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A Heuristic Approach for Modeling of the Furnace of a Utility Boiler

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Abstract

Mathematical modeling and simulation of boiler or total power plant is often required for analyzing the transient behavior of critical process parameters, designing suitable control philosophies and calculating the possible life expenditure of thick walled components during startups and load swings. Towards this purpose, the various subsystems of the boiler such as furnace, mills, fans, pumps, circulation system, super heaters, desuper-heater, re-heater, economizer and air heaters are modeled independently and integrated together. While the detailed and accurate models are available for various subsystems, a detailed but computationally simple model for furnace is not available. This is due to the fact that the furnace is a complex system wherein the surfaces of water walls and different radiant superheaters and reheaters receive heat by direct radiation and the remaining sensible energy determines the flue gas temperature at the furnace outlet plane. The quantity of direct radiation to these surfaces is further affected by the burner tilt. This paper presents a simplified but still accurate procedure for the development of model equations for the furnace of a utility boiler of high capacity based on operational experience and heuristic algorithms.

Keywords: Furnace, mathematical modeling**,** heuristic algorithm, boiler, superheaters.

1. Introduction

The furnace of high capacity utility boiler based on combustion engineering design is shown schematically in Fig.1a. The furnace consists of water walls also known as riser tubes. The drum, down comer and water walls all together constitute the circulation system as shown in Fig.1b.

The water walls where in the steam generation takes place receives heat only by direct radiation. The panel super heater which is located at the top left corner of the furnace receives heat mostly by direct radiation. The platen super heater and re heater which are also located at the top of the furnace receive heat partly by direct radiation and mostly by convection due to the fact that flue gas passes over the platen and preheater surfaces with significant flue gas velocity. The flue gas leaves the furnace outlet plane with a temperature depending on the sensible heat available in flue gas which in turn is the resultant of the net heat input minus all useful heat transferred by direct radiation from the furnace and unaccounted heat losses. The quantities of direct radiation heat transfer to different systems(waterwalls, super heaters and reheaters) are further affected by burner tilt. Thus the process dynamics within the furnace is quite complex and hence modeling of furnace poses severe challenges. The designers use sophisticated computer software such as computational fluid dynamics (CFD) and validated advanced heat transfer correlations for modeling of the furnace.

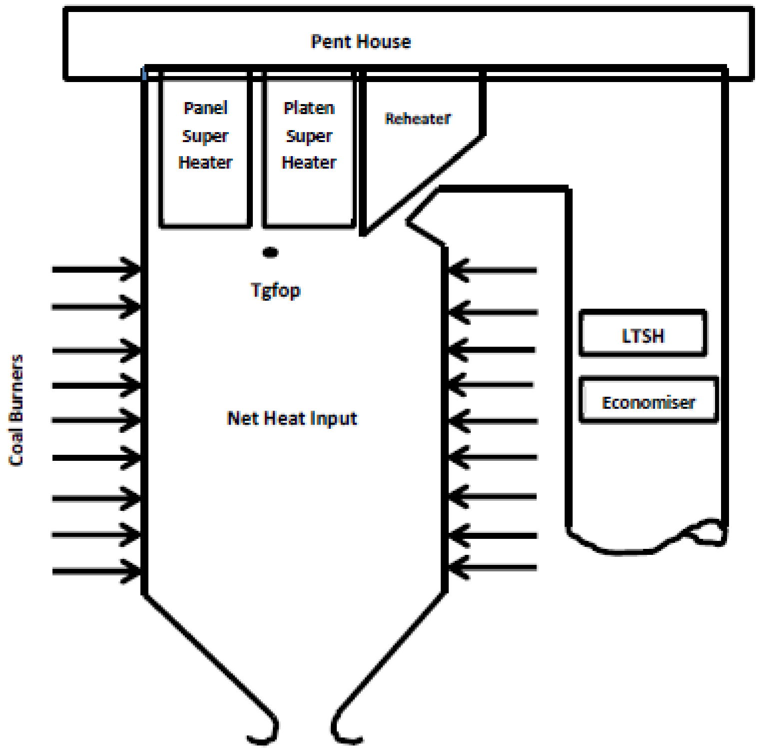

Fig. 1a. Typical Schematic diagram of Furnace of 500 MW Utility Boiler

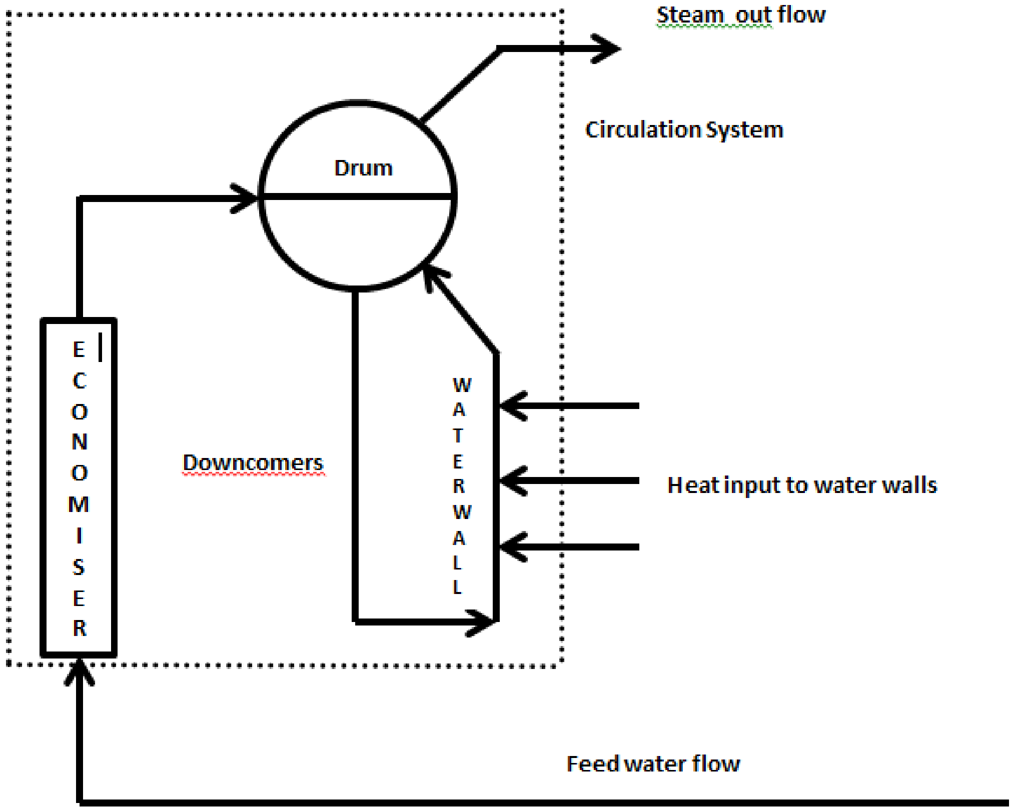

Fig.1b. Schematic diagram of Circulation System

Carvalho and Farias have briefly outlined the commonly used methods such as Zone method, Monte Carlo method and Flux method in their work on Modeling of Heat Transfer in Radiating and Combusting Systems [1]. Raymond Viskanta has reviewed four different methods for solving complex Radiative Heat Transfer equations applicable for chemically reacting and radiating combustion systems.[2]. Mohammed Bordbar and Timo Hyppane used Zone method for calculating radiative heat transfer in large pulverized utility boiler furnaces [3] and brought out that usage of CFD leads to more computational time since CFD requires a turbulent model coupled with very fine mesh. All these efforts are necessary in the design of furnace which will ensure performance optimization of boiler.

Mathematical modeling and simulation of boiler or total power plant is often required for analyzing the transient behavior of critical process parameters, designing suitable control philosophies during load swings and fuel switching (variation of calorific value of coal). From control system and simulation engineers' point of view, a detailed but still computationally simple model for furnace is desirable which can be integrated with the other subsystems of the boiler. The models for these subsystems such as mills, fans, pumps, super heaters, de-superhe ater, re-heater and economizer are well illustrated by researchers [4-17]. Wherever the physics of the processes/systems are complex and not fully understood, modeling and simulation of these systems are carried out by system identification methods [18-29], neural networks [30-33] and heuristic approaches [34-37]. This paper presents a procedure for development of furnace model based on operational experience and heuristic algorithms.

2. Problem Statement

The furnace layout shown in Fig.1a. is based on combustion engineering design for a 500 MW boiler. The Net Heat Input (NHI) to the furnace is decided by the amount of coal flow rate with known calorific value, the primary air flow rate and secondary air flow rate and the air temperatures. Knowing these input quantities, the following variables are to be determined as furnace output variables.

1. Net Heat input to the furnace in Kcal/Sec (NHI)
2. Heat transferred to water walls by direct radiation in Kcal/sec (Qgain-cir)
3. Heat transferred to Divisional Panel Super heater by direct radiation in Kcal/sec (Qgain-pn)
4. Heat transferred to Platen Superheater by radiation in Kcal/sec (Qgain-pt)
5. Heat transferred to reheater 1 by direct radiation in Kcal/sec (Qgain-rh1)
6. Heat transferred to reheater 2 by direct radiation in Kcal/sec (Qgain-rh2)
7. Temperature of flue gas at the furnace outlet plane in ^{0}C (Tgfop)
8. Flue gas mass flow rate in Kg/sec (m_g)

Based on the experience on 500 MW boiler for more than two decades, the following assumptions are made:

- The water walls receive heat from furnace totally by direct radiation.
- The Divisional panel superheater is located at the left corner and below the pent house. Hence, no convective heat transfer is possible and this section also receives heat totally by direct radiation.
- On the contrary, the platen superheater and reheater receive heat both by direct radiation and convective heat transfer.
- From operational experience, it is observed that the flue gas temperature drop across the platen superheater and reheaters is nearly a constant for varying loads.
- The flue gas temperature at the furnace outlet is to be determined based on sensible heat available at the furnace outlet plane.
- As per the design procedure, about 8% of the Net Heat input is considered as unaccounted radiative losses.

2.a. System Understanding

In this paper, a pulverized corner fired furnace is considered. There are nine elevations through which coal can be sent to the furnace. For a full load 500MW, fuel is fired through six elevations with appropriate burner tilt. The number of elevations and burner tilt vary depending upon the part load. The net heat input to the furnace is decided by the calorific value and coal flow rate and the sensible heat brought in by primary and secondary air.

The furnace and different heat exchangers like panel super heater, platen super heater and re heater which are located at the top of the furnace in the flue gas path are shown in Fig-1a. The water from economizer goes to the drum and gets converted into steam in the circulation system (drum – down comers and water walls). The steam from the drum gets super-heated in various super heaters and enters the high pressure turbine. A major portion of high pressure turbine exhaust steam comes back to the boiler and gets superheated in re heater system.The water and steam path explained is shown in Fig-2 along with the measured quantities normally available in any boiler unit.

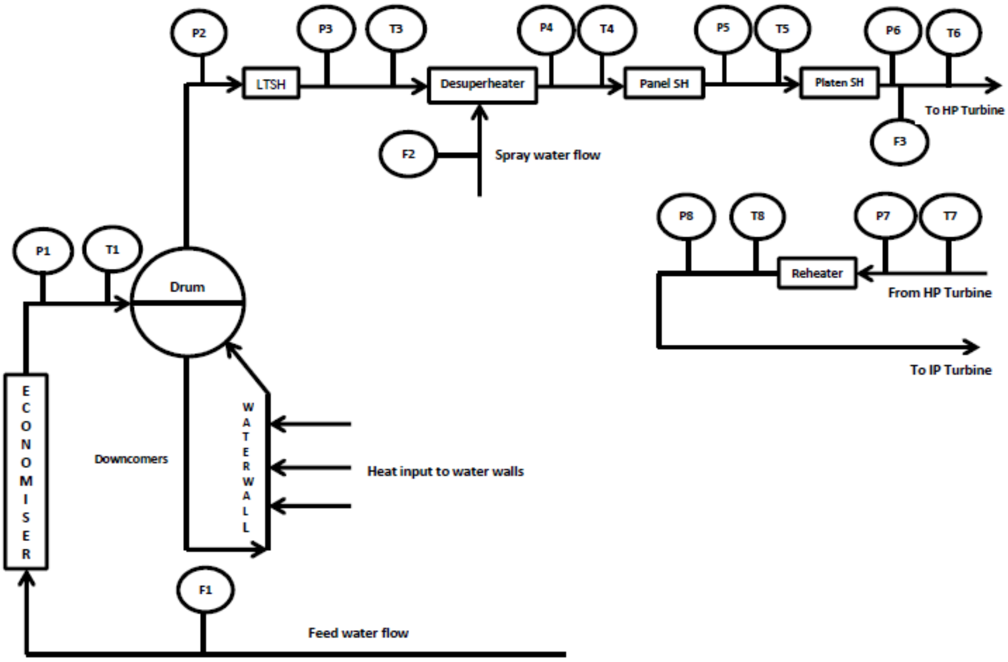

Fig. 2. Pressure and Temperature measurements at different nodes in a Boiler

The description of the variables used to represent the process measurement nodes is tabulated in Table 1.

Table 1 Description of Process Measurement Variables

S. No.	Variable	Description	Unit
1.	F1	Mass Flow rate at Economizer inlet	Kg/Sec
2.	P1	Pressure of water at the outlet of	Kg/cm^2
3.	T1	Temperature of water at the outlet of Economizer	^0C
4.	P2	Pressure at the inlet of LTSH	Kg/cm^2
5.	F2	Mass Flow rate of Super heater spray	Kg/Sec
6.	P3	Pressure at the inlet of De-super heater	Kg/cm^2
7.	T3	Temperature at the inlet of De-super heater	^0C
8.	P4	Pressure at the inlet of Panel Super heater	Kg/cm^2
9.	T4	Temperature at the inlet of Panel Super heater	^0C
10.	P5	Pressure at the inlet of Platen Super heater	Kg/cm^2
11.	T5	Temperature at the inlet of Platen Super heater	^0C
12.	F3	Mass Flow rate at the outlet of Platen Super heater	Kg/Sec
13.	P6	Pressure at the outlet of Platen Super heater	Kg/cm^2
14.	T6	Temperature at the outlet of Platen Super heater	^0C
15.	P7	Pressure at the inlet of Preheater	Kg/cm^2
16.	T7	Temperature at the inlet of Preheater	^0C
17.	P8	Pressure at the outlet of Preheater	Kg/cm^2
18.	T8	Temperature at the outlet of Re-heater	^0C

Predicted Performance of the boiler at different loads will usually be supplied by OEMs. The pressure, temperature and flow parameters at different nodes will be available. Further, the burner tilt and calorific value of the fuel will also be available. The enthalpies of water or steam can be calculated from pressure and temperature measurements using steam tables. The necessary process parameters required for the present problem at various locations are also shown in Fig-2. Based on the experience of the first author for more than thirty years in Corporate Research and Development Division of an OEM in power sector – simple heuristic models are proposed for the stated problem.

3.Furnace Modeling

The heat flux to water walls is totally by direct radiation and is directly proportional to the net heat input to the furnace. Further, this is corrected by the burner tilt. A positive burner tilt shifts the mean flame zone upwards and a negative burner tilt shifts the mean flame zone downwards. Direct radiation to the water walls gets affected with burner tilt. The following statements / assumptions hold good for the development of furnace model.

1. The heat energy transferred from furnace to water walls by direct radiation is equal to the heat gained in converting the feed water at economizer outlet to

saturation steam corresponding to drum pressure. The blow downs are assumed to be zero.

2. The heat gained in platen super heater, divisional panel and re-heaters are due to two parts, the first part is due to direct radiation from furnace and the second due to convective heat transfer due to flue gas flow. The flue gas temperature drop across the panel, platen and re heaters is observed to be constant at different loads as shown in Fig. 3a. and Fig. 3b. But the total convective heat transfer at different loads differs due to change in flue gas rate.

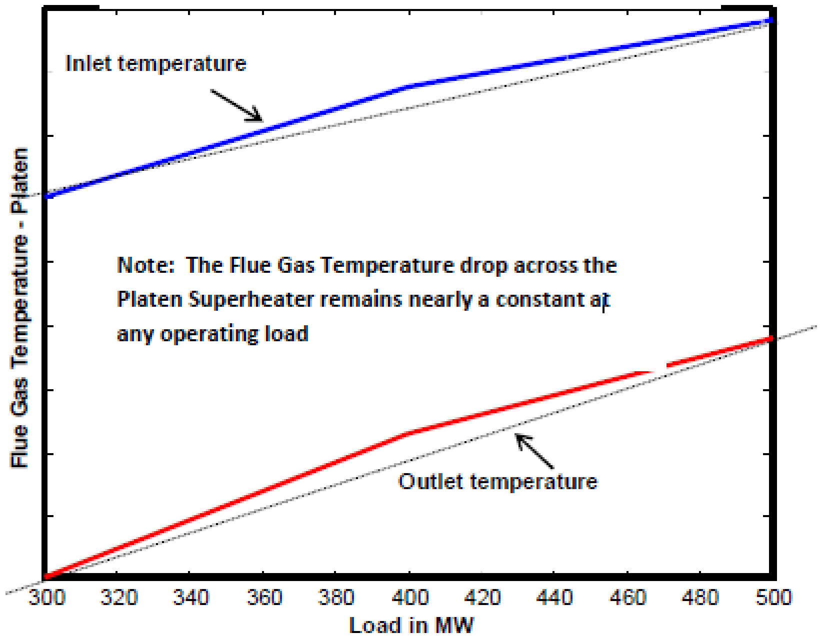

Fig.3. a. Flue gas temperature profile at the inlet and outlet of Platen (Final) Superheater

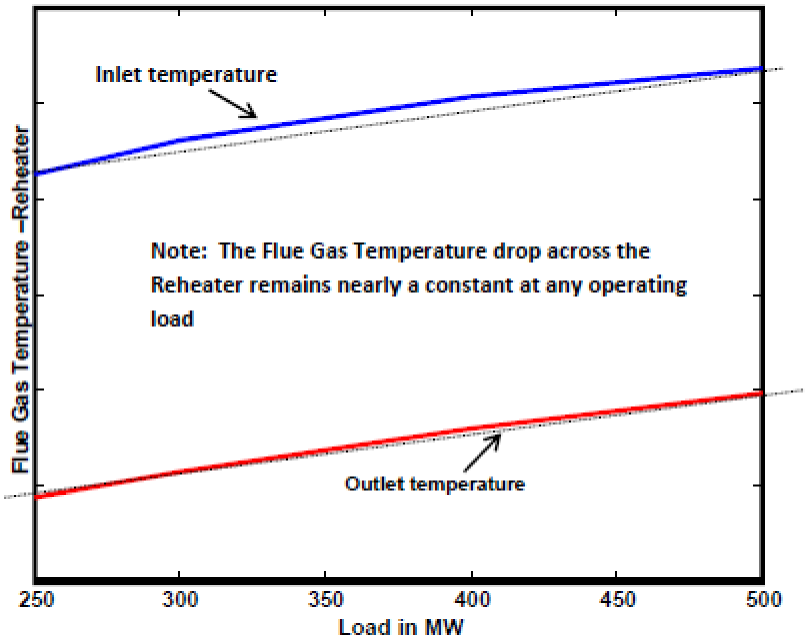

Fig.3.b. Flue gas temperature profile at the inlet and outlet of Reheater

3. The above understanding and arguments hold good for final super heater and re-heaters as well.
4. The net heat input brought into the furnace by coal and hot air (primary and secondary) results in an adiabatic flame temperature. The resultant mean flame zone along the vertical axis of the furnace is influenced by the burner tilt. In order to account for the effect of burner tilt, the net heat input is resolved into vertical and horizontal components with respect to burner tilt.

With the above stated design criteria, the following equations can be formulated.

$$Q_{gain\text{-}cir} = m_1 (h_2 - h_1) = \alpha_1 NHI \cos BT + BT*\alpha 2*NHI*\sin BT \qquad (1)$$

where
NHI -Net Heat Input to the furnace in Kcal/sec
m_1 -feed water flow rate in Kg/sec
h_1-enthalpy of feed water at economizer outlet in Kcal/Kg and $h_1 = f(P_1, T1)$
h_2-enthalpy of saturated steam in Kcal/Kg and $h_2 = f(P_2)$
α_1, α_2-unknown model parameters to be estimated

$Q_{gain\text{-}cir}$ can be calculated using measurements for more than two different loads and the two unknown parameters α_1 and α_2 can be estimated in the sense of least squares.

By knowing the value of these parameters, the Net Heat Input (NHI) can be calculated as

$$NHI = m_c * CV + (m_{sa} * C_{psa} * T_{sa}) + (m_{pa} * C_{ppa} * T_{pa}) \qquad (2)$$

where

m_c – Mass Flow Rate of Coal in Kg/Sec
CV – Calorific Value of Coal in Kcal
m_{sa} – Mass Flow Rate of Secondary Air in Kg/Sec
C_{psa} – Specific Heat Capacity of Secondary Air Kcal/Kg-^{0}C
T_{sa} – Temperature of Secondary Air in ^{0}C
mpa – Mass Flow Rate of Primary Air in Kg/Sec
C_{ppa} – Specific Heat Capacity of Primary Air Kcal/Kg-^{0}C
T_{pa} – Temperature of Primary Air in ^{0}C

For a typical 500 MW unit, the value of NHI at different loads, calculated using relation (2) is given in Table 2. The burner tilt associated with different loads is also given in Table 2.

Table 2 Net Heat Input at different Loads

S. No.	Load (MW)	NHI (Kcal/Sec)	Burner Tilt(BT) in deg
1	500	358976	-21
2	400	294076	-10
3	300	226934	+7
4	250	195478	+11

For a typical 500 MW unit, the values of $Q_{gain-cir}$ at different loads, calculated using the first part of relation (1) is given in Table 3.

Table 3 Heat flux to water walls at different loads

S. No.	Load (MW)	m_1 (Kg/Sec)	h_1 (Kcal/Kg)	h_2 (Kcal/Kg)	$Q_{gain-cir}$ (Kcal/Sec)
1	500	411.11	360.57	594.69	96251.52
2	400	315.8	346.26	601.49	80601.63
3	300	229.5	332.54	606.23	62811.09

Now the unknown values of α_1 and α_2 in the second part of Eq(1) can be solved by formulating three algebraic equations-corresponding to three different loads-using the values of NHI, Burner Tilt and $Q_{gain-cir}$ given in Table 2 and 3.

$Q_{gain\text{-}cir} = \alpha 1 * NHI * \cos(BT) + BT * \alpha_2 * \sin(BT) * NHI$

The above set of equations can be represented in matrix form as follows:

$$A * X = B \tag{3}$$

Where

$$A = \begin{bmatrix} 335104 & -2701041 \\ 289606 & -510510 \\ 225231 & 193641 \end{bmatrix}; X = \begin{bmatrix} \alpha_1 \\ \alpha_2 \end{bmatrix} ; B = \begin{bmatrix} 96251 \\ 80601 \\ 62811 \end{bmatrix}$$

Eq (3) can be solved in the sense of minimum Least Squares Method as follows:

$$[A^T A]^{-1}[A^T A]\, X = [A^T A]^{-1} A^T B$$

$$X = [A^T A]^{-1} A^T B$$

The matrix X is calculated as

$$X = \begin{bmatrix} \alpha_1 \\ \alpha_2 \end{bmatrix} = \begin{bmatrix} 0.2780 \\ -0.0011 \end{bmatrix}$$

The heat flux to the divisional panel super heater is mostly by direct radiation and is directly proportional to the net heat input to the furnace which is further corrected by the burner tilt. With this perception, a heuristic relationship for heat flux by direct radiation to panel super heater can be written as:

$$Q_{gain\text{-}pn} = \alpha 3 * NHI * \cos(BT) + BT * \alpha 4 * \sin BT * NHI = m_2 (h_5 - h_4) \tag{4}$$

Where $\alpha_{3,}$ α_4 are to be evaluated using known values of NHI, $Q_{dr\text{-}pn}$ and burner tilt for different MW ratings given in Table 4.

Table 4 Heat flux to Panel Super heater at different loads

S. No.	Load (MW)	m_2 (Kg/Sec)	h_4 (Kcal/Kg)	h_5 (Kcal/Kg)	$Q_{gain\text{-}pn}$ (Kcal/Sec)
1	500	416.11	673.22	755.01	34032.08
2	400	333	666.02	750.13	30002.42
3	300	252.5	644.51	743.77	25064.05

The Eq (4) is represented in the form of a matrix equation and solved in the sense of least squares.

A* X = B

Where

$$A = \begin{bmatrix} 335104 & -2701041 \\ 289606 & -510510 \\ 225231 & 193641 \end{bmatrix}; X = \begin{bmatrix} \alpha_3 \\ \alpha_4 \end{bmatrix} ; B = \begin{bmatrix} 34032 \\ 30002 \\ 25064 \end{bmatrix}$$

The values of α_3 and α_4 thus obtained are

$$\begin{bmatrix} \alpha_3 \\ \alpha_4 \end{bmatrix} = \begin{bmatrix} 0.1076 \\ 0.0008 \end{bmatrix}$$

The heat flux to the platen superheater is partly by convective heat transfer from flue gas and the remaining by direct radiation. However, the total heat flux to the platen super heater can be written as determined from steam side as

$$Qgain\text{-}pt = Q_{dr\text{-}pt} + Qconv_{\text{-}pt}$$

Where

$$Qgain\text{-}pt = \alpha_5 * NHI \cos (BT) + BT * \alpha_6 * NHI * \sin(BT) = m_3 (h_6 - h_5) \quad (5)$$

Here, α_5 and α_6 are to be evaluated using known values of NHI, $Q_{gain\text{-}pt}$ and burner tilt for different MW ratings

The convective component of heat transfer to the platen super heater is obtained by the relation

$$Q_{conv\text{-}pt} = m_g * C_p * \Delta T_{g\text{-}pt}$$

Where

m_g – Mass Flow Rate of Flue Gas in Kg/Sec

C_p – Specific Heat Capacity of Flue Gas in Kcal/Kg/^{0}C

$\Delta T_{g\text{-}pt}$ – Temperature drop across the Platen Super heater

The temperature drop across the Platen Super heater, $\Delta T_{g\text{-}pt}$ remains very nearly a constant for a wide range of loads. This value can usually be found from the predicted performance curves supplied by the OEMs and this drop is found to be roughly a constant at 110^{0}C.

Knowing the values of $Qgain_{\text{-}pt}$, $Qconv_{\text{-}pt}$ and $Q_{dr\text{-}pn}$ from Table 5, the matrix equations are formed and the unknown constants α_5 and α_6 are determined.

Table 5 Heat flux to Platen Super heater at different loads

S. No.	Load (MW)	m_3 (Kg/Sec)	h_5 (Kcal/Kg)	h_6 (Kcal/Kg)	$Q_{gain-pt}$ (Kcal/Sec)	$Q_{conv-pt}$ (Kcal/Sec)	Q_{dr-pt} (Kcal/Sec)
1	500	416.38	755.1	810.8	23192	16688	6504
2	400	330.05	750.6	811.5	20283	13728	6555
3	300	252.5	744.1	812	17144	10716	6428

The matrix equations are formed using the values given

$$A = \begin{bmatrix} 335104 & -2701041 \\ 289606 & -510510 \\ 225231 & 193641 \end{bmatrix}; X = \begin{bmatrix} \alpha_5 \\ \alpha_6 \end{bmatrix} ; B = \begin{bmatrix} 6504 \\ 6555 \\ 6428 \end{bmatrix}$$

The values of the unknown constants thus obtained are

$$\begin{bmatrix} \alpha_5 \\ \alpha_6 \end{bmatrix} = \begin{bmatrix} 0.0258 \\ 0.0008 \end{bmatrix}$$

As in the case of platen super heater, the heat flux to the reheater 1 and reheater 2 is also partly by convective heat transfer from flue gas and the remaining by direct radiation. However, the total heat flux to the reheater1 and 2 can be written as determined from steam side as

$$Q_{gain-rh1} = Q_{dr-rh1} + Q_{conv-rh1}$$

$$Q_{gain-rh2} = Q_{dr-rh2} + Q_{conv-rh2}$$

where

$$Q_{gain-rh1} = \alpha_7 * NHI \cos(BT) + BT*\alpha_8*NHI*\sin(BT) = m_4 (h_8 - h_7) \qquad (6)$$

$$Q_{gain-rh2} = \alpha_9 * NHI \cos(BT) + BT*\alpha_{10}*NHI*\sin(BT) = m_5 (h_{10} - h_9) \qquad (7)$$

Here, the values of co-efficient α_7, α_8 for reheater1 and α_9, α_{10} for reheater2 and are to be evaluated using known values of NHI, $Q_{gain-rh1}$, $Q_{gain-rh2}$ and burner tilt for different MW ratings

The convective component of heat transfer to the reheaters is obtained by the relation

$$Q_{conv-rh1} = m_g*C_p*\Delta T_{g-rh1} \qquad (8)$$

$$Q_{conv-rh2} = m_g*C_p*\Delta T_{g-rh2} \qquad (9)$$

Where

mg – Mass Flow Rate of Flue Gas in Kg/Sec
C_{p} – Specific Heat Capacity of Flue Gas in Kcal/Kg/^{0}C
$\Delta T_{g\text{-}rh1}$ – Temperature drop across the Reheater1
$\Delta T_{g\text{-}rh2}$ – Temperature drop across the Reheater2

The temperature drop across the reheater1 and 2, $\Delta T_{g\text{-}rh1}$ and $\Delta T_{g\text{-}rh2}$ remains very nearly a constant for a wide range of loads. This value can usually be found from the predicted performance curves supplied by the OEMs and this drop is found to be roughly a constant at ^{0}C.

Knowing the values of $Q_{gain\text{-}rh1}$, $Q_{conv\text{-}rh1}$ and $Q_{dr\text{-}rh1}$ from **Table** 6, the matrix equations are formed and the unknown constants α_7 and α_8 are determined.

Table 6 Heat gained by reheater1

S. No.	Load (MW)	m_4 (Kg/Sec)	h_7 (Kcal/Kg)	h_8 (Kcal/Kg)	$Q_{gain\text{-}rh1}$ (Kcal/Sec)	$Q_{conv\text{-}rh1}$ (Kcal/Sec)	$Q_{dr\text{-}rh1}$ (Kcal/Sec)
1	500	370.8	729.0	802.6	27291	24274	3017
2	400	299.44	729.4	805.8	22877	19967	2910
3	300	229.44	729.9	809.5	18263	15586	2677

The matrix equations are formed using the tabulated quantities and the values of the unknown constants α_7 and α_8 are determined as follows

$$\begin{bmatrix}\alpha_7\\ \alpha_8\end{bmatrix}=\begin{bmatrix}0.0110\\ 0.0003\end{bmatrix}$$

In the same way, by knowing the values of $Q_{gain\text{-}rh2}$, $Q_{conv\text{-}rh2}$ and $Q_{dr\text{-}rh2}$ from **Table** 7, the matrix equations are formed and the unknown constants α_9 and α_{10} are determined.

Table 7 Heat gained by reheater2

S. No.	Load (MW)	m_4 (Kg/Sec)	h_9 (Kcal/Kg)	h_{10} (Kcal/Kg)	$Q_{gain\text{-}rh2}$ (Kcal/Sec)	$Q_{conv\text{-}rh2}$ (Kcal/Sec)	$Q_{dr\text{-}rh2}$ (Kcal/Sec)
1	500	370.8	809.5	847.8	14201	13031	1170
2	400	299.44	805.8	846.3	12127	10732	1395
3	300	229.44	802.6	844.2	9545	8377	1168

The matrix equations are formed using the tabulated quantities and the values of the unknown constants α_9 and α_{10} are determined as follows

$$\begin{bmatrix}\alpha_9\\ \alpha_{10}\end{bmatrix}=\begin{bmatrix}0.0051\\ 0.0002\end{bmatrix}$$

The value of the co-efficient thus obtained using design coal specifications is summarized in Table 8.

Table 8 Table of co-efficient obtained for design coal

S. No.	Component	Co-efficients	Values
1	Water walls	α1	0.2780
		α2	-0.0011
2	Panel Super heater	α3	0.1076
		α4	0.0008
3	Platen Super heater	α5	0.0258
		α6	0.0008
4	Reheater1	α7	0.0110
		α8	0.0003
5	Reheater2	α9	0.0051
		α10	0.0002

Having determined the co-efficient, the temperature of the flue gas at the furnace outlet plane is found using the relation.

$$T_{gfop} = \frac{NHR-(Q_{gain-cir}+Q_{gain-pn}+Q_{gain-pt}+Q_{gain-rh1}+Q_{gain-rh2})}{m_g * C_p}$$

In order to validate the furnace model that is represented by the equations 4, 5, 6 and 7 Q_{gain} values of the various superheaters for worst coal and best coal specifications are determined using the deduced heuristic furnace model. These values are used to determine the temperature of flue gas at the furnace outlet plane and the temperature readings($T_{gfop-est}$) thus obtained are found to be in good agreement with the temperature readings($T_{gfop-actual}$) found in the predicted performance characteristics supplied by the OEM. The values of temperature for worst coal and best coal are tabulated in Table 9.

Table 9 Validation of furnace model using T_{gfop} values for worst coal and best Coal specification

S. No.	Load (MW)	Worst Coal			Best Coal		
		$T_{gfop-est}$ (deg. C)	$T_{gfop-actual}$ (deg. C)	% error	$T_{gfop-est}$ (deg. C)	$T_{gfop-actual}$ (deg. C)	% error
1	500	1246	1129	9.3	1117	1145	2.4
2	400	1172	1099	6.2	1121	1131	0.8
3	300	1090	1066	2.2	1179	1098	6.8

4. Results and Discussions

Based on the calculations, the salient equations describing the complex process taking place within the furnace of a generic 500MW boiler in are summarized in Table 10.

Table 10 Heuristic Model Equations and co-efficient values

S. No.	Component	Heuristic Model Equation	Co-efficients	Values
1	Water walls	Qgain-cir = $\alpha 1$*NHI*cos (BT) + α_2* BT* NHI *sin (BT)	$\alpha 1$	0.2780
			$\alpha 2$	-0.0011
2	Panel Superheater	Qgain-pn= $\alpha 3$*NHI* cos (BT)+ $\alpha 4$ *BT* NHI *sin (BT)	$\alpha 3$	0.1076
			$\alpha 4$	0.0008
3	Platen Superheater	Qgain-pt = α_5*NHI *cos (BT)+ α_6*BT* NHI*sin(BT)	$\alpha 5$	0.0258
			$\alpha 6$	0.0008
4	Reheater 1	Qgain–rh1 = α_7*NHI* cos (BT) + α_8*BT* NHI*sin(BT)	$\alpha 7$	0.0110
			$\alpha 8$	0.0003
5	Reheater 2	Qgain–rh2 = α_9*NHI* cos (BT) + α_{10}*BT* NHI*sin(BT)	$\alpha 9$	0.0051
			$\alpha 10$	0.0002

It can be seen from the algebraic equations that the effect of burner tilt is properly accounted. For a negative burner tilt, the heat flow rate to the waterwalls is more while for a positive burner tilt the heat flow rate to the water walls is less. A positive burner tilt results in more direct radiation to superheater sections hanging from the penthouse. This is in good agreement with the physics of the process. Further, the calculated values using the above heuristic equations (algorithms) for various sections are in good agreement with those obtained through design procedure.

Conclusion

A detailed yet computationally simple and elegant algebraic model based on operational experience for furnace of a utility boiler has been presented in this paper. The procedure can be applied to any furnace of power boiler and the model equations can be developed. This model is suitable for real time applications. The results obtained are in good agreement with those obtained by detailed design calculation procedures.

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