

**Fired Heater Simulation, Modelling and Optimization**

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In this study a dynamic model Simulation was carried out using ASPEN HYSY for industrial refinery fired heater; it was found that Increasing the number of the tube rows in convection bank from 2 to 3 allows us to recover approximately 5% of the overall efficiency, the duty furnace has increased from 65.9MW to 68.1MW and the fuel flow has increased from 5597kg/h to 5807kg/h moreover Adding more rows has a reverse return as we start to notice increase on the flue gas temperature. Furthermore, sensitivity analysis was conducted with HYSYS to determine one of the most important parameters that affect the performance of the heater based on the data generated from the simulation. MATLAB code was generated for efficiency calculation and for parameter manipulation. well-stirred model was used for mathematical model calculations and for optimization of furnace operation, the models were validated with ASPEN Exchanger Design and Rating.

**Key Words:** Efficiency, Heat Transfer, Flame Temperature, Convection Banks.**Introduction  
Furnace**

Fired heater is a device used to heat up chemicals or chemical mixtures. It's classified as direct fired or indirect fired. Direct-fired furnaces can be identified by the amount of volume, the combustion gases occupy inside the furnace. fired heater can be also classified as natural, induced, forced, or balanced draft. Fired heaters are used in many processes, including distillation, reactor processes, olefin production, and hydrocracking. The primary means of heat transfer in a fired heater are radiant heat transfer and convection and consist essentially of a battery of pipes or tubes that pass through a firebox. (Thomas 2011)

**Combustion Reaction**

The combustion reaction in the burner model of the Fired Heater performs pure hydrocarbon (C<sub>x</sub>H<sub>y</sub>) combustion calculations only. The extent of the combustion depends on the availability of oxygen which is usually governed by the air to fuel ratio. Air to fuel ratio (AF) is defined as follows:

$$AF = \frac{\text{Mass of flow O}_2}{(\sum \text{Mass of luw fuel}) / (\text{Mass ratio of o}_2 \text{ in ai r})} \quad (2)$$

The mixing of air and fuel influences flame stability, emissions, and the thermal release profile.

The role of primary air

Primary air has two major roles in jet entrainment burners:

- It controls the rate of fuel/air mixing.
- It assists with flame stability. (Alireza Bahadori 24 March 2010)

**Heating Value**

Heating value of fuel (units of KJ/kg or Mj/kg are traditionally used to quantify maximum amount of heat that can be generated by combustion with air at standard condition (STP) (25°C and 101.3 kpa). The amount of heat release from combustion of the fuel will depend on the phase of water in the product. If water is in gas phase in the product, the value of total heat denoted as the lower heating value (LHV). (Thomas 2011)

**Excess Air**

The terms excess air and excess oxygen are commonly used to define combustion. They can be used synonymously but have different units of measurements. The percentage of excess air is the amount of air above the stoichiometric requirement for complete combustion. The excess oxygen is the amount of oxygen in the incoming air not used during combustion and is related to percentage excess air., additional air beyond the theoretical "perfect ratio" needs to be added to the combustion process—this is referred to as "excess air." (Thomas 2011)

## Literature Review

### Effect of Excess Air on Fuel Consumption

Although the effect of excess air level on overall efficiency for thermal processes has been understood for many years, it is surprising how little attention is paid to this matter even today. A reduction in efficiency occurs as the oxygen level is increased above the optimal value, caused by the requirement to heat the excess oxygen and nitrogen passing through the system. This causes the flame temperature and heat transfer to reduce.

In process plants where the combustion air is preheated, the increased air flow causes a reduction in the secondary air temperature, and therefore a reduction in the flame temperature, thus requiring even more fuel to heat the charge to the required process temperature. The total increase in fuel consumption is much great-

er than that necessary to heat the excess air to the flue gas exit temperature alone.

If the excess air level in a flame is reduced below a certain level, carbon monoxide is produced. This in turn also leads to an increase in the fuel consumption, due to the incomplete combustion of carbon. The better the fuel/air mixing, the lower the excess air at which these emissions occur (Mahmoud Adam Hassan 2018)

$$\text{Excessair supplied (EA): } \frac{O_2 * 100\%}{2 - O_2\%}$$

### Effect of Stack Temperature and Excess Air on Thermal Efficiency

The effect of stack temperature has been examined and excess air on thermal efficiency and the results were found as shown in figure below.

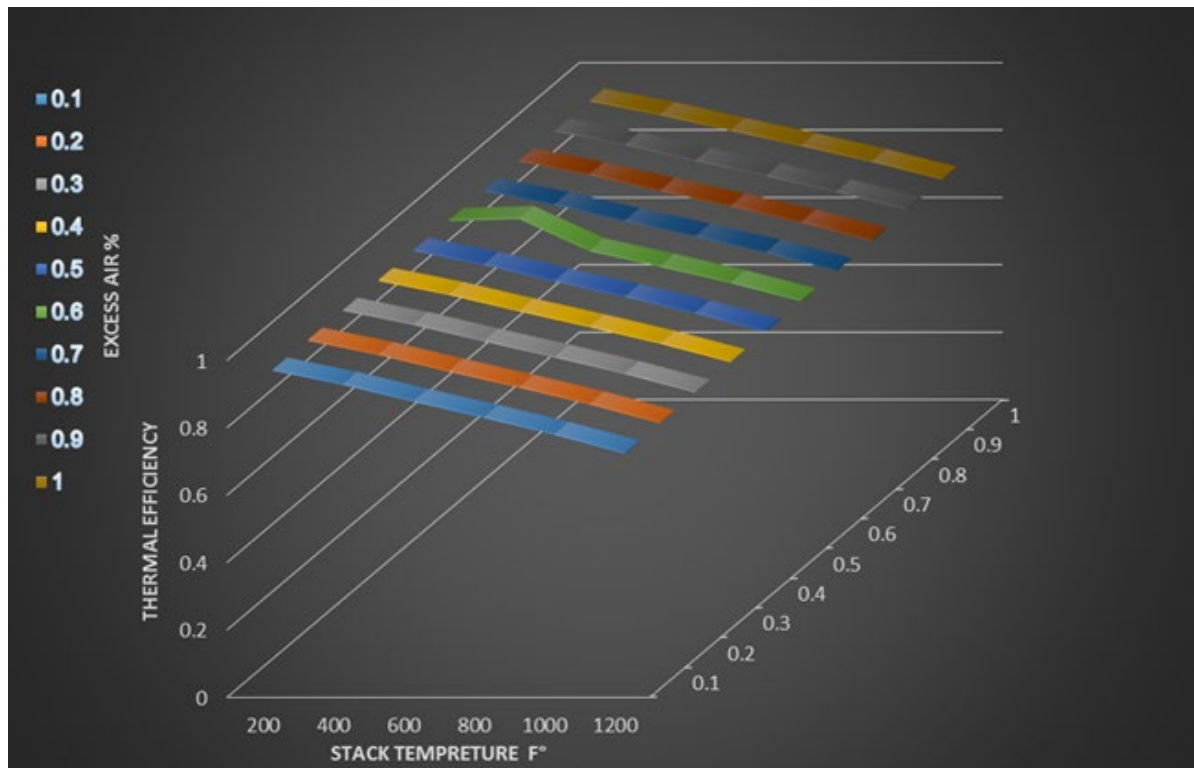
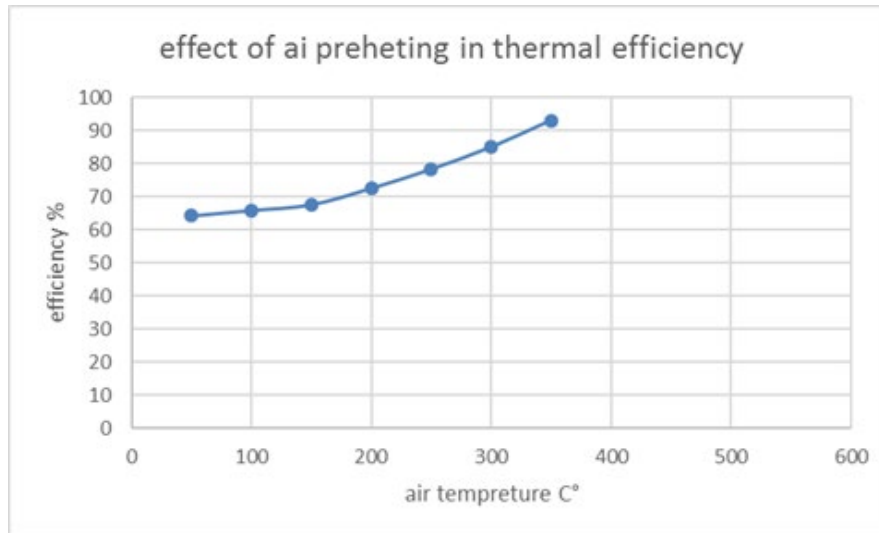


Figure 1: Effect of excess air and stack temperature on efficiency. (Mahmoud Adam Hassan 2018)

### Effect of Preheated Air on Thermal Efficiency

The effect of preheated air has been examined in HYSYS simulation and the result is shown by excel sheet in figurer below.



**Figure 2:** Effect of preheated air on efficiency

### Furnace Heat Transfer

The problem of transferring heat to crude oil in the primary furnace before it enters the crude fractionator will be undertaken. The heat from the flames passes by radiation and convection to the pipes in the furnace, by conduction through the pipe walls, and by forced convection from the inside of the pipe to the oil. Here all three modes of transfer are involved. After prolonged usage, solid deposits may form on both the inner and outer walls of the pipes, and these will then contribute additional resistance to the transfer of heat. (Rabah n.d.)

### Heat Transfer by Convection

Evaluating convective heat transfer coefficients

There are three approaches to evaluating convective heat transfer coefficients.

1. By analysis of conductive heat transfer through the boundary layer.
2. By analogy between heat, mass and momentum transfer processes;
3. By direct measurement under limited conditions and extrapolation using dynamic similarity.

We will consider the first choice by analysis of conductive heat transfer through the boundary layer.

### Streamline Flow

If a liquid enters a section heated on the outside, the fluid near the wall will be at a higher temperature than that in the center and its

viscosity will be lower. The velocity of the fluid near the wall will therefore be greater in the heated section, and correspondingly less at the center.

For the problem of heat transfer between a fluid and a tube wall, the boundary layers are limited in thickness to the radius of the pipe and, furthermore, the effective area for heat flow decreases with distance from the surface. The problem can conveniently be divided into two parts. Firstly, heat transfer in the entry length in which the boundary layers are developing, and, secondly, heat transfer under conditions of fully developed flow. [2].

For the region of fully developed flow in a pipe of length  $L$ , diameter  $d$  and radius  $r$ , the rate of flow of heat  $Q$  through a cylindrical surface in the fluid at a distance  $y$  from the wall is given by:

For the region of fully developed flow in a pipe of length  $L$ , diameter  $d$  and radius  $r$ , the rate of flow of heat  $Q$  through a cylindrical surface in the fluid at a distance  $y$  from the wall is given by:

$$Q = -k2\pi L(r - y) \frac{d\theta}{dy}$$

Close to the wall, the fluid velocity is low and a negligible amount of heat is carried along the pipe by the flowing fluid in this region and  $Q$  is independent of  $y$ .

And thus:

$$\frac{d\theta}{dy} = -\frac{Q}{k2\pi L}(r-y)^{-1} \quad \text{and} \quad \left(\frac{d\theta}{dy}\right)_{y=0} = -\frac{Q}{k2\pi Lr}$$

$$\frac{d^2\theta}{dy^2} = -\frac{Q}{k2\pi L}(r-y)^{-2} \quad \text{and} \quad \left(\frac{d^2\theta}{dy^2}\right)_{y=0} = -\frac{Q}{k2\pi Lr^2} \quad (8)$$

And then:

$$\left(\frac{d^2\theta}{dy^2}\right)_{y=0} = r^{-1}\left(\frac{d\theta}{dy}\right)_{y=0}$$

Assuming that the temperature of the walls remains constant at the datum temperature and that the temperature at any distance  $y$  from the walls is given by a polynomial, then:

$$\theta = a_0y + b_0y^2 + c_0y^3$$

Thus:

$$\frac{d\theta}{dy} = a_0 + 2b_0y + 3c_0y^2 \quad \text{and} \quad \left(\frac{d\theta}{dy}\right)_{y=0} = a_0$$

$$\frac{d^2\theta}{dy^2} = 2b_0 + 6c_0y \quad \text{and} \quad \left(\frac{d^2\theta}{dy^2}\right)_{y=0} = 2b_0 \quad (9)$$

$$\text{Thus : } 2b_0 = \frac{a_0}{r} \quad \text{from eq(8)} \quad \text{And } b_0 = \frac{a_0}{2r}$$

If the temperature of the fluid at the axis of the pipe is  $\theta_s$  and the temperature gradient at the axis, from symmetry, is zero, then:

$$0 = a_0 + 2r\left(\frac{a_0}{2r}\right) + 3c_0y^2 \rightarrow \text{Giving: } c_0 = -\frac{2a_0}{3r^2}$$

And then:

$$\theta_s = a_0r + r^2\left(\frac{a_0}{2r}\right) + r^3\left(\frac{-2a_0}{3r}\right) \quad \text{Or: } \theta_s = \frac{5}{6}a_0r$$

$$\text{and thus: } a_0 = \frac{6\theta_s}{5r} \quad \text{consequently : } b_0 = \frac{3\theta_s}{5r^2} \quad \text{and} \quad c_0 = \frac{-4\theta_s}{5r^3}$$

$$\theta = \frac{6\theta_s}{5r}y + \frac{3\theta_s}{5r^2}y^2 - \frac{4\theta_s}{5r^3}y^3 \quad \text{or: } \frac{\theta}{\theta_s} = \frac{6}{5}\frac{y}{r} + \frac{3}{5}\left(\frac{y}{r}\right)^2 - \frac{4}{5}\left(\frac{y}{r}\right)^3 \quad (10)$$

Thus the rate of heat transfer per unit area at the wall:

$$Q = -k\left(\frac{d\theta}{dy}\right)_{y=0} \quad \text{Or: } Q = -\frac{5}{6}\frac{k\theta_s}{r} \quad (11)$$

In general, the temperature  $\theta_s$  at the axis is not known, and the heat transfer coefficient is related to the temperature difference between the walls and the bulk fluid. The bulk temperature of the fluid is defined as the ratio of the heat content to the heat capacity of the fluid flowing at any section. Thus, the bulk temperature  $\theta_s$  is given by:

$$\theta_B = \frac{\int_0^r C_p \rho \theta u_x 2\pi(r-y) dy}{\int_0^r C_p \rho u_x 2\pi(r-y) dy} = \frac{\int_0^r \theta u_x (r-y) dy}{\int_0^r u_x (r-y) dy} \quad (12)$$

From Poiseuille's law:

$$u_x = \frac{-\Delta P}{4\mu L} [r^2 - (r-y)^2] = \frac{-\Delta P}{4\mu L} (2ry - y^2) \longrightarrow u_s = \frac{-\Delta P}{4\mu L} (r^2) \quad (13)$$

$$\text{And} \quad \frac{u_x}{u_s} = 2\frac{y}{r} - \left(\frac{y}{r}\right)^2 \quad (14)$$

$$\begin{aligned} \text{Thus:} \quad \int_0^r u_x (r-y) dy &= r^2 u_s \int_0^1 \left[ 2\frac{y}{r} - \left(\frac{y}{r}\right)^2 \right] \left[ \left(1 - \frac{y}{r}\right) d\left(\frac{y}{r}\right) \right] \\ &= r^2 u_s \int_0^1 \left[ 2\frac{y}{r} - 3\left(\frac{y}{r}\right)^2 + \left(\frac{y}{r}\right)^3 \right] d\left(\frac{y}{r}\right) \\ &= \frac{1}{4} r^2 u_s \end{aligned} \quad (15)$$

$$\text{Since:} \quad \frac{\theta}{\theta_s} = \frac{6}{5} \frac{y}{r} + \frac{3}{5} \left(\frac{y}{r}\right)^2 - \frac{4}{5} \left(\frac{y}{r}\right)^3$$

$$\begin{aligned} \int_0^r \theta u_x (r-y) dy &= r^2 u_s \theta_s \int_0^1 \left[ \frac{6}{5} \frac{y}{r} + \frac{3}{5} \left(\frac{y}{r}\right)^2 - \frac{4}{5} \left(\frac{y}{r}\right)^3 \right] \left[ 2\frac{y}{r} - 3\left(\frac{y}{r}\right)^2 + \left(\frac{y}{r}\right)^3 \right] d\left(\frac{y}{r}\right) \\ &= r^2 u_s \theta_s \int_0^1 \left[ \frac{12}{5} \left(\frac{y}{r}\right)^2 - \frac{12}{5} \left(\frac{y}{r}\right)^3 - \frac{11}{5} \left(\frac{y}{r}\right)^4 + 3\left(\frac{y}{r}\right)^5 + \frac{4}{5} \left(\frac{y}{r}\right)^6 \right] d\left(\frac{y}{r}\right) \\ &= r^2 u_s \theta_s \left( \frac{4}{5} - \frac{3}{5} - \frac{11}{25} + \frac{1}{2} - \frac{4}{35} \right) \\ &= \frac{51}{350} r^2 u_s \theta_s \end{aligned} \quad (16)$$

Substituting from equations 15 and 16 in equation 12:

$$\begin{aligned} \theta_B &= \frac{\frac{51}{350} r^2 u_s \theta_s}{\frac{1}{4} r^2 u_s} \\ &= \frac{102}{175} \theta_s = 0.583 \theta_s \end{aligned}$$

The heat transfer coefficient  $h$  is then given by:

$$h = -\frac{q}{\theta_B} \quad (17)$$

where  $q$  is the rate of heat transfer per unit area of tube.

$$\text{Thus: } h = \frac{-\frac{6k\theta_s}{5r}}{0.583\theta_s} = \frac{2.06k}{r} = 4.1 \frac{k}{d} \quad (18)$$

Or  $Nu=4.1^3$

### Heat Transfer by Radiation

$$Q = (SG_1)_R \sigma (T_g^4 - T_1^4) \quad (19)$$

### Furnace Efficiency

Running furnaces efficiently is a major operating concern because two thirds of a plant's fuel budget is needed for furnace fuel cost. Furnace efficiency is linked to environmental regulations that stipulate a clean operation. Most furnaces use fuel gas or fuel oil. Natural gas burns cleaner and more efficiently than oil [3]. (Barrie Jenkins 2008)

Furnace efficiency or total furnace efficiency is the ratio of heat usefully absorbed and total heat supplied.

Efficiency = heated usefully absorbed by heated medium / total heat supplied \* 100 (20)

Furnace efficiency  
There are two methods to calculate efficiency: -  
Direct method: -

$$E = \left( \frac{Q_{in} - Q_{outt}}{Q_{in}} \right) * 100\% = \left( \frac{Q_U}{Q_{in}} \right) * 100\% \quad (21)$$

$$Q_{in} = m_{fuel} * NCV + m * H_{fuel} + m_{air} * H_{air} + m_{fluid} * H_{fluid} \quad (22)$$

$$Q_U = m_{fuel} * NCV + m * H_{fuel} + m_{air} * H_{air} + m_{fluid} * H_{fluid} - (m_{fluegas} * e_{fluegas} * (t_g - t_f)) + percentage(2-5\% * m_{fuel} * NCV \quad (23)$$

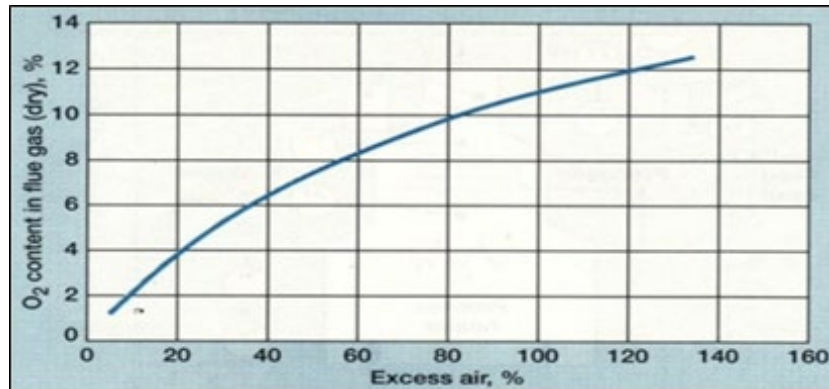
Indirect method

$$e = ((LHV + Ha + Hf) - Q_s - Q_r) / ((LHV + Ha + Hf)) * 100\% \quad (24)$$

$$\text{Weight} = (\text{volume fraction} * \text{molecular weight}) \quad (25)$$

$$\text{Heating value} = (\text{net heating value} * \text{weight}) \quad (26)$$

$$\text{Stoichiometric oxygen} = (\text{oxygen from combustion reaction} * \text{volume fraction}) \quad (27)$$



**Figure 3:** Relation between excess air % and oxygen% in flue gas.

-Actual oxygen required = Stoichiometric oxygen + (excess air \* Stoichiometric oxygen) (28)

- Actual air required = actual oxygen required \* (100/21) (29)

- Estimate stack component (CO<sub>2</sub>, H<sub>2</sub>O, SO<sub>2</sub>, O<sub>2</sub>, N<sub>2</sub>).

-Amount of CO<sub>2</sub> in flue gas = (total formed + CO<sub>2</sub> reported as fuel) (30)

-Amount of H<sub>2</sub>O in flue gas = (total formed + H<sub>2</sub>O reported as fuel) (31)

-Amount of SO<sub>2</sub> in flue gas = (total formed + SO<sub>2</sub> reported as fuel) (32)

-Amount of O<sub>2</sub> in flue gas = (actual O<sub>2</sub> supplied - actual O<sub>2</sub> used during combustion) (33)

-Amount of N<sub>2</sub> in flue gas = (79% of moles of air + N<sub>2</sub> reported as fuel) (34)

### Lower Heating Value

LHV =  $\Sigma$  heating value /  $\Sigma$  weight of component of fuel (35)

### Stack Losses

1. (amount of stack component \* molecular weight) (36)
2. (result from (1) / amount of fuel enter the furnace).
3. result from (2) \* enthalpy for each stack component).
4. summation of all result in (3).

### radiation losses: -

The radiation heat losses were determined by multiplying heat input fuel (LHV) by the radiation losses expressed as percentage. Therefore, radiation heat losses = (1% + 3%) / 2 = 2% of heat input (LHV).

sensible heat correction for combustion of air (H<sub>a</sub>): -

H<sub>a</sub> = lb of air / lb of fuel \* C<sub>p,air</sub> \* (T<sub>i</sub> - T<sub>d</sub>) (Mahmoud Adam Hassan 2018) (37)

T<sub>t</sub> = combustion air temperature.

T<sub>d</sub> = datum temperature (60°F).

sensible heat correction for fuel (H<sub>f</sub>): -

H<sub>f</sub> = C<sub>p,fuel</sub> \* (T<sub>t</sub> - T<sub>d</sub>) (38)

### Combustion and Heat Transfer Modelling

Designing an industrial furnace generally requires the simultaneous resolution of transient heat, mass and momentum transfer

phenomena in a complex chemically reacting system modelling is indisputably the most reliable technique, relying on the representation of the significant variables of the full-size system in a physical and/or mathematical model that is based on sound, proven and validated theory. Modelling is termed as partial because it is not possible to satisfy all the scaling criteria or algorithms required for a complete model. and requires the modeler to be skilled in both an understanding of the design objectives, as well as the mechanics of the modelling process.

Steps of modelling fired heater are as follows:

1. Obtain or derive process and operating data.
2. Designing the model.
3. Validate the model against measured or design process data.
5. Convert the optimized model result to an engineering solution.

### Mathematical Modelling

According to Lobo and Evans (1939) and Hottel (1954) two models' methods which are significantly important when considering modelling of fired heaters, they will be discussed in the following section [4].

### Well-Stirred Furnace Model

The most important development of fired heater model was made by Hottel (1954- 1974) and is based on the assumption that many industrial furnaces works with sufficient momentum in the air and/ or fuel streams to create a reasonably well-stirred furnace chamber.

This assumption allows most of the complex geometric problems associated with radiative heat transfer calculations to be reduced to a numerically simple solution. For this type of model, the following simplifying assumptions are made:

- (1) Mixing is intense so that there are no concentration or temperature gradients in the furnace.
- (2) Reactants-products can be considered as a single zone.

The heat release from gas can be obtained thus:

(1) Calculate the radiative heat transfer to the furnace surfaces from gases at gas temperature,

T<sub>g</sub>. This requires:

- (a) Furnace geometry and surface properties (SG)<sub>R</sub> ;

(b) Gas composition (usually known for a specific fuel and combustion air);

(c) gas flame  $\varepsilon_g$

(2) Calculate the convective heat transfer from gases at gas temperature,  $T_g$ . This requires the

convective heat transfer coefficients which can be found from:

(a) Published data for flows over surfaces, tube banks, etc.;

(b) Published data for jets expanding in a duct;

(c) Mass transfer measurements on small scale models of furnace burner system. (MOHAMMED 2007)

the net heat exchange,  $Q$ ,

$$Q = (SG_1)_R \sigma (T_g^4 - T_1^4) \quad (39)$$

If the flux geometry is calculated from:

$$(SG_1)_R = \frac{A_T}{\left( \left[ \frac{1}{\varepsilon_g} \right] + \left[ \frac{1}{\{C_s \varepsilon_1\}} \right] - 1 \right)} \quad (40)$$

and energy balance could be written as:

$$\frac{H_f - Q}{H_f} = \frac{T_g - \Delta - T_o}{T_{af} - T_o} \quad (41)$$

$$Q' = \frac{Q(T_{af} - T_o)}{H_f T_{af}} \quad D' = \frac{H_f}{\sigma(GS_1)_R T_{af}^3 (T_{af} - T_o)} \quad \Delta' = \frac{\Delta}{T_{af}} \quad \tau = \frac{T_1}{T_{af}} \quad (9)$$

(Industrial and Process Furnaces handbook n.d.)

$$Q' D' + \tau^4 = (1 - \Delta' - Q')^4 \quad (46)$$

if:

$$\Delta' = \left(1 - \frac{1}{d}\right) Q'$$

where  $d$  is a constant of proportionality, thus we have

$$Q' D' + \tau^4 = \left(1 - \frac{Q'}{d}\right)^4 \quad (47)$$

Figure (1) shows how the thermal performance of many common furnaces can be related to equation (27). It can be seen that analysis of furnaces using this formula implies that the most efficient performance for any given  $\tau$  is obtained at the largest value of  $D'$ .

From equation (19) we get:

$$\frac{Q}{(SG_1)_R \sigma} + T_1^4 = (T_g^4) \quad (42)$$

eliminating  $T_g$  from equation (21):

$$\left(1 - \frac{Q}{H_f}\right) (T_{af} - T_o) + \Delta + T_o = T_g \quad (43)$$

$$\left[T_{af} - T_o - \frac{Q}{H_f} (T_{af} - T_o) + \Delta + T_o\right]^4 = \frac{Q}{(SG_1)_R \sigma} + T_1^4 \quad (44)$$

Dividing equation (24) by  $T_{af}$  and arranging:

$$\frac{Q}{\sigma(GS_1)_R} + T_1^4 = T_{af}^4 \left[1 + \frac{\Delta}{T_{af}} - \frac{Q(T_{af} - T_o)}{H_f T_{af}}\right]^4 \quad (45)$$

following dimensionless groups are simplified equation (25)

Since  $D'$  is a function of  $(GS_1)_R$ , the implication is that  $A_T$  should be as large as possible.

For convective heat transfer, and for radiation losses from furnace openings, thus equation (39) can be rewritten as

$$Q = (SG_1)_R \sigma (T_g^4 - T_1^4) - UA_g (T_g - T_o) + (SG_0)_R \sigma (T_g^4 - T_o^4) \quad (48)$$



This can be calculated from flux geometry as

$$(SG_0)_R = \frac{A_T}{\left( \left[ \frac{1}{\epsilon_g} \right] + \left[ \frac{1}{\{C_0 \epsilon_{01}\}} \right] - 1 \right)}^{10} \quad (49)$$

To eliminate  $Tg$  between equations as before, and define some extra dimensionless groups as follows:

$$\tau_o = \frac{T_o}{T_{af}} \quad L'_R = \frac{UA_R}{\sigma(GS_1)_R T_{af}^3} \quad L'_o = \frac{(GS_0)_R}{(GS_1)_R}$$

then thus a new design equation driven as:

$$Q'D' = \left(1 - \frac{Q'}{d}\right)^4 - \tau^4 + L'_R \left[ \left(1 - \frac{Q'}{d}\right)^4 - \tau_o \right] + L'_o \left[ \left(1 - \frac{Q'}{d}\right)^4 - \tau_o^4 \right] \quad (50)$$

the furnace efficiency,  $\eta$ , is then calculated from

$$\eta = \frac{Q - \text{Losses}}{H_f} = \frac{Q'}{1 - \tau_o} - \frac{L'_R}{D'} \left[ 1 - \frac{Q'/d}{1 - \tau_o} \right] - \frac{L'_o}{D'(1 - \tau_o)} \left[ \left(1 - \frac{Q'}{d}\right)^4 - \tau_o^4 \right] \quad (51)$$

This equation indicates that the efficiency reaches a maximum value at a particular  $D'$  for any set of  $\tau$ ,  $L'_R$  and  $L'_o$  conditions, as shown in Figure above, thus we can potentially use equation (31) to optimize furnace designs.

### Two- And Three-Dimensional Zone Models

The basis of zone models is that the furnace enclosure is divided into volume and surface zones, each having consistent physical and chemical properties. From the grid thus generated, view factors and hence direct exchange areas are calculated between each zone and all other zones in the furnace using pre-calculated tables for simple geometries, or Monte Carlo methods for more complex geometric shapes. From the direct exchange areas, total exchange areas are then calculated, which account for the indirect effect of multiple reflections of radiation at other zones. The radiative and absorptive effects of the combustion gases, soot, particulates, etc.,

are allowed for by using an approximation based on the weighted sum of a clear plus a number of grey gases (typically two or three). Heat balances can then be formulated for each zone, which also accommodate enthalpy changes due to combustion, flow of gases between zones, chemical reactions, and convective heat transfer.

The first step is to evaluate the direct exchange areas between all surface and volume zones. These exchange areas must be evaluated for each absorption coefficient in the grey gas fit. (Barrie Jenkins 2008).

The next step is to calculate the total radiation incident on/reflected from each zone due to direct exchange and reflected exchange. The leaving flux density,  $W$ , at any surface zone is equal to the emitted and reflected fluxes. A radiation balance on surface zone  $j$  gives

$$A_j W_j = A_j (\epsilon_j E_{\dot{j}} + R_j) = A_j \epsilon_j E_{\dot{j}} + \gamma_j \left( \sum_k \overline{s_s s_k} W_k \right) + \sum_k \overline{s_s g_k} E_{\dot{k}} \quad (52)$$

Or:

$$\sum_k \left( \overline{s_j s_k} - \delta_k \frac{A_j}{\gamma_j} \right) W_k = - \frac{A_j \epsilon_j E_{\dot{j}}}{\gamma_j} - \sum_k \overline{s_j g_k} E_{\dot{k}} \quad (53)$$

Thus, a series of equations similar to equation (33) built up with differing values of flux density for each surface/surface and surface/gas zone pair. From these values of flux density, the total exchange areas between surface/surface, surface/gas and gas/gas zone pairs can be calculated. Thus

$$\overline{(S_j S_k)}_n = \frac{A_j \epsilon_j}{\gamma_j} (W_k - \delta_k \epsilon_j) \quad (54)$$

$$\overline{(G_j S_k)}_n = \frac{A_j \epsilon_j}{\gamma_j} W_k \quad (55)$$

$$\overline{(G_j G_k)}_n = \overline{g_i g_k} + \sum \overline{g_k S_w g} W_w \quad (56)$$

where  $n$  is the number of absorption coefficients,  $K_n$ , in the grey gas fit.

For a given temperature field in the system, the total flux areas for each zone pair are calculated using the temperature dependent grey gas weighting factors, thus, we have

$$\overline{S_j S_k} = \sum_n (a_n(T_i)) \overline{(S_j S_k)}_n \quad (57)$$

And (58)

$$\overline{(S_j S_k)} = \sum_n (a_n(T_k)) \overline{(S_j S_k)}_n$$

Total energy balances can now be written for all surface and gas zones taking into account convection to surface zones, enthalpy changes in the zone,  $Q_{ej}$ , energy release within gas and surface zones,  $Q_{ej}$ , and transient terms for storage of energy within gas zones,  $Q_{ugj}$ . Thus, we have for surface zone  $A_j$

$$Q_{inlet} = \sum_k \overline{S_k S_j} E_k + \sum_k \overline{G_k S_k} E_k + U_j A_j (T_k - T_j) - A_j \epsilon_j E_j \quad (59)$$

And for volume zone  $V_j$ :

$$\sum_k \overline{G_k S_k} E_k + \sum_k \overline{S_k S_j} E_k - \sum_n 4a_n K_n V_j E_k + Q_p = Q_{ugj} - Q_e \quad (60)$$

The resulting non-linear heat balances are rapidly converged to a solution using a combination of the Gauss-Seidel and Newton-Raphson methods.

### Simulation

In this section refinery fired heater is simulated and the Long furnace models and Simple well-stirred furnace models are validated.

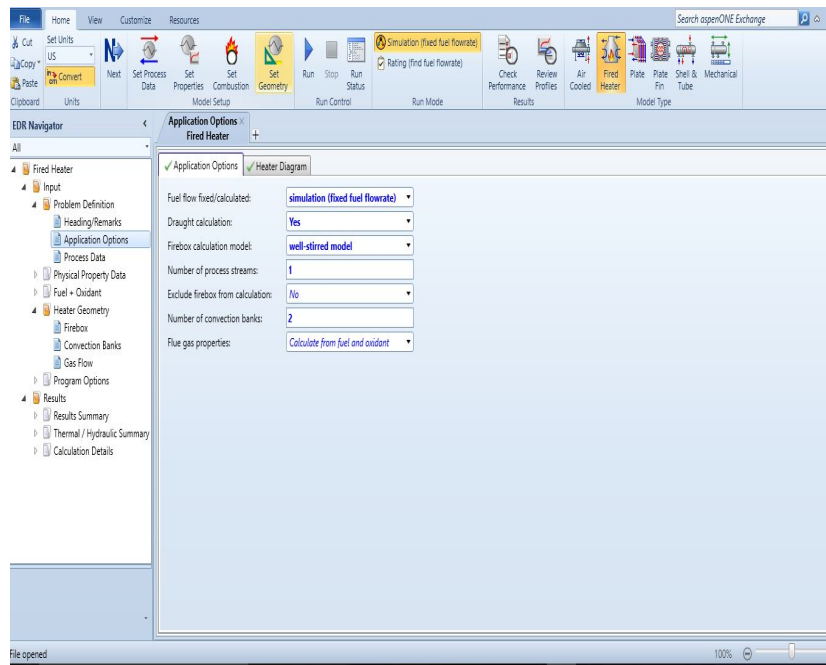
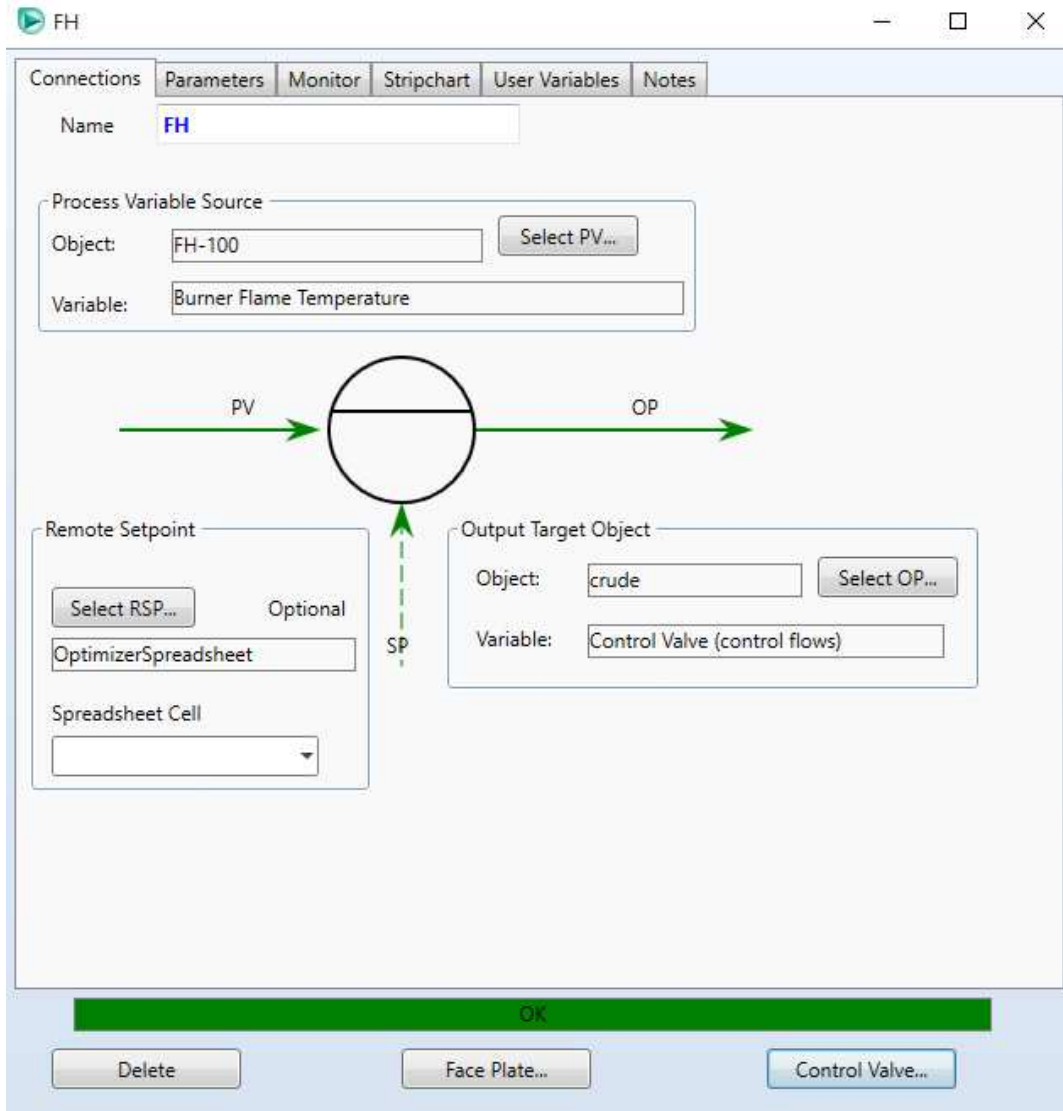
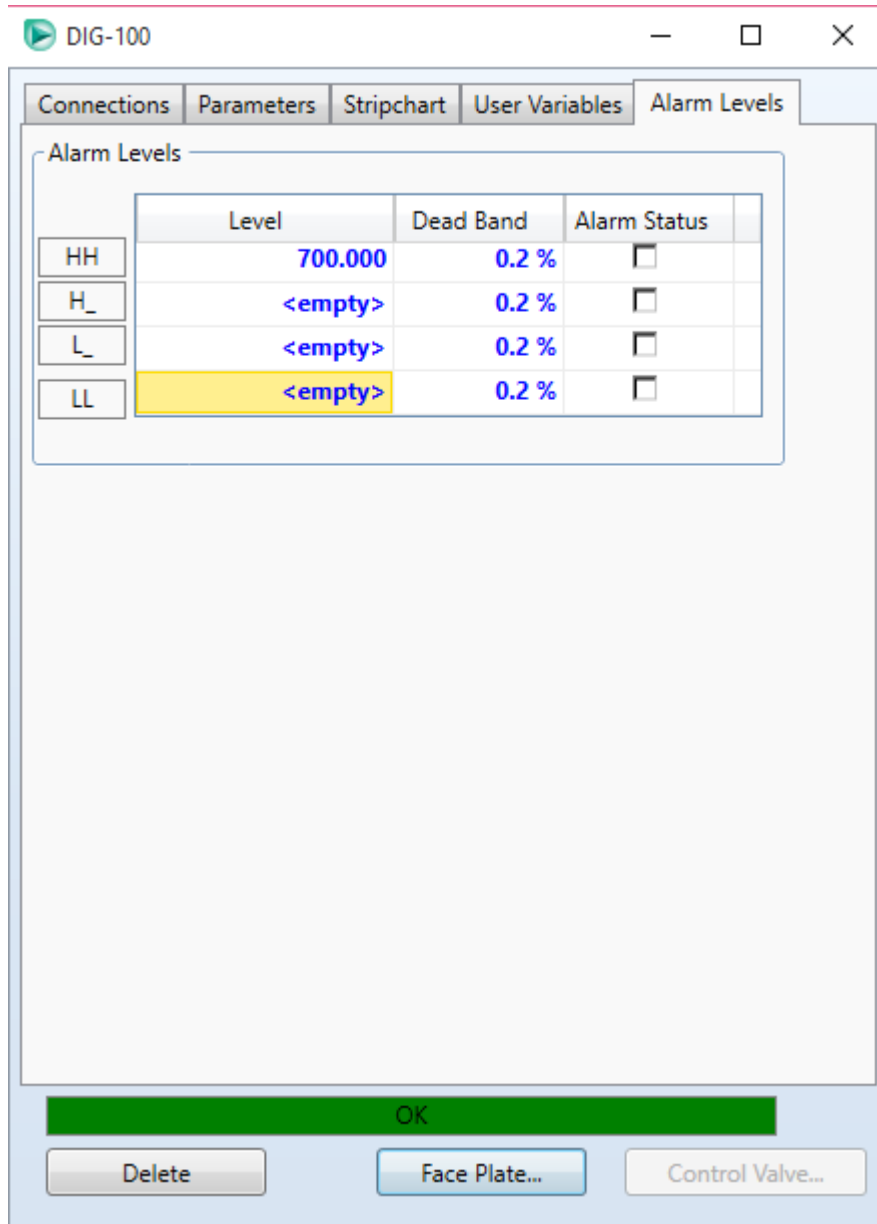


Figure 4: well-stirred model validation



**Figure 5:** controller



**Figure 6:** setting up PID controller

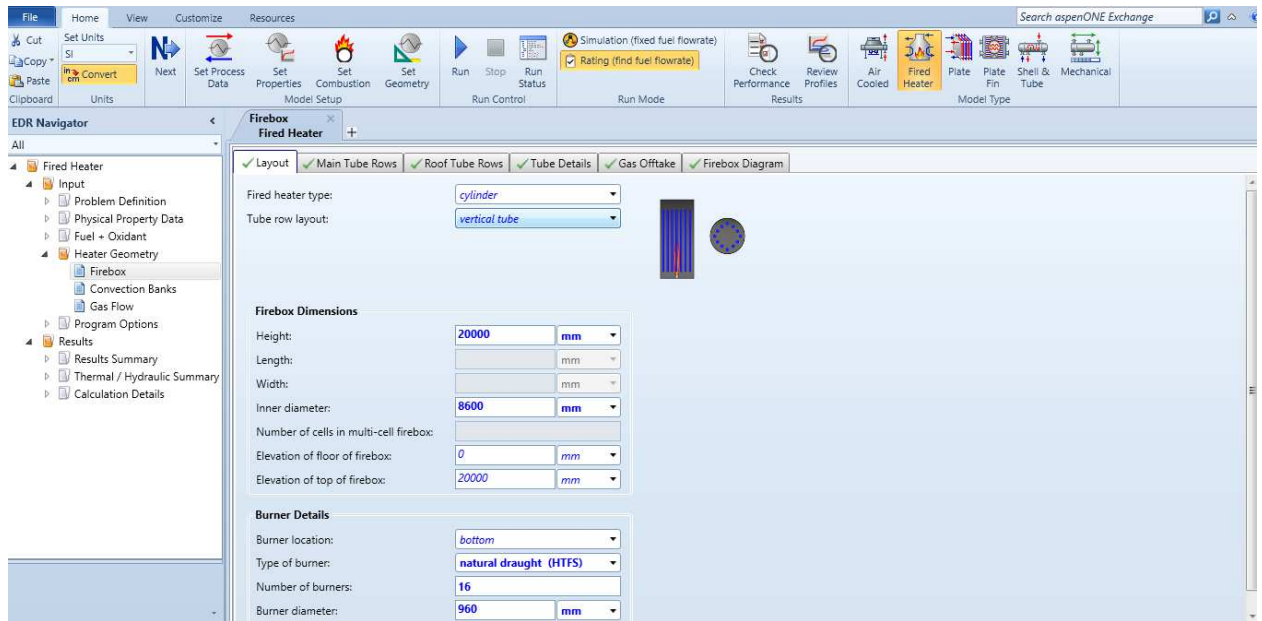


Figure 7: EDR fired heater

### Case study

Studying different effects of parameters to the furnace by inserting conditions

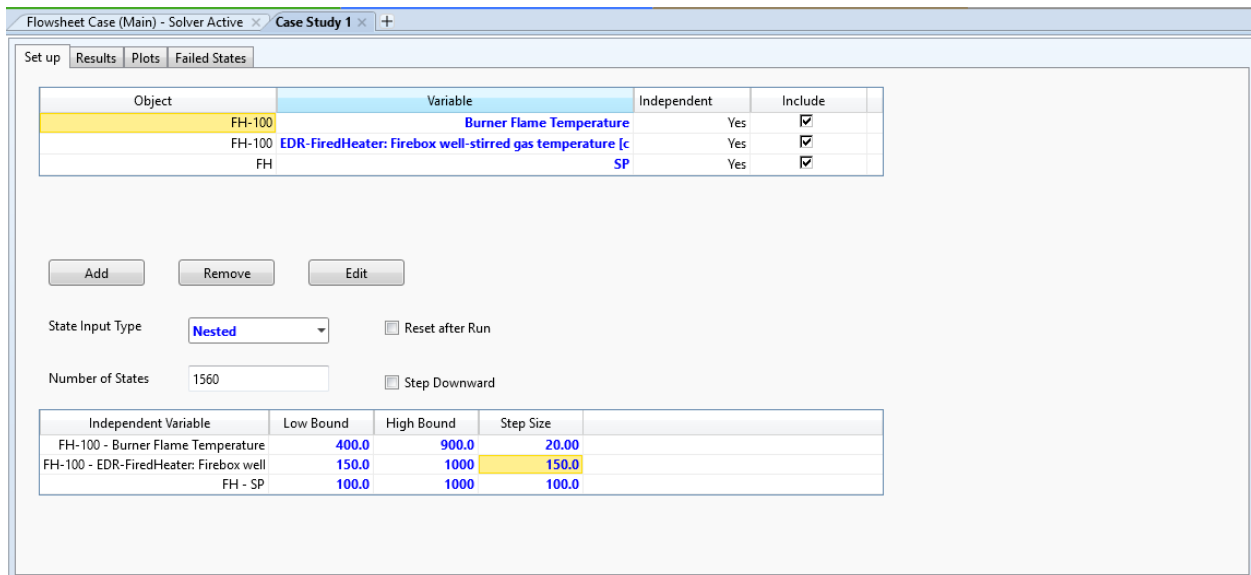
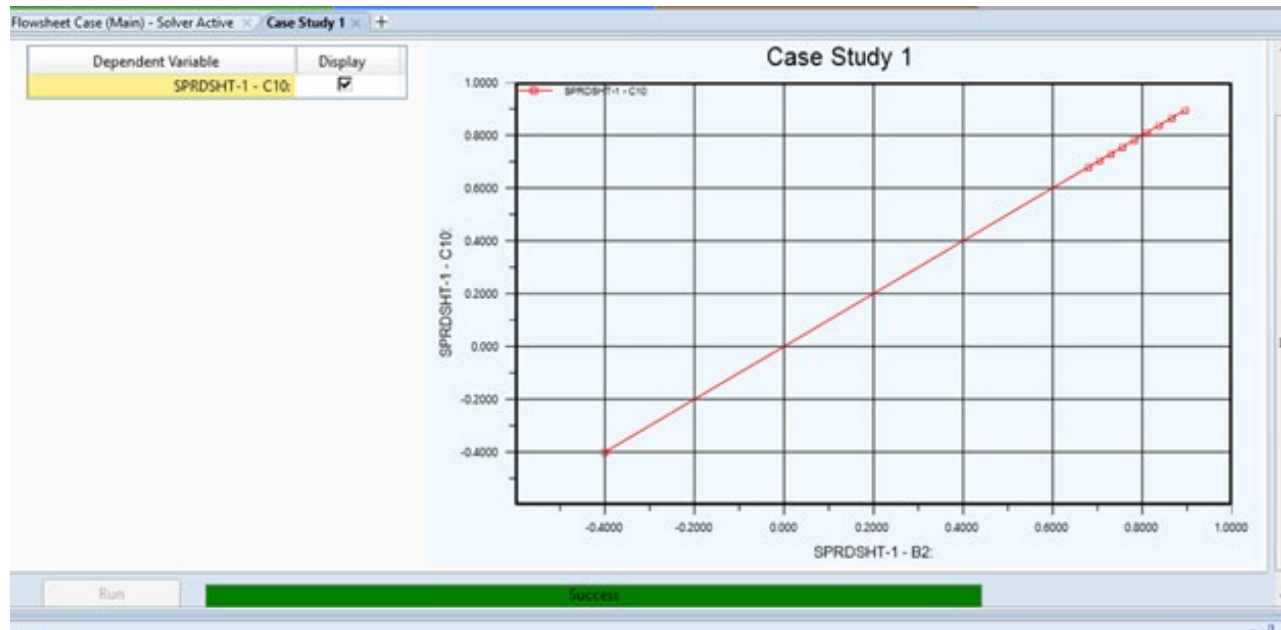


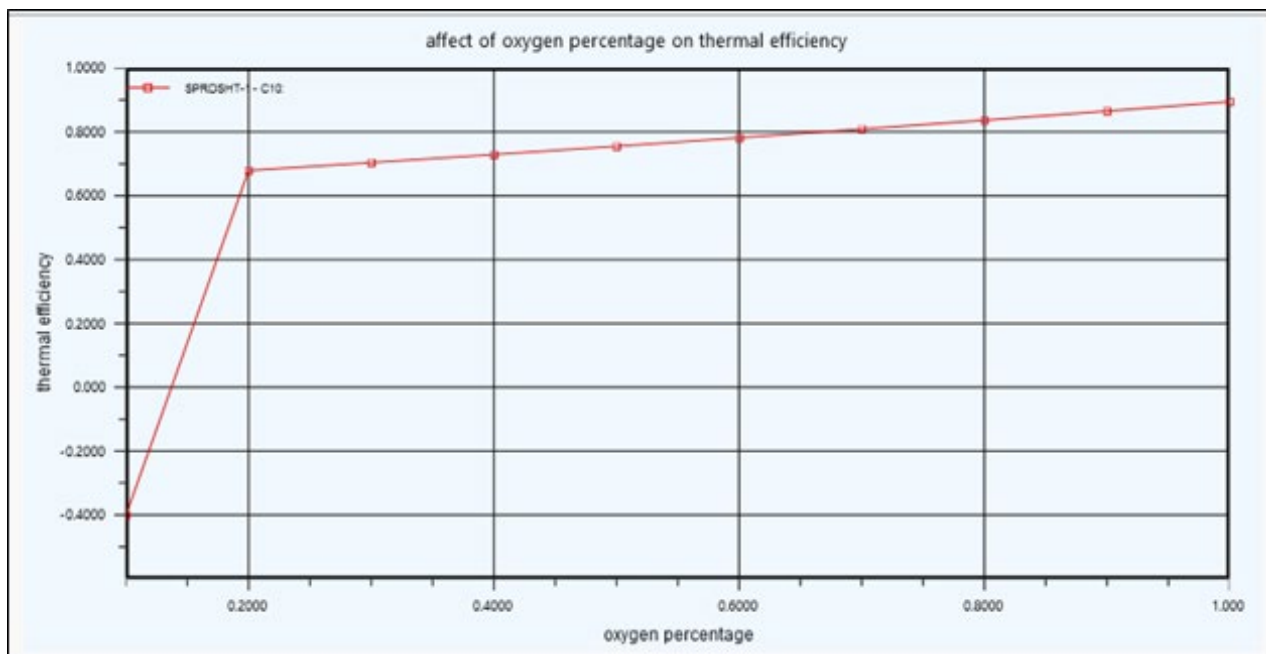
Figure 8: running case study





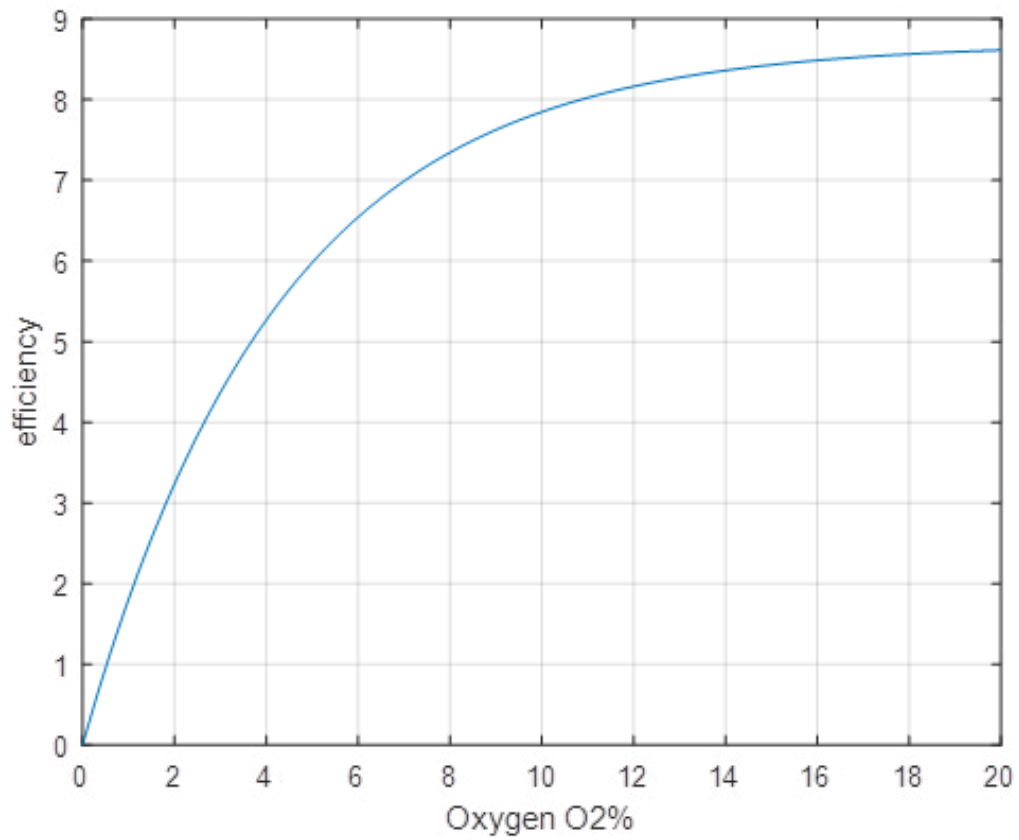
**Figure 11:** flame temperature effect

The effect of oxygen percentage has been examined in HYSYS simulation environment and the result shown in figure below.



**Figure 12:** Effect of  $O_2$  ratio

Effect of  $O_2$  in inlet air on furnace thermal efficiency has been inducted by MATLAB (SEE APPENDIX) giving the result in the figure shown below



**Figure 13:** effect of oxygen in inlet air

Excel program is used to calculate the efficiency of the furnace using the indirect method



component	mole/hr	net.value	weight	heat.value	stoichiometric oxygen	co2	h2o	so2										
ch4	49.67	21500	794.72	17086480	99.34	49.67	99.34	0										
h2	9.48	51600	18.96	978336	4.74	0	9.48	0										
c2h6	19.57	20420	587.1	11988582	68.495	39.14	58.71	0										
c2h4	3.12	20290	87.36	1772534.4	9.36	6.24	6.24	0										
c3h8	5.11	19930	224.84	4481061.2	25.55	15.33	20.44	0										
c3h6	3.09	19690	129.78	2555368.2	13.905	9.27	9.27	0										
c4h10	1.93	19670	111.94	2201859.8	12.545	7.72	9.65	0										
c4h8	0.92	19420	51.52	1000518.4	5.52	3.68	3.68	0										
c5h12	0.84	19500	60.48	1179360	6.72	4.2	5.04	0										
N	1.38	0	38.64	0	0	0	0	0										
co	2.44	4345	68.32	296850.4	1.22	2.44	0	0										
co2	2.43	0	106.92	0	0	0	0	0										
h2s	0.0036	6550	0.1224	801.72	0.0054	0	0.0036	0.0036										
sum			2280.7024	43541752.12	247.4004	137.69	221.8536	0.0036										
reaction	oxygen	co2	h2o	so2														
ch4	2	1	2	0														
h2	0.5	0	1	0		total fuel					2280.702 lb/hr							
c2h6	3.5	2	3	0		LHV					19091.38							
c2h4	3	2	2	0		theoretical oxygen					247.4004							
c3h8	5	3	4	0		by oxygen analyzer oxygen in flue gas					3.10%							
c3h6	4.5	3	3	0		from figure excess air					1							
c4h10	6.5	4	5	0		actual oxygen required					494.8008							
c4h8	6	4	4	0		actual air required					2356.194 mole/hr							
c5h12	8	5	6	0							68329.63 lb/mole							
n	0	0	0	0							29.95991 lb/lb of fuel							
co	0.5	1	0	0														

Figure 14: efficiency calculation (1)

From the above calculation the efficiency reaches 83%

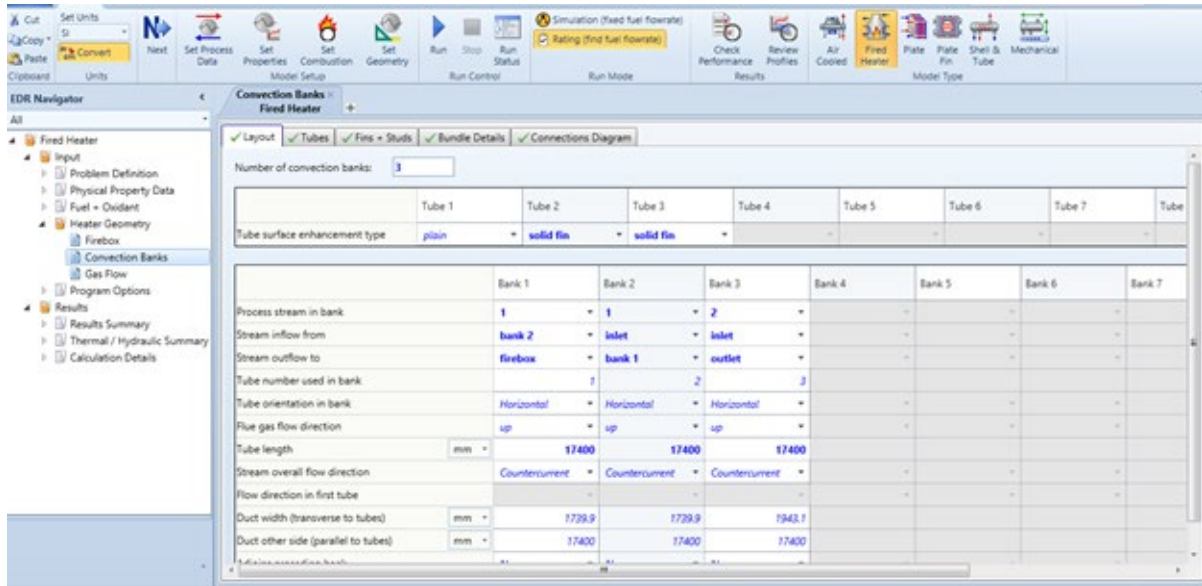


Figure 15: Convection banks

Recap of Design

Current selected case: A

		A
Overall efficiency (%)	-	87.1
Firebox efficiency (%)	-	58.01
Convection section efficiency % (LCV)	-	29.09
Firebox process stream inlet temperature	°C	195
Firebox process stream outlet temperature	°C	280
Firebox process stream duty	kW	31174.2
Fuel flowrate	kg/s	1.5005
Volumetric heat release	kW/m <sup>3</sup>	43.2
Firebox well-stirred gas temperature	°C	917
Bridgewall gas temperature	°C	841
Peak firebox tube outer wall temperature	°C	320
Hottel bridge-wall parameter		0.06615718
Stream1 temperature in	°C	149
Stream1 temperature out	°C	280
Stream1 pressure in	bar	14
Stream1 pressure out	bar	12.14389

Buttons: Delete, Select Case, Customize

Figure 16: EDR result

It was founded that Increasing the number of finned tube rows in convection bank (from 3 to 6) allows us to almost recover the operating efficiency. Adding more rows has a diminishing return as we start to get pinched on the flue gas temperature difference

### Conclusion

we can notice that the comparison of the effect of oxygen percent on the inlet air of the furnace to the above results in figure (13) and in figure (14) which were conducted by MATLAB and

ASPEN HYSYS environment simultaneously its almost has the same effect, increasing oxygen percent improve the efficiency of the furnace

### Future Studies

Using Smaller tube diameter and more tubes on edr fired heater convection banks Use different simulation like CHEMCAD and compare results

### Nomenclature

$e$  = Net thermal efficiency.

LHV = Lower heating value of fuel (BTU/LB).

$H_a$  = Heat input in form of sensible heat of air (BTU/LB).

$H_f$  = Heat input in form of sensible heat of fuel (BTU/LB).

$Q_s$  = Heat stack losses (BTU/LB).

$Q_r$  = Radiation heat losses (BTU/LB).

$A$  = area

$c_d$  = drag coefficient

$C$  = constant

$C_o$  = furnace openings fraction

$C_p$  = specific heat at constant pressure

$C_s$  = cold surface fraction =  $A_S / A_T$

$C_t$  = Curtet number

$D$  = firing density

$d$  = diameter, constant of proportionality

$g$  = gravitational constant

$G$  = generation rate of turbulent energy

$(GS_0)_R$  = exchange area between gas and ambient atmosphere in radiative equilibrium

$(GS_1)_R$  = exchange area between gas and surface 1 in radiative equilibrium

$h$  = enthalpy

$H$  = heat input

$I$  = interphase source term

$k$  = jet shape parameter, turbulent kinetic energy

$L$  = half width, heat loss

$m$  = mass flowrate, Craya-Curtet parameter

$P$  = pressure

$q$  = primary jet momentum flux

$q_{rad}$  = energy from thermal radiation

$Q$  = total momentum flux, net heat exchange

$r$  = radius

$R$  = momentum ratio  $q / Q$ , volume fraction

$S$  = model scale, phase source term

$t$  = time

$T$  = temperature

$u$  = velocity

$U$  = overall convective heat transfer coefficient

$v$  = volumetric flowrate

$V$  = velocity vector  
 $x, X$  = length scale  
**Subscripts**  
 $a$  =secondary  
 $af$  = adiabatic flame  
 $d$  = dynamic  
 $e$  = entrained  
 $eff$  = effective (also called turbulent or eddy)  
 $f$  =flame  
 $g$  = gas  
 $i$  =refers to phase I  
 $j, k, n$  = identifiers  
 $k$  = kinetic  
 $o$  =starting or ambient  
 $R$  = refractory  
 $S$  = sink  
 $T$  = total  
 $1, 2$  = identifiers  
**Superscripts**  
 $'$  =dimensionless  
 $*$  = characteristic  
**Greek letters**  
 $\Delta$ = temperature difference  
 $\varepsilon$  =emissivity, rate of energy dissipation  
 $\eta$ = thermal efficiency  
 $\mu$ = viscosity – cannot find this in the m/s  
 $\rho$ = density  
 $\sigma$  =Stefan-Boltzmann constant  
 $\sigma_k$  =diffusion constant  
 $\sigma_\varepsilon$  =dissipation constant  
 $\tau$  =temperature ratio  
**Dimensionless groups**  
 $Re$  \_ Reynolds number =  $ux\rho/\mu$   
 $Sc$  \_ Schmidt number =  $\mu/\rho x$

## References

1. Charles E. Thomas. (2011). "Process Technology Equipment and Systems Third Edition".
2. Coulson & Richardson's, chemical engineering, volume 1.
3. Barrie Jenkins, Peter mullinger. (2008). Industrial and Process Furnaces Principles, Design and Operation.
4. Lobo, W. E., & Evans, J. E. (1939). Heat transfer in the radiant section of petroleum heaters. Trans. Am. Inst. Chem. Engrs, 35, 748-778.
5. Garg, A. (1997). Optimize fired heater operations to save money. Hydrocarbon Processing, 76, 97-112.
6. Jyh-Yunan Chen, A. Carlos Fernandez-Pello. (2011). Fundamental of combustion process.
7. Ali Rabah, Chemical engineering heat transfer, detailed approach, University of Khartoum, Volume 1.
8. Shuaibu Ndache MOHAMMED, Investigating the Effect of Changes in Furnace Operating Conditions Using the Well-Stirred Model, ISSN 1853-1078.
9. Jenkins, B., & Mullinger, P. (2011). Industrial and process furnaces: principles, design and operation. Elsevier.

10. Hottel, H. C., & Sarofim, A. F. Radiative Transfer, McGraw-Hill, New York, 1967.
11. Energy and Environmental Research Corporation, 18 Mason, Irvine, CA 92714, January (1982).
12. Mahmoud.A.H. salih, American journal of quantum chemistry and molecular spectroscopy.
13. Manual of Khartoum Refinery Company (KRC) limited.2011.
14. HYPERLINK "<https://doi.org/10.1016/j.fuproc.2010.03.033>" Bahadori, A., & Vuthaluru, H. B. (2010). Estimation of energy conservation benefits in excess air-controlled gas-fired systems. Fuel Processing Technology, 91(10), 1198-1203.

## APPENDIX 1

function indirectmethod=indirectefficiency

% This program calculates the thermal %efficiency using direct Method

% calculate volume fraction of each %component

HVF=0.5;

COVF=2.44;

CO2CF=2.43;

CH4VF=49.67;

C2H6VF=19.57;

C2H4VF=3.12;

C3H8VF=5.11;

C3H6VF=3.09;

C4H10VF=1.93;

C4H8VF=0.92;

C5H12VF=0.84;

H2SVF=0.0036;

NVF=1.38;

O=0.031; %O2 Percent in flue gas

% Net heat value of each component

HNV=51600;

CONHV=4345;

CH4NHV=21500;

C2H6NHV=20420;

C2H4NHV=20290;

C3H8NHV=19930;

C3H6NHV=19690;

C4H10NHV=19670;

C4H8NHV=19420;

C5H12NHV=19500;

H2SNHV=6550;

NNHV=0;

CPair=0.24;

Tt=471.146; %combustion air yemperature

Tf=78.8; %fuel temperature

Td=60;

CPfuel=0.53;

STO2=247.4004;

CO2=137.69;

SO2=0.0036;

H2O=221.8536;

mO2Used=39.58;

N2infuel=0;

% calculate Weight of each component (mole/hr)

% weight=molecular weight(i)\* volume fraction(i)

HW= 2\*HVF;

COW= 28\*COVF;

CO2W=44\*CO2CF;

CH4W=16\*CH4VF;

C2H6W= 30\*C2H6VF;

C2H4W= 28\*C2H4VF;

C3H8W= 44\*C3H8VF;

C3H6W= 42\*C3H6VF;

C4H10W= 58\*C4H10VF;

C4H8W= 56\*C4H8VF;

C5H12W= 72\*C5H12VF;

H2SW= 34\*H2SVF;

NW= 28\*NVF;

% Calculating heating value of each component

% heating value (i)=net heat value(i)\* weight(i)

HHV=HNV\*HW;

COHV=CONHV\*COW;

CO2HV=CO2W\*CO2W;

CH4HV=CH4NHV\*CH4W;

C2H6HV=C2H6NHV\*C2H6W;

C2H4HV=C2H4NHV\*C2H4W;

C3H8HV=C3H8NHV\*C3H8W;

C3H6HV=C3H6NHV\*C3H6W;

C4H10HV=C4H10NHV\*C4H10W;

C4H8HV=C4H8NHV\*C4H8W;

C5H12HV=C5H12NHV\*C5H12W;

H2SHV=H2SNHV\*H2SW;

NHV=NNHV\*NW;

H V = H H V + C O H V + C H 4 H V + C 2 H 6 H V + C 2 H 4 H V + C 3 H 8 H V + C 3 H 6 H V + C 4 H 1 0 H V + C 4 H 8 H V + C 5 H 1 2 H V + C O 2 H V + H 2 S H V + N H V ;

W=HW+COW+CH4W+C2H6W+C2H4W+C3H8W+C3H6W+C4H10W+C4H8W+C5H12W+H2SW+NW;

LHV=HV/W;

% calculate the % excess air supplied (EA)

EA=O\*100/(21-O);

ActualO2=STO2+(EA\*STO2); %Actual oxygen required

%calculate actual amount of oxygen

Mo2influegas=ActualO2-mO2Used;

ActualAir=ActualO2\*(100/21);

%calculate actual amount of Nitrogen in flue gas

MN2influegas=0.79\*(ActualAir)+N2infuel;

WCO2=44\*CO2; %weight/hr

WH2O=18\*H2O; %weight/hr

WSO2=64\*SO2; %weight/hr

WO2=32\*Mo2influegas; %weight/hr

WN2=28\*MN2influegas; %weight/hr

fuel=2280.7024; %lb/hr

```

hco2=124;
hh2o=205;
hso2=98;
ho2=124;
hn2=142;
%calculate weight of each stack component in feul
weightco2=WCO2/fuel;
weighth2o=WH2O/fuel;
weightso2=WSO2/fuel;
weightn2=WN2/fuel;
weighto2=WO2/fuel;
co2heatcontent=hco2*weightco2;
h2oheatcontent=hh2o*weighth2o;
so2heatcontent=hso2*weightso2;
o2heatcontent=ho2*weighto2;

n2heatcontent=hn2*weightn2;
% Output:
% Estimate all heat losses
Qr=0.02*LHV;
Qs=co2heatcontent+h2oheatcontent+so2heatcontent+o2heatcont-
ent+n2heatcontent;
Hf=CPfuel*(Tf-Td); % sensible heat correction for fuel (Hf)
Ha=(ActualAir/fuel) *CPair*(Tt-Td); %sensible heat correction
for combustion of air
fprintf('percentage heat losses due to dry flue gas %10.2e kJ/h\
n',Qs)

% calculate thermal efficiency by indirect method
E=((LHV+Ha+Hf)-Qs-Qr)/(LHV+Ha+Hf) *100;
fprintf('thermal efficiency %8.2f %%\n',E)

```

## APPENDIX 2

Simulation data:

**Table 1: Combustion work sheet**

Component of fuel	Vol. fraction mole/hr	Net heating value BTU/Lb
Methane	49.67	21500
Hydrogen	9.48	51600
Ethane	19.57	20420
Ethylene	3.12	20290
Propane	5.11	19930
Propylene	3.09	19690
Butane	1.93	19670
Butylene	0.92	19420
Pentane	0.84	19500
Nitrogen	1.38	0
Carbone monoxide	2.44	4345
Carbone dioxide	2.43	0
Hydrogen sulfide	0.0036	6550
Total	99.9836	

**Table 2: Combustion reaction**

Reaction	Moles of oxygen require
$H_2 + 0.5O_2 \rightarrow H_2O$	0.5
$CO + 0.5O_2 \rightarrow CO_2$	0.5
$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$	2
$C_2H_6 + 3.5O_2 \rightarrow 2CO_2 + 3H_2O$	3.5
$C_2H_4 + 3O_2 \rightarrow 2CO_2 + 2H_2O$	2
$C_3H_8 + 5O_2 \rightarrow 3CO_2 + 4H_2O$	5
$C_3H_6 + 4.5O_2 \rightarrow 3CO_2 + 3H_2O$	4.5
$C_4H_{10} + 6.5O_2 \rightarrow 4CO_2 + 5H_2O$	6.5

$C_4H_8+6O_2 \rightarrow 4CO_2+4H_2O$	6
$C_5H_{12}+8O_2 \rightarrow 5CO_2+6H_2O$	8
$C_6H_{14}+9.5O_2 \rightarrow 6CO_2+7H_2O$	9.5
$S+O_2 \rightarrow SO_2$	1
$H_2S+1.5O_2 \rightarrow SO_2+H_2O$	1.5

**Table 3: EDR data.**

<b>Process data</b>	
1-stream	
Total mass flow rate	125 T/Hr
Inlet temperature	270 C°
Out let temperature	320 C°
Inlet vapor mass fraction	3%
Out let vapor mass fraction	40%
<b>Flue gas: -</b>	
Inlet temperature to convection section	290 C°
Ambient temperature	33.1 C°
<b>Injection steam: -</b>	
Mass flow rate	360 kg/Hr
Pressure	2.5 Mpa
Temperature	420 C°
<b>Firebox: -</b>	
Fire heater type	Twin box
Tube row layout	Refractory baked
<b>Fire box dimension: -</b>	
Height	13700 mm
Length	18834 mm
Width	16530 mm
Evaluation of floor firebox	2550 mm
Evaluation of top fire box	13700 mm
<b>Burner details: -</b>	
Burner location	Bottom
Type of burner	Flat flame
No. of burner	98
Burner diameter	100 mm
<b>Main tube rows: -</b>	
Process steam in firebox	4
Tube passes	4
Evaluation of main tube in firebox	Horizontal
Tube straight length	18420 mm
Height of lowest tube above firebox	500 mm
Tube to wall clearance	200 mm
Tube –U bend location	Inside firebox

Tube lay out angle	U-shell
Flow direction in first tube	Up flow

<b>Tube location: -</b>				
<b>Tube location</b>	<b>Main</b>	<b>main</b>	<b>Main</b>	<b>Main</b>
No. of tube per pass	12	6	4	2
Tube material	316L	316L	316L	316L
Pipe schedule	80	80	80	80
Tube outside diameter(mm)	114.3	168.28	219.08	273.05
Tube wall thickness (mm)	8.56	10.97	12.7	15.09
Tube Spacing(mm)	203.2	304.8	406.4	508
<b>Gas of take: -</b>				
Flue gas off take width (mm)	2800			
Flue gas off take length (mm)	40430			
External diameter (m <sup>2</sup> )	3070			
<b>Convection bank: -</b>				
Process stream in bank	2	2		
Stream inflow form	Bank2	Inlet		
Stream out flow	Firebox	Bank1		
Tube No. used in bank	6	6		
Tube alienation in bank	Horizontal	Horizontal		
Flue gas flow direction	Up	Up		
Duct width(mm)	1553	1553		
Duct other side(mm)	1553	1553		
Tube length(mm)	15700	15700		
<b>Gas flow: -</b>				
Stack diameter at bottom	3070			
Stack diameter at top	2800			
Height to bottom of stack	14903			
Height to top of stack	55217			
Height of damper in stack	14903			



**Table 4: Inlet stream parameter**

Item	Unit	Design thermal load		
Calculated thermal load	Kw	4755	682	25754
Name		Crude	steam	Heater feed
Flow	Kg/hr	123010	9650	162294
Inlet pressure	Mpa	0.9	1.25	2.85
Outlet pressure	Mpa	0.57	1.15	0.65
Inlet temperature	C°	270	191	366
Outlet temperature	C°	320	300	500

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