Fired Heater Simulation, Modelling and Optimization

Taj Alasfia M brakat

University of khartoum

Mahmoud Adam Hassan (✉ mahmoud.adam1@outlook.sa)

University of khartoum

Research Article

Keywords: Efficiency, heat transfer, flame temperature, firebox

Posted Date: September 1st, 2021

DOI: <https://doi.org/10.21203/rs.3.rs-847461/v1>

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Fired Heater Simulation,Modelling and Optimization,

Authors:

Mahmoud Adam Hassan

Taj alasfia M barakat

Abstract

In this study a dynamic model Simulation was carried out using ASPEN HYSYS for industrial refinery fired heater, PID controller was applied to control the flue gas exit temperature. the simulation shows perfect agreement with the datasheet of the furnace. 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, hence the duty furnace has increased from 65.9MW to 68.1MW and the fuel flow in like manner 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, the code was designed to be flexible as possible, user will just need to replace the nominal fuel and air characteristics. for gaining optimum furnace operation the heat transfer inside the furnace was studied, and the results were compared with previous research involving furnace analysis to validate models, the heat transfer coefficient is determined by analysis of conductive heat transfer through the boundary layer. 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(EDR) alongside ASPEN HYSYS

Keywords: Efficiency, heat transfer, flame temperature, firebox

Scope of this study: -

The scope of this project is to give detailed study for the furnace in delayed coking unit (DCU)

Introduction

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.¹

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.

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:

$$Q = -k2\pi L(r - y)\frac{d\theta}{dy} \quad (7)$$

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:

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

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

And then:

$$\left(\frac{d^2\theta}{dy^2}\right)_{y=0} = r^{-1} \left(\frac{d\theta}{dy}\right)_{y=0} \quad (8)$$

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:

$$\theta = a_0 y + b_0 y^2 + c_0 y^3 \quad (9)$$

Thus:

$$\frac{d\theta}{dy} = a_0 + 2b_0 y + 3c_0 y^2 \quad \text{and} \quad \left(\frac{d\theta}{dy}\right)_{y=0} = a_0$$

$$\frac{d^2\theta}{dy^2} = 2b_0 + 6c_0 y \quad \text{and} \quad \left(\frac{d^2\theta}{dy^2}\right)_{y=0} = 2b_0$$

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

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

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

And then:

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

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

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

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 θ_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 θ_B 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:

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

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

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

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

Substituting from equations 15 and 16 in equation 12:

$$\theta_B = \frac{\frac{51}{350} r^2 u_s \theta_s}{\frac{1}{4} r^2 u_s}$$

$$= \frac{102}{175} \theta_s = 0.583 \theta_s$$

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.

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

Or $Nu=4.1^3$

3.3 HEAT TRANSFER BY RADIATION

It follows the equation below

$$Q = (SG_l)_R \sigma (T_g^4 - T_l^4)$$

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.⁴

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

$$\text{Efficiency} = \frac{\text{heat usefully absorbed}}{\text{total heat supplied}} * 100 \quad (1)$$

Furnace efficiency: -

There are two methods to calculate efficiency: -

1- Direct method: -

$$e = \frac{\text{heat absorb by crude} + \text{heat absorb by steam}}{\text{fuel compustion} * \text{calorific value}} * 100\% \quad (3)$$

Indirect method: -

$$e = \frac{(LHV+H_a+H_f)-Q_s-Q_r}{(LHV+H_a+H_f)} * 100 \% \quad (4)$$

This research focus on indirect method as a source of high accuracy because all losses (stack losses, radiation losses) are taken into account.

$$\text{Weight} = (\text{volume fraction} * \text{molecular weight}). \quad (5)$$

$$\text{Heating value} = (\text{net. heating value} * \text{weight}) \quad (6)$$

$$\begin{aligned} \text{Stoichiometric oxygen} \\ &= (\text{oxygen from combustion reaction} \\ &\quad * \text{volume fraction}) \end{aligned} \quad (7) \quad .$$

$$\begin{aligned} \text{—Actual oxygen required} \\ &= \text{Stoichiometric oxygen} + (\text{excess air} * \text{Stoichiometric oxygen}) \end{aligned} \quad (8)$$

$$\text{— Actual air required} = \text{actual oxygen required} * \left(\frac{100}{21}\right) \quad (9)$$

- Estimate stack component (CO_2 , H_2O , SO_2 , O_2 , N_2).

$$\text{—Amount of } \text{CO}_2 \text{ in flue gas} = (\text{total formed} + \text{CO}_2 \text{ reported as fuel}) \quad (10)$$

$$\text{—Amount of } \text{H}_2\text{O} \text{ in flue gas} = (\text{total formed} + \text{H}_2\text{O reported as fuel}) \quad (11)$$

$$\text{—Amount of } \text{SO}_2 \text{ in flue gas} = (\text{total formed} + \text{SO}_2 \text{ reported as fuel}) \quad (12)$$

$$\begin{aligned} \text{—Amount of } \text{O}_2 \text{ in flue gas} \\ &= (\text{actual } \text{O}_2 \text{ supplied} - \text{actual } \text{O}_2 \text{ used during combustion}) \end{aligned} \quad (13)$$

$$\text{—Amount of } \text{N}_2 \text{ in flue gas} = (79\% \text{ of moles of air} + \text{N}_2 \text{ reported as fuel}) \quad (14)$$

3.1. Lower heating value: -

$$\text{LHV} = \frac{\Sigma \text{ heating value}}{\Sigma \text{ weight of component of fuel}} \quad (15)$$

3.2. stack losses: -

- 1- (amount of stack component * molecular weight) (16)
- 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).

3.3. 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).

3.4. sensible heat correction for combustion of air (H_a): -

$$H_a = \text{lb of air} / \text{lb of fuel} * C_{p_{\text{air}}} * (T_t - T_d) \quad (17)$$

T_t = combustion air temperature.

T_d = datum temperature (60°F).

3.5. sensible heat correction for fuel (H_f): -

$$H_f = C_{p_{\text{fuel}}} * (T_t - T_d) \quad (18)$$

3.5 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.²

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

3.51 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_g . This requires:

- (a) Furnace geometry and surface properties $(SG_1)_R$;
- (b) Gas composition (usually known for a specific fuel and combustion air);
- (c) gas flame ϵ_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.⁵

the net heat exchange, Q ,

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

If the flux geometry is calculated from:

$$(SG_1)_R = \frac{A_T}{\left(\left[\frac{1}{\epsilon_g} \right] + \left[\frac{1}{\{C_s \epsilon_1\}} \right] - 1 \right)} \quad (20)$$

and energy balance could be written as:

$$\frac{H_f - Q}{H_f} = \frac{T_g - \Delta - T_o}{T_{af} - T_o} \quad (21)$$

From equation (19) we get:

$$\frac{Q}{(SG_1)_R \sigma} + T_1^4 = (T_g^4) \quad (22)$$

eliminating T_g from equation (21):

$$\left(1 - \frac{Q}{H_f}\right)(T_{af} - T_o) + \Delta + T_o = T_g \quad (23)$$

$$\left[T_{af} - T_o - \frac{Q}{H_f}(T_{af} - T_o) + \Delta + T_o\right]^4 = \frac{Q}{(SG_1)_R \sigma} + T_1^4 \quad (24)$$

Dividing equation (24) by T_{af}^4 and arranging:

$$\frac{Q}{\sigma(GS_1)_R} + T_1^4 = T_{af}^4 \left[1 + \frac{\Delta}{T_{af}} - \frac{Q(T_{af} - T_o)}{H_f T_{af}}\right]^4 \quad (25)$$

following dimensionless groups are simplified equation (25)

$$Q' = \frac{Q(T_{af} - T_o)}{H_f T_{af}} \quad D' = \frac{H_f}{\sigma(GS_1)_R T_{af}^3 (T_{af} - T_o)} \quad \Delta' = \frac{\Delta}{T_{af}} \quad \tau = \frac{T_1}{T_{af}} \quad (26)$$

And then:

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

where d is a constant of proportionality, thus we have

$$Q' D' + \tau^4 = \left(1 - \frac{Q'}{d}\right)^4 \quad (27)$$

For convective heat transfer, and for radiation losses from furnace openings, thus equation (19) can be rewritten as

$$Q = (SG_1)_R \sigma (T_g^4 - T_1^4) - UA_g (T_g - T_o) + (SG_0)_R \sigma (T_g^4 - T_o^4) \quad (28)$$

This can be calculated from flux geometry as

$$(SG_0)_R = \frac{A_T}{\left(\left[\frac{1}{\varepsilon_g} \right] + \left[\frac{1}{\{C_0 \varepsilon_{01}\}} \right] - 1 \right)}^{10} \quad (29)$$

To eliminate T g 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:

$$Q'D' = \left(1 - \frac{Q'}{d} \right)^4 - \tau_o^4 + L'_R \left[\left(1 - \frac{Q'}{d} \right)^4 - \tau_o^4 \right] + L'_o \left[\left(1 - \frac{Q'}{d} \right)^4 - \tau_o^4 \right] \quad (30)$$

the furnace efficiency, η , is then calculated from

$$\eta = \frac{Q - Losses}{H_f} = \frac{Q'}{1 - \tau_o} - \frac{L'_R}{D'} \left[1 - \frac{Q'}{1 - \tau_o} \right] - \frac{L'_o}{D'(1 - \tau_o)} \left[\left(1 - \frac{Q'}{d} \right)^4 - \tau_o^4 \right] \quad (31)$$

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

3.6 Simulation:

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

figure (6): well-stirred model validation

Connections Parameters Monitor Stripchart User Variables Notes

Name FH

Process Variable Source

Object: FH-100

Select PV...

Variable: Burner Flame Temperature

Remote Setpoint

Select RSP...

Optional

OptimizerSpreadsheet

Spreadsheet Cell

Output Target Object

Object: crude

Select OP...

Variable: Control Valve (control flows)

OK

Delete

Face Plate...

Control Valve...

Figure (8):setting up PID controller

Figure: edr fired heater

3.61 Case study:

Studying different effects of parameters to the furnace by inserting conditions

Running case study

The screenshot displays the AspenONE Exchange software interface, specifically the Case Study 1 setup and execution window. The interface is divided into several sections:

- Top Menu Bar:** Includes File, Home, Economics, Dynamics, View, Customize, and Resources.
- Left Sidebar:** Contains a tree view of the project structure, including Workbook, UnitOps, Streams, Stream Analysis, Equipment Design, Model Analysis, Data Tables, Strip Charts, Case Studies, Case Study 1, and Data Fits.
- Main Workspace:** Displays the Case Study 1 setup and execution details.

Case Study 1 Setup:

- State Input Type:** Nested
- Number of States:** 5
- Reset after Run:** ☐
- Step Downward:** ☐
- Independent Variable Table:**

Independent Variable	Low Bound	High Bound	Step Size
COKER-100 - Furnace Temperature	200.0	1000	200.0

Case Study 1 Execution:

- Stop:**
- Running State - 1:**

Case Studies Setup - Case Study 1:

- Set up:** Results, Plots, Failed States
- Object Table:**

Object	Variable	Independent	Include
FH-100	Combustion Efficiency	Yes	☒
FH-100	Fired Heater Calculation Enum	Yes	☒
FH-100	Burner Flame Temperature	No	☒
FH-100	Performance Table (Performance Table_2-Hot oil: 0)	No	☒
FH-100	Stack (exit) Temperature	No	☒

Case Study 1 Execution:

- State Input Type:** Nested
- Reset after Run:** ☐
- Step Downward:** ☐
- Number of States:** 110
- Independent Variable Table:**

Independent Variable	Low Bound	High Bound	Step Size
FH-100 - Combustion Efficiency	10.00	100.00	10.00
FH-100 - Fired Heater Calculation Enum	500	1500	100.0

Case Study 1 Execution:

- Run:**
- Success:**

Figure (9): furnace simulation

Figure: Case study

4. Results:

It was found that increasing the flame temperature has a direct effect on the thermal efficiency as shown in the figure below:

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

Figure(14): Effect of O_2 ratio

Effect of O₂ in inlet air on furnace thermal efficiency has been induced 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

Figure(16): efficiency calculation (1)

Figure(17): efficiency calculation (2)

From the above calculation the efficiency can be raised to reach 83%

Cut

Set Units

Copy

Paste

Clipboard

SI

Convert

Next

Set Process Data

Set Properties

Set Combustion

Set Geometry

Run

Stop

Run Status

Run Control

Simulation (fixed fuel flowrate)

Rating (find fuel flowrate)

Check Performance

Review Profiles

Air Cooled

Fired Heater

Plate

Plate Fin

Shell & Tube

Mechanical

Model Type

EDR Navigator

All

Fired Heater

Input

Problem Definition

Physical Property Data

Fuel + Oxidant

Heater Geometry

Firebox

Convection Banks

Gas Flow

Program Options

Results

Results Summary

Thermal / Hydraulic Summary

Calculation Details

Convection Banks

Fired Heater

Layout

Tubes

Fins + Studs

Bundle Details

Connections Diagram

Number of convection banks:

3

	Tube 1	Tube 2	Tube 3	Tube 4	Tube 5	Tube 6	Tube 7	Tube
Tube surface enhancement type	plain	solid fin	solid fin					

	Bank 1	Bank 2	Bank 3	Bank 4	Bank 5	Bank 6	Bank 7
Process stream in bank	1	1	2				
Stream inflow from	bank 2	inlet	inlet				
Stream outflow to	firebox	bank 1	outlet				
Tube number used in bank		1	2	3			
Tube orientation in bank	Horizontal	Horizontal	Horizontal				
Flue gas flow direction	up	up	up				
Tube length	mm	17400	17400	17400			
Stream overall flow direction	Countercurrent	Countercurrent	Countercurrent				
Flow direction in first tube							
Duct width (transverse to tubes)	mm	1739.9	1739.9	1943.1			
Duct other side (parallel to tubes)	mm	17400	17400	17400			

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 ³	43.2
Firebox well-stirred gas temperature	°C	917
Bridgwall 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

Edr result

. It was founded that Increasing the number of finned tube rows in convection bank (from 2 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

Increasing the fraction of the oxygen in the inlet air will lead to increase in thermal efficiency based on the fact that nitrogen absorb heat hence decrease thermal efficiency.

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
$\text{H}_2 + 0.5\text{O}_2 \rightarrow \text{H}_2\text{O}$	0.5
$\text{CO} + 0.5\text{O}_2 \rightarrow \text{CO}_2$	0.5
$\text{CH}_4 + 2\text{O}_2 \rightarrow \text{CO}_2 + 2\text{H}_2\text{O}$	2
$\text{C}_2\text{H}_6 + 3.5\text{O}_2 \rightarrow 2\text{CO}_2 + 3\text{H}_2\text{O}$	3.5
$\text{C}_2\text{H}_4 + 3\text{O}_2 \rightarrow 2\text{CO}_2 + 2\text{H}_2\text{O}$	2
$\text{C}_3\text{H}_8 + 5\text{O}_2 \rightarrow 3\text{CO}_2 + 4\text{H}_2\text{O}$	5
$\text{C}_3\text{H}_6 + 4.5\text{O}_2 \rightarrow 3\text{CO}_2 + 3\text{H}_2\text{O}$	4.5
$\text{C}_4\text{H}_{10} + 6.5\text{O}_2 \rightarrow 4\text{CO}_2 + 5\text{H}_2\text{O}$	6.5
$\text{C}_4\text{H}_8 + 6\text{O}_2 \rightarrow 4\text{CO}_2 + 4\text{H}_2\text{O}$	6
$\text{C}_5\text{H}_{12} + 8\text{O}_2 \rightarrow 5\text{CO}_2 + 6\text{H}_2\text{O}$	8
$\text{C}_6\text{H}_{14} + 9.5\text{O}_2 \rightarrow 6\text{CO}_2 + 7\text{H}_2\text{O}$	9.5
$\text{S} + \text{O}_2 \rightarrow \text{SO}_2$	1
$\text{H}_2\text{S} + 1.5\text{O}_2 \rightarrow \text{SO}_2 + \text{H}_2\text{O}$	1.5

Table 3. EDR data¹¹.

Process data:-	
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%
Flue gas:-	
Inlet temperature to convection section	290 C°
Ambient temperature	33.1 C°
Injection steam:-	
Mass flow rate	360 kg/Hr
Pressure	2.5 Mpa
Temperature	420 C°
Firebox:-	
Fire heater type	Twin box
Tube row layout	Refractory baked
Fire box dimension:-	
Height	13700 mm
Length	18834 mm
Width	16530 mm
Evaluation of floor firebox	2550 mm
Evaluation of top fire box	13700 mm
Burner details:-	
Burner location	Bottom
Type of burner	Flat flame
No. of burner	98
Burner diameter	100 mm
Main tube rows :-	
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		
Tube location:-				
Tube location	Main	main	Main	Main
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
Gas of take:-				
Flue gas off take width (mm)		2800		
Flue gas off take length (mm)		40430		
External diameter (m ²)		3070		
Convection bank:-				
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
Gas flow:-		
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

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

Δ = temperature difference

ϵ =emissivity, rate of energy dissipation

η = thermal efficiency

μ = viscosity – cannot find this in the m/s

ρ = density

σ =Stefan-Boltzmann constant

σ_k =diffusion constant

σ_ϵ =dissipation constant

τ =temperature ratio

Dimensionless groups

Re _ Reynolds number = $ux\rho/\mu$

Sc _ Schmidt number = $\mu/\rho x$

APPENDIX 1

```
function indirectmethod=indirectefficiency
% 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
HNHV=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=HNHV*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;
HV=HHV+COHV+CH4HV+C2H6HV+C2H4HV+C3H8HV+C3H6HV+C4H10HV+C4H8HV+C5H12HV+CO2H
V+H2SHV+NHV;
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+o2heatcontent+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)
```

References

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Figures

Figure 1

Flue gas temperature

Figure 2

Fired heater simulation

Figure 3

Fired heater