at 70°C)	30
Percentage of weight of vapour	0.0
Outlet conditions	
Temperature (°C)	250 from convection 355 from radiant section
Pressure (Mpa)	0.91
Percentage of weight of vapour	0.0
Design conditions	
Minimum calculated efficiency %	75.78
Radiation losses %	5.0
Flue gas velocity though convection	2.677 (kg/m ² .s)
Fuel characteristics	
Type of fuel	Natural gas
Nett calorific value (kJ/kmol)	927844.41
Molar heat (kJ/kmol.K)	39.26
Temperature (°C)	25
Flow of fuel (kmol/h)	120
Molecular weight (kg/kmol)	19.99
Composition (% mol)	CH ₄ (80.43), C ₂ H ₆ (9.02), C ₃ H ₈ (4.54), iso-C ₄ H ₁₀ (0.20), n-C ₄ H ₁₀ (0.32), iso-C ₅ H ₁₂ (0.04), n-C ₅ H ₁₂ (0.02), CO ₂ (3.52), H ₂ S (0.09), N ₂ (1.735).
Air characteristics	
Molar heat (kJ/kmol.K)	33.915+1.214×10 ⁻³ ×T
Flow of air (kmol/h)	1589.014
Air temperature (°C)	25
Percentage of excess air	25%
Flue gas characteristics	
Molar heat (kJ/kmol.K)	29.98+3.157×10 ⁻³ ×T
Specific heat(kJ/kg.K)	1.0775+1.1347×10 ⁻⁴ ×T
Flow of flue gas (kmol/h)	1720.9
Molecular weight (kg/kmol)	27.82336
Composition (% mol)	CO ₂ (8.234), H ₂ O (15.968), O ₂ (3.82), N ₂ (71.79), SO ₂ (0.188)

Table 3. Process data sheet for box-type fired heater.

The flame temperature equation, derived using programme (1), has the following form:

$$F(t_f) = a \times t_f + b \times t_f^2 + c \times t_f^{-1} + d \times t_f^3 + e \quad (24)$$

The first derivative of the flame temperature equation is:

$$\frac{dF(t_f)}{dt_f} = a + 2 \times b \times t_f - c \times t_f^{-2} + 3 \times d \times t_f^2 \quad (25)$$

Where a, b, c, d and e are constants estimated dependent on the type of fuel and its gross calorific value, the percentage of excess air and the operating conditions of the fired heater. These constants can then be estimated using Programme 2 as follows (Table 4):

$$F(t_f) = 29.9825 \times t_f + 0.0021 \times t_f^2 + 9.7421 \times 10^4 t_f^{-1} + 2.9648 \times 10^{-7} \times t_f^3 - 6.4657 \times 10^4 \quad (26)$$

$$\frac{dF(t_f)}{dt_f} = 29.9825 + 0.0042 \times t_f - 9.7421 \times 10^4 \times t_f^{-2} + 8.8943 \times 10^{-7} \times t_f^2 \quad (27)$$

These equations were solved by the Newton-Raphson method using programme 2 to give the actual flame temperature of 2128 K.

The effective gas temperature equation, derived using programme (3), has the following form:

$$F(Tg) = C \times Tg^4 + D \times Tg - B \quad (28)$$

The first derivative of the effective gas temperature equation is:

$$\frac{dF(Tg)}{dTg} = 4 \times C \times Tg^3 + D \quad (29)$$

where B, C and D are constants dependent on the type of fuel, percentage of excess air, operating conditions and geometrical characteristics of the fired heater. These constants can then be estimated using Programme 2 as follows (Table 5):

$$F(Tg) = 9.0748 \times 10^{-5} \times Tg^4 + 7.9153 \times 10^4 \times Tg - 1.5533 \times 10^8 \quad (30)$$

$$\frac{dF(Tg)}{dTg} = 3.6299 \times 10^{-4} \times Tg^3 + 7.9153 \times 10^4 \quad (31)$$

These equations were solved by the Newton-Raphson method in programme (2) to give an effective gas temperature in the fire box equal to 1278 K.

Flow rate of fuel (kmol/h) = 120
Flow rate of flue gases (kmol/h) = 1720.9
Percentage of heat losses = 0.05
Gross calorific value of fuel (kJ/kmol) = 976029.6
Molar fraction of CO ₂ = 0.08234
Molar fraction of H ₂ O = 0.15968
Molar fraction of N ₂ = 0.7179
Molar fraction of O ₂ = 0.382
Molar fraction of SO ₂ = 0.00188

Table 4. Data for the determination of flame temperature equation.

Inlet temperature of process fluid (°C) = 210
Outlet temperature of process fluid (°C) = 355
Stack temperature (°C) = 400
Flow rate of fuel (kmol/h) = 120
Flow rate of combustion air (kmol/h) = 1589.014
Flow rate of flue gases (kmol/h) = 1720.9
Number of tubes in radiation section = 60
Number of shield tubes = 8
Effective tube length (m) = 20.024
External diameter of tube in convection section (m) = 0.219
Centre-to-Centre distance of tube spacing (m) = 0.394
Net Calorific Value of fuel (kJ/kmol) = 927844.41
Molar heat of fuel (kJ/kmol.K) = 39.26

Table 5. Data for determination of the effective gas temperature equation.

5. Simulation of Heat Transfer in the Convection Section

The bases for the calculation of heat transfer in the convection section were laid for the first time by Monrad. Subsequently Schweppe and Torrijos developed a method based on the work done by Lobo and Evans on the radiation section. Other work done on the heat transfer in the convection section includes work by Briggs and Young and the work of Garner on the efficiency of finned tubes.

Heat transfer in the convection section is composed in general of the following:

1. Direct convection from the combustion gases.

A film coefficient based on pure convection for flue gas flowing normal to a bank of bare tubes may be estimated using an equation developed by Monrad:

$$h_c = 0.018 \times C_{p_{fluegas}} \times \frac{G_{max}^{2/3} \times T_{avg}^{0.3}}{D_o^{1/3}} \quad (32)$$

where $C_{p_{fluegas}}$ is the average specific heat of flue gas, and can be determined using equation (33):

$$C_{p_{\text{flue gas}}} = 1.0775 + 1.1347 \times 10^{-4} T_{\text{avg}} + 3.157 \times 10^{-8} T_{\text{avg}}^2 \quad (33)$$

2. Direct radiation from the combustion gases.

An approximate value for the radiation coefficient of the hot flue gas may be obtained using the following equation:

$$h_{\text{rg}} = 9.2 \times 10^{-2} \times T_{\text{avg}} - 34 \quad (34)$$

3. Radiation from refractory walls.

Re-radiation from the walls of the convection section usually ranges from 6 to 15% of the total heat transfer by both direct convection and radiation from the combustion gases. A value of 10% represents a typical average, i.e. $0.1 (h_c + h_{\text{rg}})$.

Based on the above equations the total convection heat transfer coefficient can be calculated from the following equation:

$$h_o = (1.1) \times (h_c + h_{\text{rg}}) \quad (35)$$

and the coefficient of overall heat transfer by convection and radiation (overall heat exchange coefficient, U_c) can then be determined using the following equation:

$$\frac{1}{U_c} = \frac{1}{h_i} + f(e, \lambda) + \frac{1}{h_o} \times \frac{S_i}{S_o} \quad (36)$$

where

$$f(e, \lambda) = \frac{R_i}{\lambda} \times \frac{R_o}{R_i} \quad (37)$$

and λ is the thermal conductivity of the tube wall which can be determined using an equation derived by curve-fitting thermal conductivity data for tube wall materials.

$$\lambda = -0.157 \times 10^{-4} \times T_w^2 + 79.627 \times 10^{-3} \times T_w + 28.803 \quad (38)$$

The value of h_i for turbulent flow, $10000 < \text{Re} < 120000$, and $L/D_o \geq 60$, is given by equation 39:

$$h_i = 0.023 \times \frac{k}{D_i} \times \text{Pr}^{1/3} \times \text{Re}^{0.8} \times \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (39)$$

where k is the thermal conductivity of the process fluid which can be estimated using an equation derived by curve-fitting thermal conductivity data of the process fluid.

$$k = 0.49744 - 29.4604 \times 10^{-5} \times t \quad (40)$$

$$\ln(\mu) = -0.2207 \times \ln^2(t) + 0.5052 \times \ln(t) - 11.8201 \quad (41)$$

Equation (41) was also obtained by curve-fitting viscosity values of the process fluid.

4. Radiation escaping from the combustion chamber into the shield section. This can be estimated using the general equation for radiation heat transfer:

$$Q_f = \sigma \times (\alpha \times Acp_{shld}) \times F \times (T_g^4 - T_w^4) \quad (42)$$

where:

$$Acp_{shld} = N_{(tube)_{shld}} \times S_{tube} \times L_{tube} \quad (43)$$

and T_w is the mean tube wall temperature which can be estimated in terms of the inlet and outlet process fluid temperatures, T_{in} and T_{out} , respectively using equation 23.

Since all heat directed towards the shield tubes leaves the radiant section and is absorbed by these tubes, the relative absorption effectiveness factor, α , for the shield tubes can be taken to equal one.

Total heat transfer in the convection section is then equal to the sum of the total heat transferred by convection and radiation into the tubes and the escaping radiation across the shield section, if applicable.

$$Q_c = Q_f + U_c \times A_c \times LMTD \quad (44)$$

Where:

Q_c = total heat transfer in the convection section.

Q_f = Escaping radiation.

A_c = Area of heat transfer.

$$LMTD = \frac{(T_1 - t_1) - (T_2 - t_2)}{\ln \frac{(T_1 - t_1)}{(T_2 - t_2)}} = \text{Log mean temperature difference}$$

A method proposed by Davalos, Fernandez and Vallejo for the simulation of cylindrical fired heaters may be used for predicting the overall behaviour of the convection section without giving information on the heat flux and temperature gradients. Such information may, however, be obtained by carrying out calculations for each layer or segment of the tubes in the convection section. This implies the use of iterative methods. A pre-requisite for such heat transfer analysis is the determination of the flue gas and process fluid temperatures in the zone separating the convection and the radiation sections, which may be calculated by solving the heat balance equation given above (Eq. 13) using the Newton-Raphson method, as previously mentioned. The intermediate flue gas and process fluid temperatures can then be calculated and the results presented in the form of temperature profiles for the combustion gases, tube walls and process fluid.

The following algorithm may be used for the calculation of the intermediate flue gas and process fluid temperatures.

1. Assume heat absorption by the first layer of tubes.
2. Calculate the flue gas and process fluid temperatures by means of an appropriate heat balance.
3. Calculate the log mean temperature difference.
4. Calculate the heat transfer coefficient for convection and radiation from the flue gas.
5. Determine the contributions of escaping radiation if the tubes in the convection section are close to the combustion chamber.
6. Compare the calculated heat absorption with the assumed value, and if close agreement found, proceed to the following layer of tubes. The total heat absorption in the convection section is determined by the summation of the amounts of heat absorbed in all layers.

Based on the above analysis, a Matlab computer programme (Appendix 4) was written for the convection section of the box-type fired heater used for heating crude oil at Homs Oil Refinery. The results obtained by this analysis are given in Table 6.

Fig. 1 shows a flow sketch for the furnace in which are indicated the combustion products, mass balance and overall energy balance and heat losses. Fig. 2 shows the temperature profiles for the process fluid, flue gas and tube wall and the amount of heat absorbed per layer in the convection section.

Outlet temperature of radiation gases (°C)	800
Outlet temperature of convection gases(°C)	400
Process fluid at radiation inlet (°C)	250
Heat liberated by combustion (kJ/h)	1.1193×10^8
Calculated heat absorption (kJ/h)	8.821×10^7
Heat absorbed in radiation section (kJ/h)	6.182×10^7
Heat absorbed in convection section (kJ/h)	2.30×10^7
Flow of fuel (kmol/h)	120
Transfer area required (m ²)	464.77
Number of required shield tubes	8

Table 6. Results of the Box-Type Heater heat transfer Simulation.

6. Thermal Efficiency

Furnace thermal efficiency is usually defined as the percent ratio of the total heat absorbed by the process fluid to the total heat input. The total heat input is the sum of the calorific value of the fuel (gross or net) and the sensible heat of all incoming streams including combustion air, fuel and atomization steam (if used). Heat losses calculated from the difference between the heat input and heat absorbed comprise both stack heat losses and radiation heat losses through furnace walls.

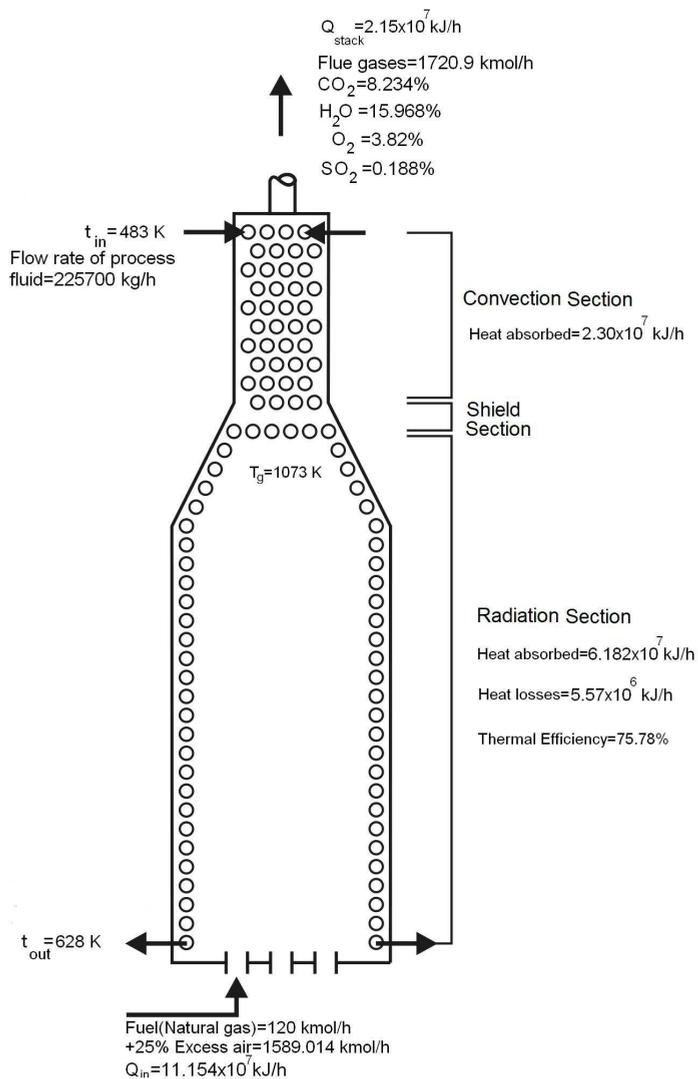


Fig. 1. Flowsketch of fired process heater

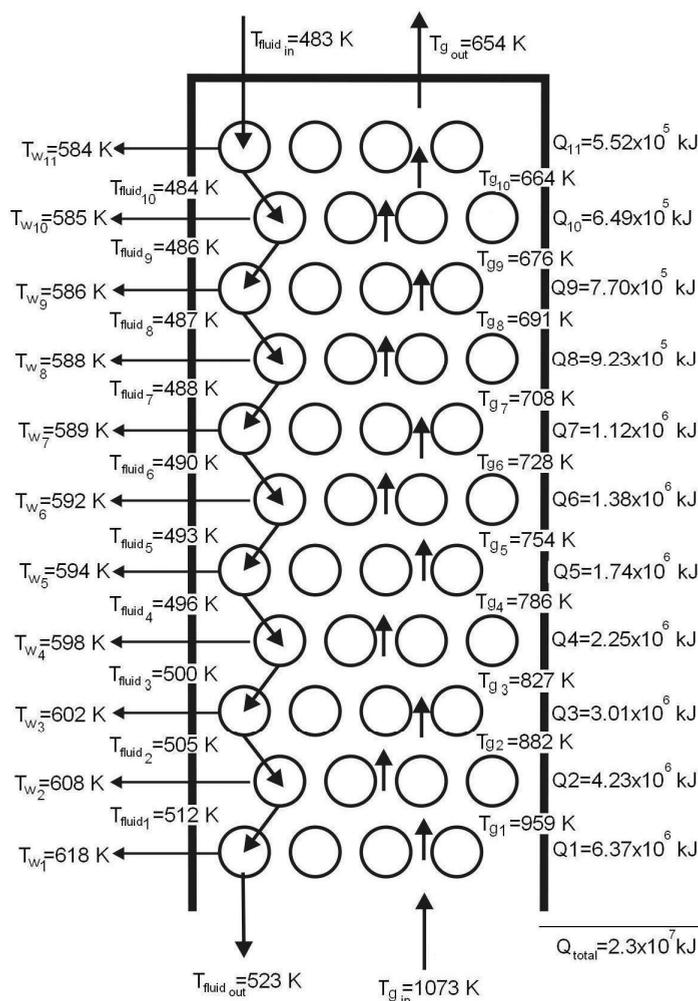


Fig. 2. Temperature profiles for combustion gases, tubewall and fluid process and absorbed heat per layer in the convection section

Stack heat losses are sensible and latent heats carried by the hot flue gases discharged through the stack. They are a function of the flue gas flow rate and temperature. Of these two factors, flue gas temperature is the main factor in furnace heat losses, and the thermal efficiency may be greatly improved if the flue gases are cooled before being discharged from the chimney. In order to cool the flue gases, a cold fluid must be available that needs to be heated. Flue gas cooling is limited, however, by corrosion problems caused by sulphuric acid condensation due to the presence of sulphur compounds in the fuels burned. If the fluid to be heated is at a temperature that is too high to give a sufficiently low flue gas temperature, i.e. a satisfactory thermal efficiency, either one of these two solutions may be

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