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Zonal Analysis of Radiant Chambers in Direct Fired-Heaters

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***Abstract*— Fired Heaters are easily the most energy-extensive units in process industry. Often times, the operating cost of Fired Heaters can amount to almost 20-50% of a total plant's operating cost. Monitoring the efficiency of these heaters and continuous improvement in design are critical. Significant advancements have been made in simulating the temperature distribution inside Fired-Heaters. This paper focuses on simulating the temperature distribution inside the Radiant Section of a Fired-Heater using the Long-Furnace model; a more accurate extension of the Lobo-Evans method widely accepted in industry. A novel mathematical model is developed and is being construed for application to a Fired-Heater in a Syrian Refinery. The model is potentially more accurate than the Lobo-Evans method and computationally less extensive than a CFD approach.**

***Keywords*— Zonal Analysis, Long Furnace Model, Fired Heaters, Radiant Chamber, Temperature Distribution**

I. INTRODUCTION

Fired Process Heaters are complex unit-operations used in many industrial settings. Their typical functionality is to transfer heat to process streams. The type of heat applied can vary (sensible or latent), depending on the requirements of the specific process.

Of particular interest to this study are Fired Process Heaters, which only apply sensible heat to process streams. These heaters have tubes mounted inside the furnace, through which the process fluid flows. The combustion of supplied fuel results in hot flue gases, which flow counter-current to the flow of the process fluid. Heat is then transferred from the flue gases to the tubes containing the process stream.

In the modern refineries, fired heaters are typically constructed with two distinct heating sections. There is a radiant section; where the dominant mode of heat transfer is radiation, followed by the convection section where the dominant mode of heat transfer is convection. The convection section is often labelled as the "economizer", as it tends to maximize heat transfer from residual flue gas. Fired Heaters were previously constructed with just a radiant-section or a convection-section. The combination of these two sections allows for a more economically practical fired heater (Jegla, 2006).

Figure 1 depicts various furnace orientations that commonly exist. The two most common shapes for fired process heaters are cabin-type (1a) and cylindrical (1c). The cabin-type process heater has a box-like structure and is mostly used for heavy heat duty applications. The cylindrical furnaces are used for smaller-scale heat duty applications. As the name suggests, cylinder furnaces have cylindrical body in both wall furnace. In this study, we will focus primarily on the cabin-type orientation. Specifically, the focus is on a heater orientation with a single radiant chamber. Figure 2 depicts the internals of the box-type heater in more detail.

II. OBJECTIVE

Lobo-Evans method has been utilized for radiant section calculation for the past couple of decades. This method assumes constant conditions throughout the radiant chamber and treats it as one big zone. The simple objective of this paper is to challenge these assumptions. For radiant chambers with drastic changes in temperature, the properties of the flue gas and process fluid are likely to vary. This causes major inaccuracies in results obtained from the Lobo-Evans method.

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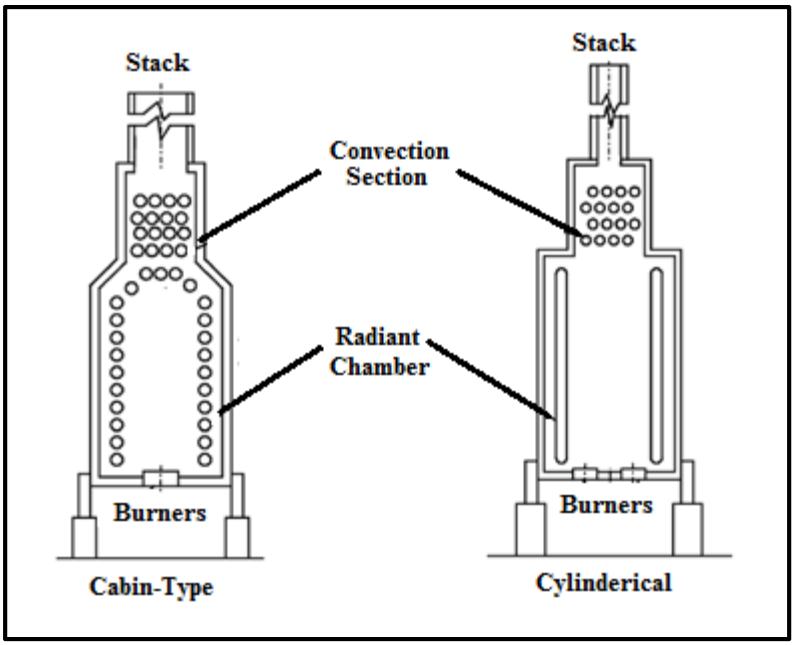

Fig 1. Typical industrial configurations of Fired Process Heaters. 1a) is a box type furnace with one radiant chamber. 1b) is a vertical cylindrical furnace with one radiant chamber. Each of the depicted furnaces has one convection section (economizer).

In this study, the radiant chamber is to be split up into distinct zones, and a solution to energy balance along each zone will provide a temperature distribution along the length of the radiant section (Figure 2). Each zone is comprised of three things: a flue gas moving towards the right (g), load or process fluid to be heated moving counter-currently to the flue gas in tubes (l) and a wall (w). The model will provide temperature distributions for the flue gas, process fluid and the wall.

The long furnace model in consideration assumes that there is no radiant flux between the surfaces and volume zones in different zones. This will greatly reduce computational requirements. Plug flow and insignificant pressure drop are also assumed. The rest of this section focuses on derivation of the generic mathematical model that can help simulate the 1D temperature fields.

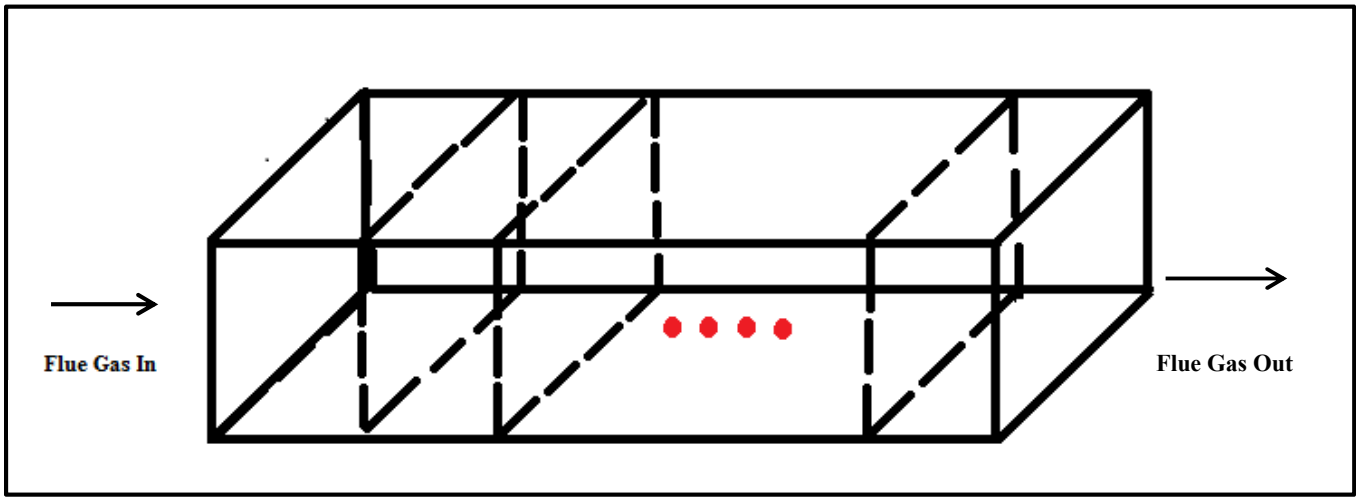

Fig 2. Discrete Zone formulations to simulate temperature distribution inside a typical radiant chamber.

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III. DERIVATION OF MATHEMATICAL MODEL

Radiation Transfer Equation (RTE)

$$(\hat{n}\nabla)\cdot I_\lambda = -\left(K_{a,\lambda}+K_{s,\lambda}\right)I_\lambda + K_{a,\lambda}I_{b,\lambda} + \frac{K_{s,\lambda}}{4\pi}\int_0^{4\pi} I_\lambda\, dw \tag{1}$$

Solution of the above general radiative equation can be obtained using Flux Models, Monte-Carlo Technique or Zonal Method. The solution would provide the intensity of a monochromatic beam radiating in a control volume in a specified direction. The main focus of this study is to implement a specific Zonal method, called the Long-Furnace Model (Type I) to predict temperature distribution in a radiant chamber (Hottel, 1967). The long furnace model is essentially a discretized solution of (1). If the number of zones was increased to infinity within the zonal model, (1) above would re-postulate itself.

A tubular direct-fired heater is the specific geometry of interest, with a cabin-type chamber configuration. This implies that the radiant compartment is of rectangular geometry but the process fluid to be heated travels inside tubes, confined by the chamber.

Derivation of the Long-Furnace Model (Type I)

The net energy balance through the flue gases in ith zone yields:

$$\boldsymbol{\dot{Q}_{G,net_i} + \dot{Q}_{a_i} + \dot{Q}_{enth_i} - \dot{Q}_{conv_i} - \dot{Q}_{Rad_i} = 0} \tag{2}$$

Deriving expressions for the first three terms

The amount energy transferred through combustion of the fuel can be quantified using:

$$\boldsymbol{\dot{Q}_{G,net_i} + \dot{Q}_{a_i}} = \dot{V}_{fuel}\left[GCV_i + R_s\left[1+\frac{X}{100}\right]\cdot\rho^o_{air}\cdot H_{air}\left(T_{g_i}\right)\right] \tag{3}$$

The cumulative amount of energy released from combustion across the length of the chamber can be computed using:

$$Y(x) = 1 - \exp\left(-4.6\frac{x}{L_f}\right) \tag{4}$$

The flame length can be calculated using:

$$L_f = 6.65\frac{\dot{m}_o}{\sqrt{\dot{G}_o\cdot\rho_f}}\left(1+\frac{R_s\cdot\rho_f}{\rho_G}\right) \tag{5}$$

Equation (4) above can provide information about the Gross Calorific Value in each zone through:

$$GCV_i = [Y(x_i) - Y(x_{i-1})]\cdot CV_{net} \tag{6}$$

The enthalpy term is easily computed as (neglected the insertion of heat capacities for now):

$$\boldsymbol{\dot{Q}_{enth_i}} = m^+_{i-1}\cdot H_g\left(T_{g,i-1}\right) + m^-_{i+1}\cdot H_g\left(T_{g,i+1}\right) - [m^-_i + m^+_i]\cdot H_g\left(T_{g_i}\right) \tag{7}$$

As the model includes the potential of any recirculating eddies due to turbulent nature of the flow, the mass balance is as below:

$$m^+_{i-1} + m^-_{i+1} - m^-_i - m^+_i = 0 \tag{8}$$

Deriving expressions for the Convection Term

Convection between the load and the wall yields (assuming thin tubes):

$$\dot{\boldsymbol{Q}}_{\boldsymbol{conv_i}} = A_L h_L \left(T_{g_i} - T_{L_i}\right) + A_w h_w \left(T_{g_i} - T_{w_i}\right) \tag{9}$$

Deriving expressions for the Radiation Term

Radiation occurs between the gas zone, load zone and the wall zone. This can be summarize mathematical using:

$$\dot{\boldsymbol{Q}}_{\boldsymbol{Rad_i}} = \left[\overrightarrow{GS_L} + \overrightarrow{GS_W}\right] \cdot \sigma T_{g_i}^4 - \overrightarrow{GS_L} \cdot \sigma T_{L_i}^4 - \overrightarrow{GS_w} \cdot \sigma T_{w_i}^4 \tag{10}$$

The symbols with arrows on top represent directed flux areas and their calculations will be explained later on

Substituting all the equations above into (2) will provide an equation which is a function of the following unknowns:

- Flue Gas Temperature **($\mathbf{T_{g,i}}$)**
- Wall Temperature **($\mathbf{T_{w,i}}$)**
- Load Temperature **($\mathbf{T_{L,i}}$)**

Hence, in addition to the volume zone energy balance, two additional equations are required to solve for the temperatures. These are derived from independent energy balances on the surface of the wall and the load.

Energy Balance on the Load

$$\dot{Q}_{L_i} = \overrightarrow{GS_L}\left(\sigma T_{g_i}^4\right) + \overrightarrow{S_L S_w}\left(\sigma T_{w_i}^4\right) - A_L \varepsilon_L \left(\sigma T_{L_i}^4\right) + A_L h_L \left(T_{g_i} - T_{L_i}\right) \tag{11}$$

Energy Balance on the Wall

$$\dot{Q}_{W_i} = \overrightarrow{GS_W}\left(\sigma T_{g_i}^4\right) + \overrightarrow{S_L S_w}\left(\sigma T_{L_i}^4\right) - A_W \varepsilon_W \left(\sigma T_{W_i}^4\right) + A_W h_W \left(T_{g_i} - T_{W_i}\right) \tag{12}$$

Further, QL and Qw must be written in terms of the unknown temperatures, in order to simultaneously solve the 3 energy balances for the unknown temperatures.

Writing Q_L and Q_w as functions of volume and surface zone Temperatures

The unknowns on the left-hand side must be written as a function of temperature. This is done by conducting a simple enthalpy balance on the load and computing a wall heat loss equation that can be solved iteratively/

Load Heat Transfer

$$\dot{Q}_{L_i} = \dot{m}_L \cdot \left(H_L\left(T_{L_{out}}\right) - H_L\left(T_{L_{in}}\right)\right) \tag{13}$$

$$T_{L_i} = \frac{T_{L_{out}} + T_{L_{in}}}{2} \tag{14}$$

Wall Heat Loss

$$Q_{w_i} = A \cdot \frac{(T_2 - T_o)}{\sum \frac{\lambda_i}{x_i}} \tag{15}$$

$$Q_{w_i} = A\varepsilon\sigma\left(T_o^4 - T_\infty^4\right) + Ah_o(T_o - T_\infty) \tag{16}$$

$$h_o' = \frac{\varepsilon\sigma\left(T_o^4 - T_\infty^4\right) + h_o(T_o - T_\infty)}{(T_o - T_\infty)} \tag{17}$$

$$Q_{w_i} = A \cdot \frac{(T_2 - T_\infty)}{\left(\sum \frac{\lambda_i}{x_i}\right) + \frac{1}{h_o'}} \tag{18}$$

The following algorithm explains how the combination of equations (15-18) above and the zonal mathematical model can be used to solve for T_0.

1. Assume T_0
2. Use Equation (17) to calculate h_0'
3. Use Equation (18) to write Q_w as a function of $T_{w,i}$
4. Use $Q_w(T_{wi})$ in the mathematical model to solve for Q_w and $T_{w,i}$
5. Substitute Q_w into Equation (16) and solve for new T_0
6. Repeat 1-5 until convergence is achieved – i.e. assumed T_0 is within to the T_0 obtained from step 5

Direct Exchange Areas (J.M Rhine, 1991)

A pre-requisite for solving the equations above is to calculate the directed-surface fluxes which are used heavily from Equations (9) to (12). Directed-Surface Flux, as defined by Jegla, are weighted averages of direct exchange areas. The latter are essentially the relative area between two surfaces in direct visibility of each other, and consequently involved in mutual radiative heat transfer.

The following assumptions greatly simplify the direct and total exchange areas that need to be calculated:

- The load is unable to see itself in each zone, simply due to the fact that it is a generally flat or convex surface
- All points on the wall have an equal view of the load
- Radiation attenuation and emission by Combustion products is calculated using the mean beam length (Hottel, 1967):

$$\boldsymbol{L_M = 3.5\frac{V}{A}} \tag{19}$$

These assumptions hold true for vertical cylindrical furnaces, where a pair of concentric cylinders exist.

The direct-exchange areas are simply:

$$\begin{gathered}\overline{S_L S_L} = 0 \\ \overline{S_L S_w} = A_L\tau \\ \overline{S_w S_L} = A_L\tau \\ \overline{S_w S_w} = (A_w - A_L)\cdot\tau\end{gathered} \tag{20 a,b,c,d}$$

Total Exchange Areas

For the two grey gas components, and a cabin-type geometry, the total exchange area becomes:

$$\overline{S_L S_w} = \overline{S_w S_L} = \frac{A_L\varepsilon_L\varepsilon_w}{\left(\frac{A_L}{A_w}\right)\rho_w(1-\rho_L\tau) + \frac{1}{\tau} - \rho_w} \tag{21}$$

$$\overline{GS_L} = \frac{A_L\varepsilon_L\left(\left(\frac{A_L}{A_w}\right)\rho_w(1-\tau) + \frac{1}{\tau} - 1\right)}{\left(\frac{A_L}{A_w}\right)\rho_w(1-\rho_L\tau) + \frac{1}{\tau} - \rho_w} \tag{22}$$

$$\overline{GS_w} = \frac{A_w \varepsilon_w \left(\left(\frac{A_L}{A_w}\right) \rho_L (1-\tau) + \frac{1}{\tau} - 1 \right)}{\left(\frac{A_L}{A_w}\right) \rho_w (1-\rho_L \tau) + \frac{1}{\tau} - \rho_w} \tag{23}$$

For the clear gas component ($\tau = 1$):

$$\overline{S_L S_w} = \frac{A_L \varepsilon_L \varepsilon_w}{\left(\frac{A_L}{A_w}\right) \rho_w (1-\rho_L) + 1 - \rho_w} \tag{24}$$

Calculation of Transmissivity

Clear Gas Component

$$\tau = 1 \; (k_{g,1} = 0) \tag{25}$$

Grey Gas Component

$$\tau = \exp\left(-k_{g,2}\left(P_{H_2O} + P_{CO_2}\right) \cdot L_M\right) = \exp\left(-1.88\left(P_{H_2O} + P_{CO_2}\right) \cdot L_M\right) \tag{26}$$

$$\tau = \exp\left(-k_{g,3}\left(P_{H_2O} + P_{CO_2}\right) \cdot L_M\right) = \exp\left(-68.8\left(P_{H_2O} + P_{CO_2}\right) \cdot L_M\right) \tag{27}$$

The pressure drop along the length of the radiant chamber can be neglected.

Calculation of Directed Flux Areas (J.M Rhine, 1991)

Surface-Surface Directed Flux

$$\overrightarrow{S_L S_w} = a_1(T_L) \cdot \overline{S_L S_W}(K_1) + a_2(T_L) \cdot \overline{S_L S_w}(K_2) + a_3(T_L) \cdot \overline{S_L S_w}(K_3) \tag{28}$$

$$\overleftarrow{S_L S_w} = a_1(T_w) \cdot \overline{S_L S_W}(K_1) + a_2(T_w) \cdot \overline{S_L S_w}(K_2) + a_3(T_w) \cdot \overline{S_L S_w}(K_3) \tag{29}$$

Note that $k_{s,i}$ for the clear gas does not equal 0 in this model.

Gas-Surface Directed Flux

$$\overrightarrow{GS_L} = a_2\left(T_g\right) \cdot \overline{GS_L}(K_2) + a_3\left(T_g\right) \cdot \overline{GS_L}(K_3) \tag{30}$$

$$\overrightarrow{GS_w} = a_2\left(T_g\right) \cdot \overline{GS_w}(K_2) + a_3\left(T_g\right) \cdot \overline{GS_w}(K_3) \tag{31}$$

$$\overleftarrow{GS_L} = a_2(T_L) \cdot \overline{GS_L}(K_2) + a_3(T_L) \cdot \overline{GS_L}(K_3) \tag{32}$$

$$\overleftarrow{GS_w} = a_2(T_w) \cdot \overline{GS_w}(K_2) + a_3(T_w) \cdot \overline{GS_w}(K_3) \tag{33}$$

The constants in the equations above can be computed using the empirical correlations as below:

$$a_i(T_i) = b_{1,i} + b_{2,i}(T_i) - for\ i = 1, 2\ and\ 3 \tag{34}$$

$$K_i = k_{g,i}\left(P_{H_2O} + P_{CO_2}\right) \tag{35}$$

Note that $k_{g,i}$ for the clear gas is equal to 0 in this model.

Also, assuming complete combustion, the reaction stoichiometry is:

$$CH_4 + 2 \cdot O_2 \rightarrow CO_2 + 2 \cdot H_2O \tag{36}$$

Constant Values for Grey-Gas Model

TABLE I. Constants for computation of a (equation 34)

n	$\mathbf{b_{1,n}}$	$\mathbf{b_{2,n}}$	$\mathbf{k_{g,n}}$
1	0.437	0.0000713	0
2	0.39	-0.0000052	1.88
3	0.173	-0.0000661	68.8

Convective Heat Transfer Coefficients

An excellent paper from Monrad approximates the convective heat transfer coefficient in a cabin-type radiant chamber to be:

$$h_L = 0.018 \cdot C_{p_{flue}} \frac{G_{max}^{2/3} \cdot T_{avg}^{0.3}}{D_o^{1/3}} \tag{37}$$

For approximation purposes, it can be assumed that the heat transfer to the refractory walls can be modelled as:

$$h_w = 0.1 \cdot h_L \tag{38}$$

The emissivity of the wall and the load can be obtained from empirical information.

The mathematical model is now completely defined and can be programmed on MatLab or Python for an explicit temperature distribution.

IV. Conclusion

In conclusion, Zonal Analysis of Radiant Chambers can allow for a 1-dimensional simulation of the temperature distribution. This analysis is computationally more extensive than the Lobo-Evans method, but has the potential to provide more accurate results for the total heat transfer inside the radiant chamber. This is because it accounts for varying conditions inside the radiant chamber. We are currently working on applying this model using data from an operating fired-heater and comparing results between the derived mathematical model and the Lobo-Evans Method. A complete version of this paper will provide an applied comparison of these approaches in the future.

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VI. NOMENCLATURE

Subscripts
$g - gas$
$l - load$
$w - wall$

Parameters
$A - Surface\ Area$
$C_p(T) = Heat\ Capacity\ Correlation$
$\varepsilon_i = Emissivity$
$\dot{G}_o = Mass\ Velocity\ at\ Minimum\ Cross - Section$
$GCV_i - Gross\ Calorific\ Value$
$\overrightarrow{GS_L} - Directed\ Flux\ Area\ from\ Gas\ Zone\ to\ Load\ Surface\ Zone$
$\overrightarrow{GS_W} - Directed\ Flux\ Area\ from\ Gas\ Zone\ to\ Wall\ Surface\ Zone$
$H_i - Enthalpy$
$h = Convective\ Heat\ Transfer\ Coefficient$
$I_\lambda - Intensity\ of\ a\ specified\ monochromatic\ beam$
$K_{a,\lambda} - Absorption\ Coefficient$
$K_{s,\lambda} - Scattering\ Coefficient$
$L_f - Flame\ Length$
$L_M - Mean\ Beam\ Length$
$m_i - Mass\ Flow\ Rate$
$\hat{n} - Direction\ Vector$
$\dot{Q}_{G,net_i} + \dot{Q}_{a_i} - Energy\ Released\ Due\ to\ Combustion\ and\ Excess\ Air$
$\dot{Q}_{enth_i} - Energy\ Change\ in\ Enthalphy\ of\ moving\ fluid$
$\dot{Q}_{conv_i} - Energy\ Transferred\ due\ to\ Convection$
$\dot{Q}_{Rad_i} - Energy\ Transferred\ due\ to\ Radiation$
$R_s - Air\ to\ Fuel\ Volume\ Ratio$
$\overline{S_iS_j} - Direct\ Exchange\ Area\ between\ i\ and\ j$
$\overline{S_iS_j} - Total\ Exchange\ Area\ between\ i\ and\ j$
$\overrightarrow{S_LS_w} - Directed\ Flux\ Area\ from\ Load\ Surface\ Zone\ to\ Wall\ Surface\ Zone$
$T_i - Temperature$
$V - Volume\ of\ Radiant\ Chamber$
$\dot{V}_{fuel} - Volumetric\ Flow\ Rate\ of\ Fuel$
$X - Excess\ Air\ Percentage$
$x - Length\ of\ the\ chamber\ from\ the\ bottom$
$Y - \%\ of\ Combustion\ Energy\ Releasd$

Greek Symbols
$\lambda = Conduction\ Heat\ Transfer\ Coefficient$
$\tau - transmissivity$
$\varepsilon_i = Emissivity$
$\rho_i - Density$

BIOGRAPHY

Rois Fatoni is an Assistant Professor, and Department Chair of Chemical Engineering Program, Universitas Muhammadiyah Surakarta, INDONESIA. He earned S.T. in Chemical Engineering from Universitas Gadjah Mada, Yogyakarta, Indonesia, M.Sc. in Environmental Process Design from University of Manchaster, UK, and Ph.D in Chemical Engineering from University of Waterloo, Canada. During his PhD studies he was supervised by Professors **Ali Elkamel** and **Leonardo Simon**. His research interests include modeling, simulation, optimization, scheduling, process safety, mixture design, advanced materials, biomass, and innovative technology for medium-small scale industries. He can be reached via email at: rois.fatoni@ums.ac.id

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Ali Elkamel is a Professor of Chemical Engineering at the University of Waterloo. He holds a BSc in Chemical Engineering and BSc in Mathematics from Colorado School of Mines, MS in Chemical Engineering from the University of Colorado-Boulder, and PhD in Chemical Engineering from Purdue University (West Lafayette), Indiana. His specific research interests are in computer-aided modelling, optimization and simulation with applications to energy production planning, sustainable operations and product design. He has supervised over 70 graduate students (of which 30 are PhDs) in these fields and his graduate students all obtain good jobs in the chemical process industry and in academia. He has been funded for numerous research projects from government and industry. His research output includes over 190 journal articles, 90 proceedings, over 240 conference presentations, and 30 book chapters. He is also a co-author of four books; two recent books were published by Wiley and entitled Planning of Refinery and Petrochemical Operations and Environmentally Conscious Fossil Energy Production.

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