Mathematical Modeling and Simulation of Heat Transfer Analysis of Thermal Power Plant Components

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Mathematical Modeling and Simulation of Heat Transfer Analysis of Thermal Power Plant Components

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Submitted in partial fulfillment of the requirements

For the Degree of

Master of Technology in Mechanical Engineering (Thermal Engineering)

By

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Abstract

Boiler consists of several types of interconnected heat exchangers like superheater, reheater, economizer and boiler drum in which fluid flow and heat transfer processes takes place. It is very important to carry out heat transfer analysis and estimate the wall temperatures of various components for the safer and efficient operation of these components. In the present study, the dynamic behavior of flue gases to water/steam heat exchangers of a typical 500 MW Thermal power plant components are studied using the mathematical modeling and computer simulation. Various boiler components like superheater, reheater, economizer and boiler drum are modeled under connected conditions. The convective heat transfer coefficient, non-luminous heat transfer coefficient and heat transfer calculations are done using correlations available in the open literature. In the sections where radiation is applicable, appropriate amount is considered. The equations represented in the entire processes of boiler are in the form of ordinary differential equations. Based on the mass balance, heat balance and heat transfer models, differential and algebraic equations are formulated. The steady state and dynamic equations thus formed are solved using Excel along with Visual Basic Application (VBA). Advanced Continuous Simulation Language (ACSL-X) solver is used for solving the differential and algebraic equations. All the fluid/steam properties and transport properties for flue gases (with moisture) are considered under variable temperature conditions. From the simulation, convective heat transfer coefficient, non-luminous heat transfer coefficient, heat transfer rate for steam/water and gas side, wall/metal temperature at inlet and outlet have been determined. The deviation between simulation results and tested data was found within 0% to 1 % in the case of heat transfer coefficients, 2% to 7% in heat transfer rate and 0% to 0.7% deviation in steam temperature at outlet of superheater and reheater; which shows very good agreement. Inspite boiler heating sections, we also done modeling and simulation for turbine casing where similar methodology applied as heating sections of boiler. The obtained results of turbine casing was also found good agreement with expected trends.

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Nomenclature

W_{dc}	Downcomer flow rate($^{Kg}/_{hr.}$)
P	Larson miller parameter
$Q_{metal.}$	Heat transfer through metal($^{KCal}/_{hr}$)
K	Thermal conductivity ($^{W}/_{mK}$)
A	Surface area (m^2)
U	Overall heat transfer coefficient ($^{W}/_{m^2K}$)
D	Outer tube diameter (m)
d	Inside tube diameter (m)
$F_{f,o}$	Outside film fouling
$F_{f,i}$	Inside film fouling
Nu	Nusselt number
Re	Reynolds number
Pr	Prandtl number
V	velocity of fluid ($^{m}/_{hr.}$)
h_i	Inside heat transfer coefficient of steam ($^{W}/_{m^2K}$)
$h_c^{'}$	Basic convection heat transfer coefficient ($^{KCal}/_{hr.m^2.k}$)
h_c, r_c	Overall convective heat transfer coefficient ($^{KCal}/_{hr.m^2.k}$)
$h_n^{'}$	Basic non-luminous heat transfer coefficient ($^{KCal}/_{hr.m^2.k}$)
h_n, r_n	Non-luminous heat transfer coefficient ($^{KCal}/_{hr.m^2.k}$)
F_H	factor depends on tube arrangement
F_N	factor depends on number of rows
F_A	factor depends on angle of gas flow
L	Radiant beam length (m)
S_T	Transverse pitch (m)
S_L	Longitudinal pitch (m)
P_C	Partial pressure exerted by$CO_2(KPa)$
P_W	Partial pressure exerted by tube wall (KPa)
P_H	Partial pressure exerted by H_2O (KPa)
$T_{g,abs.}$	Absolute mean gas temperature ($^\circ C$)
$T_{M,abs.}$	Absolute outside metal temperature ($^\circ C$)
K_t	Factor associated in correction of h_N for wall temperature
K_e	Factor associated in correction of h_N for gas absorptivity
E_w	Tube wall emissivity
A_g	CO_2 andH_2O mixture absorptivity
Q_f	Heat transfer through fluid (steam/water) side($^{kCal}/_{hr.}$)
Q_g	Heat transfer through gas side($^{kCal}/_{hr.}$)

$\dot{m}_g$	Mass flow rate of gas $(^{Kg}/_{hr.})$
$\dot{m}_s$	Mass flow rate of steam $(^{Kg}/_{hr.})$
α_{hxmi}	Total heat transfer coefficient inside tube $(^{W}/_{m^2k})$
α_{hxmo}	Total heat transfer coefficient Outside tube $(^{KCal}/_{hr.m^2.k})$
A_{hxmi}	Inside surface area of tube (m^2)
A_{hxmo}	Outside surface area of tube (m^2)
$\alpha_{I,tube} = h_i$	Combined heat transfer coefficient at inside of tube $(^{W}/_{m^2k})$
$\alpha_{O,tube} = r_t$	Combined heat transfer coefficient at Outside of tube$(^{KCal}/_{hr.m^2.k})$
r_o	Outer radius (m)
r_i	Inner radius (m)
$r_{m,i}$	Inner zone centre radius (m)
$r_{m,o}$	Outer zone centre radius (m)
r_w	Wall thickness (m)
C_p	specific heat at constant pressur $(^{KCal}/_{KgC})$
C_h	Specific heat for hot side $(^{KCal}/_{KgC})$
C_c	Specific heat for cold side $(^{KCal}/_{KgC})$
C_m	Specific heat of metal $(^{KCal}/_{KgC})$
m_h	Mass of hot side (Kg)
m_c	Mass of cold side (Kg)
$T_{g,in}$	Inlet gas temperature $(^\circ C)$
$T_{g,out}$	Outlet gas Temperature $(^\circ C)$
$T_{g,avg.}$	Average gas temperature $(^\circ C)$
$T_{s,in}$	Inlet steam temperature $(^\circ C)$
$T_{s,out}$	Outlet steam temperature $(^\circ C)$
$T_{s,avg.}$	Average steam temperature $(^\circ C)$
$T_{wall,i}$	Inner wall temperature $(^\circ C)$
$T_{wall,o}$	Outer wall temperature $(^\circ C)$
$T_{m,i}$	Inner metal temperature $(^\circ C)$
$T_{m,o}$	Outer metal temperature $(^\circ C)$
$T_{h,i}$	Inlet hot temperature $(^\circ C)$
$T_{h,o}$	Outlet hot temperature $(^\circ C)$
$T_{c,o}$	Outlet cold temperature $(^\circ C)$
$T_{c,i}$	Inlet cold temperature $(^\circ C)$
$P_{g,in}$	Inlet gas pressure $(^{Kg}/_{cm^2})$
$P_{g,out}$	Outlet gas pressure $(^{Kg}/_{cm^2})$
$P_{s,in}$	Inlet steam pressure $(^{Kg}/_{cm^2})$
$P_{s,out}$	Outlet steam pressure $(^{Kg}/_{cm^2})$

Greek Words

ρ	Density of fluid ($^{Kg}/_{m^3}$)
η	Efficiency
μ	viscosity of fluid ($^{Kg}/_{m.sec}$)
σ	Stephan Boltzmann constant ($^{W}/_{m^2k}$)
ϵ	Heat exchanger effectiveness
$\delta_{r,i}$	Depth of inner zone of tube(m)
$\delta_{r,o}$	Depth of inner zone of tube (m)

Subscripts

m	Metal
h	Hot side
c	Cold side
i/in	Inlet
$o/out.$	Outlet
$min.$	Minimum
$max.$	Maximum
f	Film, fluid
g	Gas
s	Steam
$avg.$	Average
I	Inside
O	Outside
w	Wall
b	Boundary
C	Convective
n	Non-luminous
T	Transverse
L	Longitudinal
$abs.$	Absolute
WW	Water-walls
DC	Downcomer
ECO	Economizer

Abbreviations

BHEL	Bharat Heavy Electrical Limited
NTPC	National Thermal Power Plant Corporation
CFD	Computational Fluid Dynamics
FEM	Finite Element Method
FDM	Finite Difference Method
FVM	Finite Volume Method
LMTD	Log-Mean Temperature Difference
ASME	American Society of Mechanical Engineers
VBA	Visual Basic Application
ACSL-X	Advanced Continuous Simulation Language
SH	Superheater
RH	Reheater
WW	Water Wall
LTSH	Low Temperature Superheater
PSH	Platen Superheater
RHF	Reheater Front
RHR	Reheater Rear
ECO	Economizer

Chapter 1

Introduction

1.1 General

1.1.1 Bharat Heavy Electrical Limited (BHEL)

BHEL is an integrated power plant equipment manufacturer and places strong prominence on innovation and creative development. The Corporate R & D division at Hyderabad aimed not only at improving the performance and efficiency of existing products, but also developing new products using advanced technologies and processes. Power plant dynamics and simulation laboratory has ability in mathematical modeling of system, control system design and tuning, power plant performance diagnostics and optimization and development of power plant training simulators and dynamic simulation studies [1].

1.1.2 SIMHADRI power plant

SIMHADRI thermal power plant owned by NTPC is a modern coal fired plant of total capacity 2000 MW commissioned from 2002. This plant is a combination of independent generation of four units and each units has a capacity of 500 MW. The quantity of coal for 1 unit is approximately $340\,TPH$ having calorific value is around $3300\,Kcal/kg$. It is situated in the Vishakhapatnam, Andhra Pradesh. 1000 MW Power generated by the units 1 and 2 is supplied to companies under the government of Andhra Pradesh and remaining 1000 MW from units 3 and 4 is supplied to Odisha, Tamil Nadu and Karnataka [2].

1.1.3 Need of simulation

Dynamic simulation is the replica of the operations of a actual world processes over time or we can say that it is an attempt to model a real-life or hypothetical situation on a computer so that it can be studied to see how the system works. Simulation has to be done for trained the employees on the computer for their tasks and required operations by preparing a hypothetical model, which behave same as actual power plant. The tasks like starting up a plant following an overhaul, and also it helpful to control and maintain the different parameters of power plant. When the power plant is in running condition then at that time, parameter of actual plant can not be changed so that first required parameter tested on simulated model and then apply it to actual plant. So that plant operators can monitored online easily. Hence, problems can be marked in good time and corrected. So that the efficiency of power plant can be improved under best possible operating conditions. But, first an mathemtical model is required to simulate something which presents the main characterstics, behaviour and functions of the actual plant. Hence, we can say that model represent the system itself whereas simulation represents operations of plant with time. Hence, through the resolution of mathematical model we can represent the behaviour of real process called simulation[3]. The simulation also carried out for performance optimization, safety engineering, testing, training, education. The models are accurate in the range of few tenths up to a few thousand of a second as they signify whole plant.

1.2 Thermal power plant

Thermal power plant generates electricity by converting energy stored in coal into shaft work which further converted into electrical energy by the help of generator. In the whole process of thermal power plant first, generation of steam at high temeperature and pressure in the boiler is take place by burning of fuel which gives energy to water after it shaft work produces by the turbine due to expainsion of steam at low pressure. In the condenser, exhaust steam from turbine is condensed by releasing heat to cooling water in a river or sea and then by the help of pump, the condensed water again sent back to the boiler, and this cycle is repeated to generate electricity. Figure 1.1 shows the typical illustration of thermal power plant.

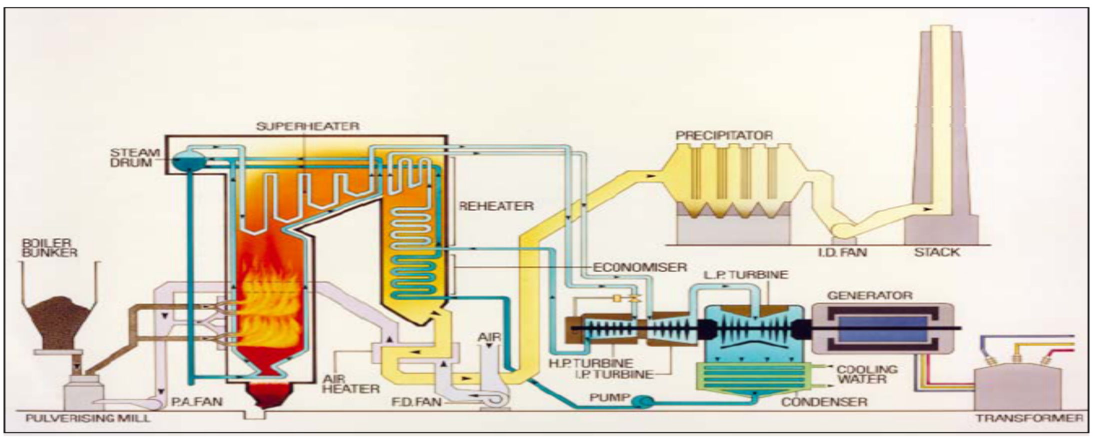


Figure 1.1: Illustration of a typical thermal power plant [4].

1.3 Power plant cycle

The basic cycle used in thermal power plant is called rankine cycle which works on dual (vapour and liquid) phase cycle where water is used as working fluid. In the ideal Rankine cycle, there are two isentropic and two constant pressure processes. At state 1, water enters to pump as saturated liquid and isentropically compressed to operating pressure of boiler hence temperature of water increases due to slight decrease in specific volume of water. At state 2, water enters to boiler as compressed liquid and exit as a superheated vapour at state 3. After it superheated vapour enter to turbine where isentropically expansion takes place and produce work by rotating shaft connected to generator. At state 4, exhaust steam from turbine enters to condenser where steam is condensed at constant pressure and reject heat to a cooling medium like lake, river etc. After it, steam enter to pump by leaving condenser as saturated liquid and complete the cycle. Figure 1.2 shows all the processes of ideal rankine cycle.

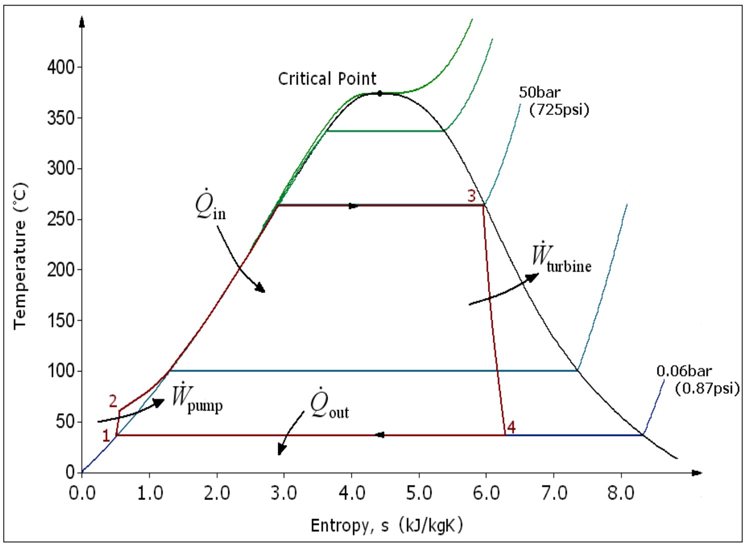


Figure 1.2: Ideal rankine cycle [5].

The actual vapour power cycle is different from ideal rankine cycle due to irreversibility in various components. The main reason for irreversibility are fluid friction and heat loss to surrounding, hence pressure and cycle efficiency reduces respectively. For improving the efficiency of a rankine cycle, reheating and regeneration principles are described. In the reheating, before going to the intermediate pressure turbine, reheaters are used to increase the temperature of exhaust steam coming from the high pressure turbine and removes moisture carried by steam at final stage of expansion process.Whereas according to regenerative principle, in the subcooled region, the specific heat of water is very high so that large amount of heat is required in this region. The steam extracted from turbine is used to heat the condenser water so that water enters the boiler at a higher temperature and consequently the heat supplied in the boiler is less and thermal efficiency can be improved [6]. Figure 1.3 descibes all the processes of reheat and regenerative cycle.

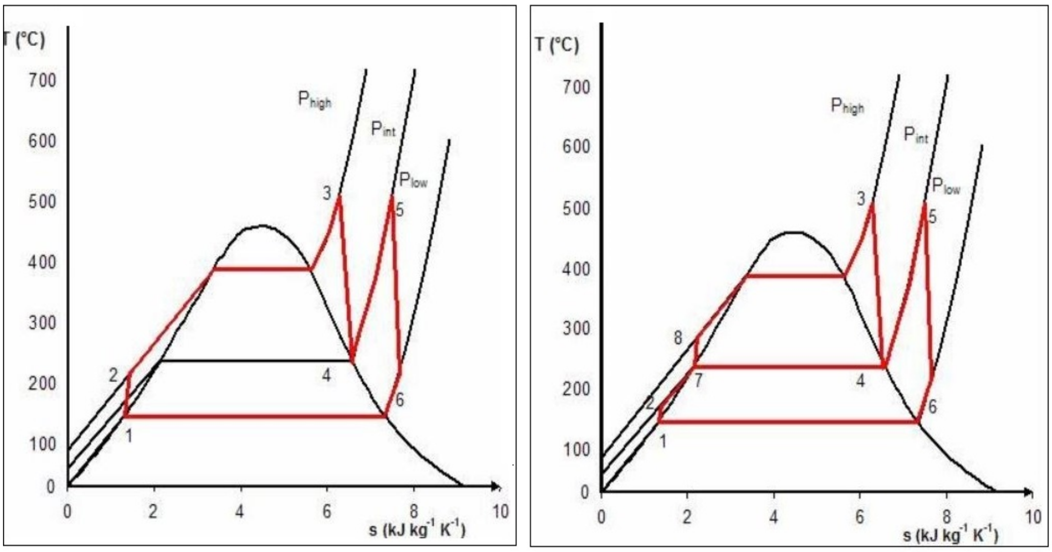


Figure 1.3: Reheat cycle [7] and regenerative cycle[8].

1.4 Boiler

Boiler is used for steam generation which consist of several heat generating units like furnace, boiler drum, superheaters, reheater, economiser, safety valves, drain valves, control valves etc. To achieve required steam at rated parameter, a complex coordination of all these units is required [9]. In the present modeling, the classification of boiler as shell and tube arrangement are as follows:

- It is water tube boiler where steam or water flows in a tube and heat is given from outside surface by the help of flue gas
- It is a fired type boiler where heat is produced by the help of product of fuel combustion.
- It is a forced circulation boiler.
- It is internally fired type because furnace is completely surrounded by water cooled surfaces.
- The rated efficiency of boiler is 87%.

Hence the simulators should build accurate and refined performance prediction boiler model to represent the actual processes happening during the operation. Figure 1.4 shows the layout of boiler.

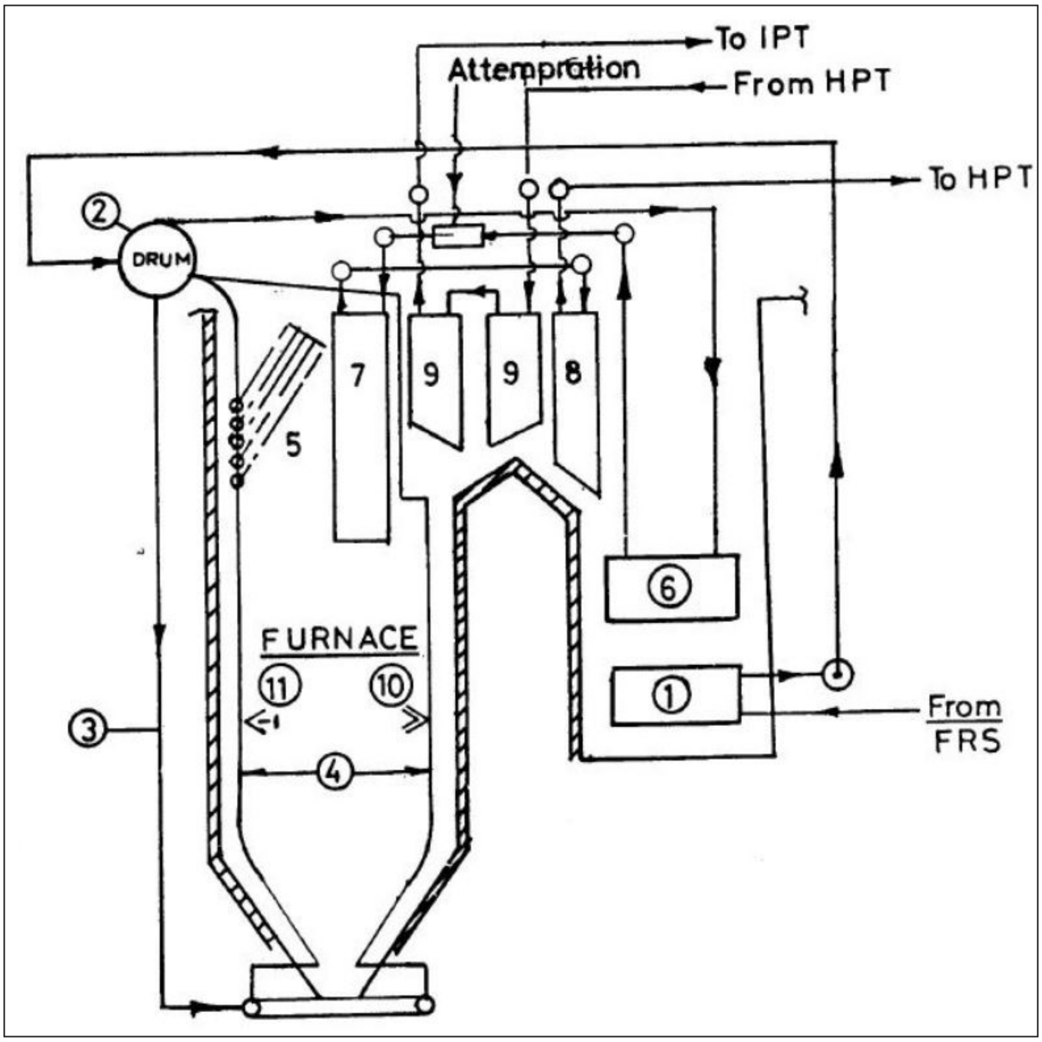


Figure 1.4: Layout of boiler [10].

In the above figure, the several components are denoted as below,

1. Economiser.
2. Boiler drum.
3. Downcomers.
4. Water walls.
5. Water wall platen.
6. Low temperature superheater.
7. Panel superheater.
8. Platen superheater.
9. Front and Rear reheater.
10. Burners.
11. Ignitors.
12. FRS is called feed regulating station.

1.5 Major components of boiler

The description of all the main components of boiler are as follows:

1.5.1 Furnace

Furnace is used for generating steam in which chemical energy of fuel is converted into heat energy which is absorbed by the heating surfaces of boiler. There are two types of furnace i.e. chamber type (flame) furnace and grate-fired furnace. In the chamber type, combustion takes place in the space above fuel bed. The size of fuel particles, mixing of fuel and surface area are the physical factors while concentration of reactants and temperature are the chemical factors of combustion of fuel in furnace. In a pulverized coal firing system, the mixture of powdered coal and air is drawn through the burners into the furnace where coal particles are heated at high temperature flame as a result volatile matter is distilled off, hence coal particles become very small sponge-like masses of fixed carbon and ash. Whereas due to mixing of oxygen of air with volatile matter, quick burning takes place. so that due to reaction of oxygen of hot air with carbon surface, energy is released. Ash formed due to combustion are partly falls to the bottom of furnace and remaining taken by gas stream as fly ash to flue gas outlet or set down on heating surfaces of boiler[9]. Whereas, water is circulated through the water wall tubes and generates steam. These water wall tubes are joined together in headers and connected to steam drum.

Through the forced draft fan, atmospheric air is heated in the air heaters and sent to the furnace as combustion air. Induced draft fan extract the flue gas from furnace and balance draft in the furnace (-6 to -15mm of water column) with forced draft fan. So that, various superheaters takes heat energy due to emmission of flue gases and finally these flue gases goes to the electrostatic precipitator through air preheaters. Hoppers are take place at bottom precipitator which collected these ash for disposal.

1.5.2 Water walls

These are vertical tubes and connected at the top and bottom to headers inside the furnace, used for heating and evaporating the feed water supplied to the boiler from the downcomers. These tubes take water from the boiler drum by means of downcomers joined between drum and water walls lower header. Absorbed heat by water walls is used as evaporation of water supplied to boiler. From the top of water walls tubes, mixture of steam and water is discharged into the upper wall header and then passes through riser tubes to the steam drum, where steam is separated and water with the incoming feed water is returned to the water walls through the downcomers [9].

1.5.3 Boiler drum

It is situated at upper front of boiler where separation of water will take place from the steam generated in the furnace water walls. In a drum, mixture of water and steam enters where water and steam gets seperated, so steam goes to superheaters and saturated water comes downward.

Drum is used to store water and steam in sufficient amount to fullfill the need at time varying load, to maintained circulation ratio, for seperation of vapour from the water and steam mixture and also for providing sufficient surface area for liquid and vapour disengagement.

1.5.4 Superheater

Superheaters are used to increase the main steam temperature from saturated condition. These are the tubular cross-flow heat exchangers. They also increases overall efficiency and turbine internal efficiency by reducing moisture content in the final stages of turbine. They required large heating surface area as more than 40% of total heat absorbed in the generation of steam are happend in superheaters. Their classification is according to their shape of tube banks and header position. And also according to either they absorbed heat by convection, radiation or combination of both. Following are the different types of superheater:

- **Pendant superheater or radiant superheater:** Pandant superheaters are supported or hanging from their headers like hanger superheater and waterwall hanger whereas radiant superheaters are located at top of the furnace and absobs heat by direct (luminous) radiation from the furnace like platen superheater. Hence, radiant superheaters are type of pandant superheaters.

- **Horizontal superheater or Convection superheater:** They are horizontal type with arrangement of tubes across the boiler and also they are self draining which is useful during lighting up hence these are used as primary superheater. They absorb heat mainly due to flow of hot gas around tubes. Example: Low temperature superheater, superheater cavity, superheater screen[9].

1.5.5 Reheaters

After loosing heat energy in the high pressure turbine, the exhaust steam comes into the reheater where temperature of steam increases and goes to the intermediate pressure turbine for further expansion. Hence, the main aim of reheating is to add energy to the partially used steam. Reheaters increases the capacity of power plant, reduces corrosion and steam consumption of steam turbine. Their construction and arrangement are alike superheater[9].

1.5.6 Economiser

Economizer are used in a steam generating unit to add sensible heat to the feed water by absorbing heat from the flue gas before the water enters into the evaporater of boiler. Hence, it is a heat exchanger which increases feed water temperature leaving the highest pressure feedwater heater to saturation temperature corresponding to boiler pressure. It is located at ahead of airheaters and following low temperature superheater and reheater, hence they all take place in same casing. In this, water is flow from bottom to top so that if steam is formed during heat transfer can go along with water hence overheating and failure of economizer tube can be neglected[9].

1.6 Circulation of water in boiler

In natural and controlled circulation, water is circulated through the steam generating surface from the drum and back to the drum after seperation of steam. Through the downcomers, water leaves the drum where temeprature is slightly less than saturation temperature. The water flow in the water wall is at saturation temperature whereas heat absorbed by water in water walls is latent heat of vapourization which forms mixture of water and steam. Hence, circulation ratio is defined as ratio of weight of water and weight of steam of the mixture leaving the heat absorption surfaces.

1.7 Gas flow path in boiler

The mixture of fuel and hot air drawn into the furnace where combustion will take place hence flue gas produced having high luminosity goes to panel and platen superheater. Due to direct contact of luminosity, the radiation effect is also high in the platen superheater. After it flue gas drawn on to the front and rear reheater and then goes to low temperature superheater and economizer. Figure 1.5 shows block diagram of typical arrangement of heating section of boiler where gas and steam path are clearly shown.

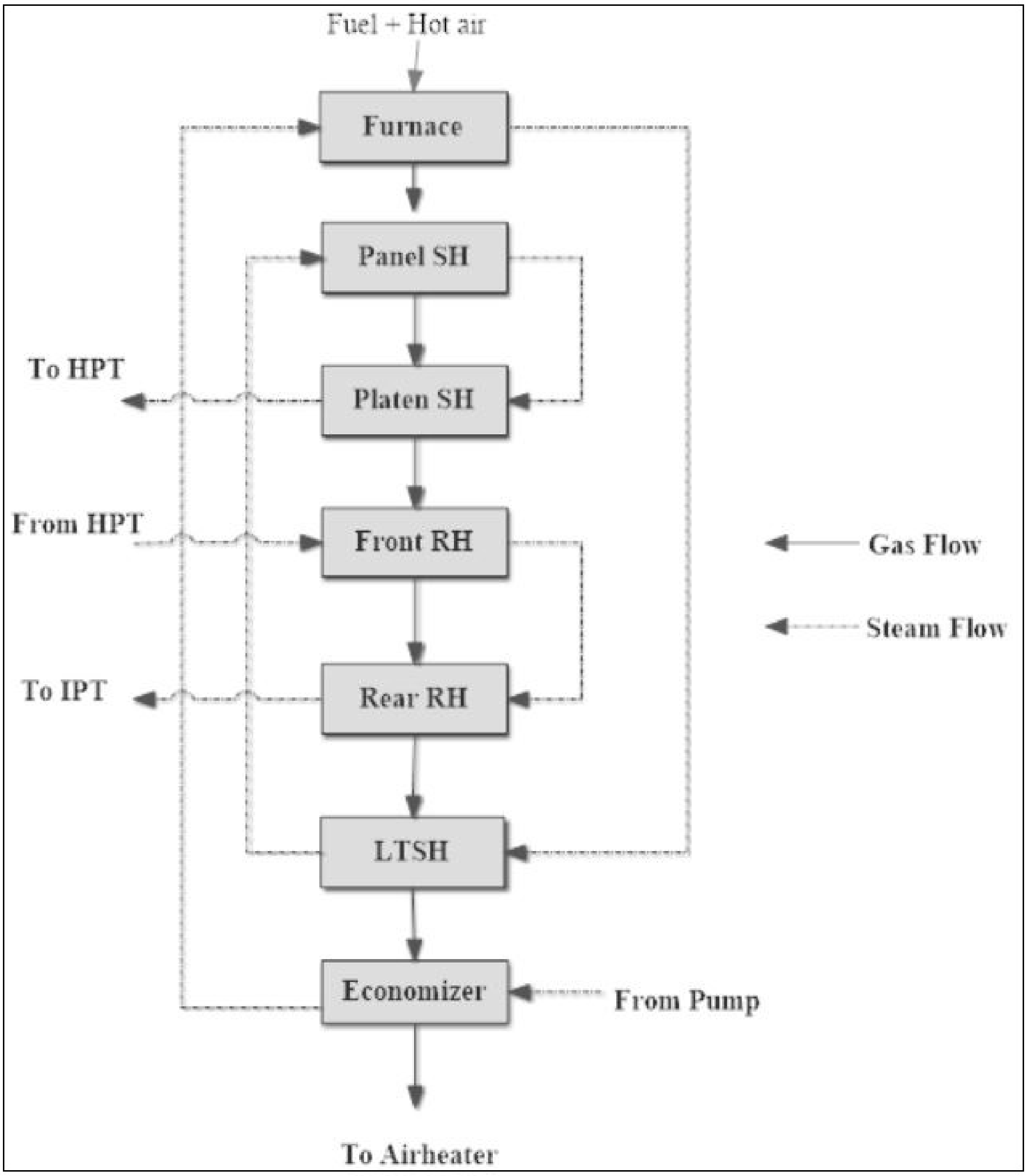


Figure 1.5: A Typical arrangement of boiler heating sections

1.8 Steam flow path in boiler

In the furnace, flue gas heats the water present in the water walls to form vapour. The vapour goes to the steam drum, where separation of steam and water will take place. From the downcomers, the water comes down. The steam present in the drum is saturated steam which cannot rotate turbine so first it become superheated by the help of several heat exchangers like superheaters and reheaters.

Hence, after drum the steam goes to the low temperature superheater where small increase of temperature will take place. Then it goes through the panel and platen superheater where steam become further superheated and goes to the high pressure turbine. From the high pressure turbine, the steam sent to the front and rear reheaters section and then sent to intermediate pressure turbine. After it, the reheated steam can rotate the turbine and produced work by shaft connected to generator. The spent steam from the low pressure turbine sent to the condenser, where steam gets condensed and form water. By the help of pump and deaerator the water sent to the economizer where preheating of boiler feed water will take place before going to the steam drum. Figure 1.6 gives detailed flow chart for steam path in the boiler.

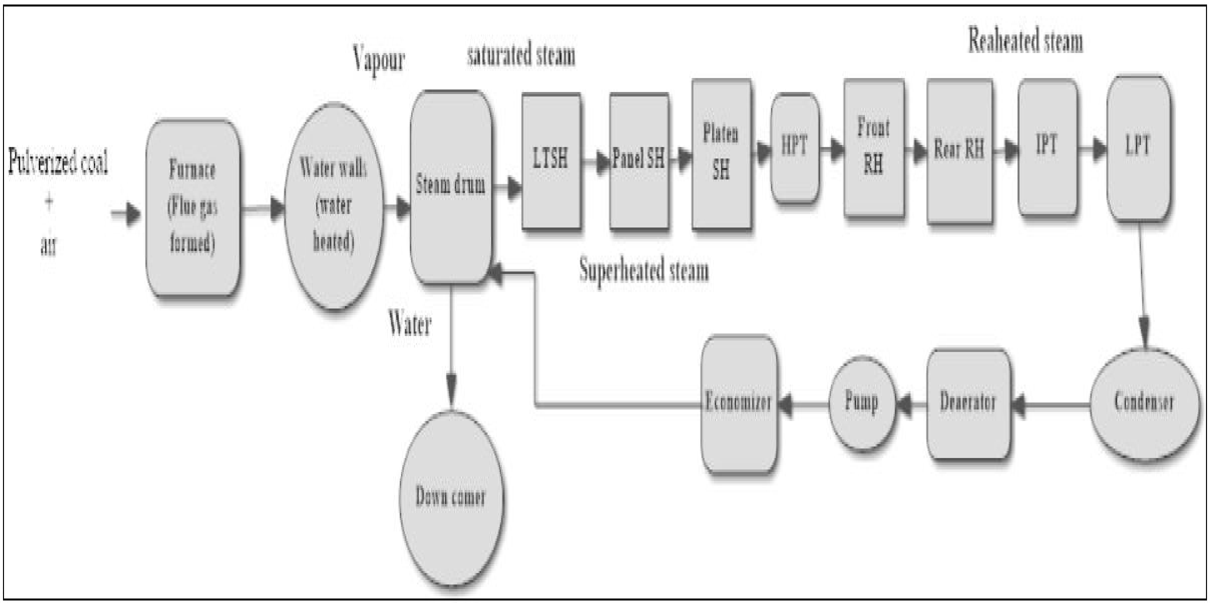


Figure 1.6: Flow of steam through all components of boiler

1.9 Turbine casing

It is modeled as cylinderical pipe inside which flow of steam is take place. Inside the casing there is a shaft on which moving blades are attached wheres the static blades are at inside surface of casing. Casing is surrounded by insulation to minimize the heat loss. Broadly turbine casing are classified in three catogories i.e. high pressure casing, intermediate pressure casing and low pressure casing.

- **High pressure casing:** For good sealing against steam leakage, the top and bottom halves of casing are linked together at flange joint with the help of heat tightened studs whereas on the nozzle box, four steam chests i.e. two on top and two on sides are welded. For circulate steam flow to the turbine according to requirement of load, four control valves are fitted to steam chests[9].

- **Intermediate pressure casing:** They are made in two parts i.e front and exhaust part and they connected by vertical joint. Both the parts containing two halves having horizontal joint which is protect by help of studs and nuts. On the casing itself the control valves are take place whereas nozzle boxes are cast with the casing. Steam is going through the cross over pipes into the double flow low pressure cylinder from the casing[9].

- **Low pressure casing:** In this, there are three parts i.e. one middle part and two exhaust parts. By the help of vertical flange, the exhaust and middle casing are bolted. Through the turbine centre line, casins are seperated in the horizontal plane. From the top, steam comes into the middle casing and seperates into two equal, axial and opposed flows and pass through four stages and then steam expands below to the condenser pressure whereas through the last stages remaining steam is expands[9].

Chapter 2

Literature Review

2.1 General

In this chapter literature pertaining to analysis of thermal power plant components is presented.

2.2 Literature review

Gomez et al. [11] have done the mathematical modeling and simulation in convective zone of power generation boiler for calculate shell side, tube side thermal field and tube wall temperature. For shell side continuity, momentum and enthalpy equations are solved whereas for tube side only enthalpy conservation equation are solved and for tube wall only energy conservation equation is solved. Then CFD simulation is done in which individual tubes are considered as sub-grid features and validated by comparing with simple heat exchanger geometries which can be solved with analytical methods then applied to an actual 350 MW power station boiler. Figure 2.1 described convective zone of boiler.

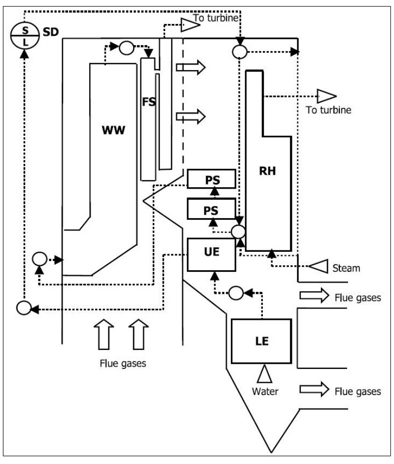


Figure 2.1: Schematic of the simulated convective zone

Mudafale [12] presented mathematical modelling of reheater section. Because in power plants, the major losses are take places in heat exchangers like reheater and superheater. In design of Reheater, spacing of tubes, fuel gas velocity, steam temperature control and material selection are important parameters. Material used by BHEL in reheater system are ASTM specification. Model prepared is based on conservation of mass, momentum and energy principles and explain fundamental physical process that determine interactions among the input variables like steam temperature, inlet mass flow rate, inlet flue gas temperature and mass flow rate of flue gas whereas output variables are various thermal parameters like main steam pressure, temperature etc., so this mathematical model of reheater can be helpful to reduce the faults.

Li et al. [13] prepared computer program on drum boiler start-up simulation programme (DBSSP) for simulate start up process for controlled circulation and natural circulation boilers in which mathematical model is developed on the principle of mass, energy and momentum conservation equations. Lump Model is used for both drum and downcomer. For controlled circulation boiler, downcomer flow rate(W_{DC}) is determined by balance between driving force of circulation pump, driving head of natural circulation and pressure drop of loop.

$$\frac{dW_{DC}}{dt} = \left(\frac{\rho d_c g H_{DC} - \int_0^{Lr} \rho r g Sin\theta dz - \triangle P_r - \triangle P_{DC} + \triangle P_p}{L_{DC}/A_{DC}} \right) \tag{2.1}$$

Whereas for natural circulation boiler, there is no circulation pump on downcomer.

$$\frac{dW_{DC}}{dt} = \left(\frac{\rho d_c g H_{DC} - \int_0^{Lr} \rho r g Sin\theta dz - \triangle P_r - \triangle P_{DC}}{L_{DC}/A_{DC}} \right) \tag{2.2}$$

Also, in downcomer, pressure loss due to friction are calculated, whereas by calculating mass conservation and energy equation and apply to each subsection of riser, enthalpy of each subsection is determined. Similarly in drum, drum pressure, mass balance and energy equations are determined in both liquid and steam zone. Then the programme DBSSP is applied to simulate the cold start up process of 600MW controlled circulation boiler and obtained result compare to their simulation results, so that with this programme minimum start up time can be obtained.

Trojan and Taler [14] investigated model for superheater taking into account the process occurs on the side of steam and flue gas. Physical properties of fluid are functions of temperature, axial heat conduction in tube wall and fluid heat conduction in fluid flow direction is negligible, temperature and flue gas velocity are constant over channel cross section before superheater and heat transfer coefficient on inner and outer tube surface are uniform. Then, heat balance equations for steam, flue gas, tube wall covered with ash deposit are solved.

The convection and radiation heat transfer coefficient are also calculated by equation:

$$h_t = h_c + h_n \tag{2.3}$$

Where,

$$Nu_c = {}^{h_c(d_o+2\delta_a)}/_{K_t}$$

$$h_n = \sigma \frac{1+\varepsilon_w}{2}.\varepsilon_t \left(\frac{T_g^4 - T_w^4}{T_g - T_w} \right)$$

So the method is used for modelling operation of superheater under various operating conditions like ash fouling on flue gas side, scale deposit on inner surface of tube and non-uniformity of flue gas temperature distribution over duct cross section, changes in superheater construction, also this method allows calculation of temperature of tube wall not only in parallel or counter flow heat exchanger but also in mixed flow heat exchanger. Figure 2.2 gives detailed information about coal fired utility boiler including natural circulation.

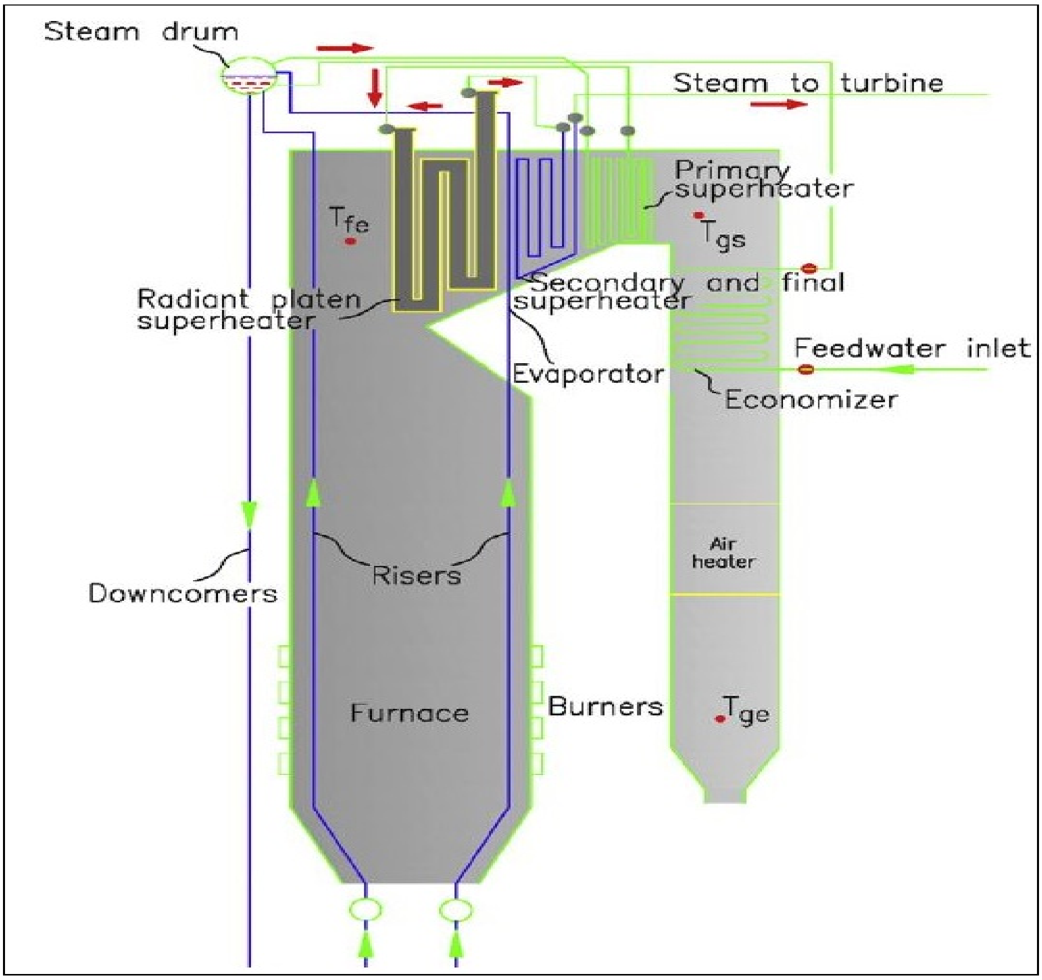


Figure 2.2: Coal-fired utility boiler with natural circulation

Krishna and Singh [15] have done modeling to calculate heat flux by least square approach for unsteady state heat transfer over gas turbine wall, steel reheat furnace and ceramic kiln by assuming flat plate having thermal properties are constant with no heat generation within metal. For this, the estimated temperature match the measured temperature as closely as possible over a specified time domain, this estimated temperature are computed from the solution of direct problem by using the estimated heat flux components, where measured temperature are recorded using a sensor and to ensure optimal matching between measured and estimated temperature, we require that the least square norm is minimized with respect to each of unknown heat flux components. Using MATLAB, solution of least square approach is solved for wall of high pressure gas turbine. Equation for cold side of wall as function of time,

$$T_{cool} = \left(\frac{T_{s,s} - T_{hot}}{t_{final}} \right) t + T_{hot} \tag{2.4}$$

The steady state temperature reached after 60sec. and temperature measurement were taken at four evenly spaced time of 15 sec. i.e. 15,30,45,60 sec., so using these conditions of starting and ending times, temperature, sensor measurement, MATLAB programme produced surface heat flux. With continuous monitoring of temperature distribution and the properties of material, heat flux corresponding to temperature change could be obtained and thereby heat transfer coefficient at every instant can be estimated. Figure 2.3 shows variation of heat flux with respect to the time for gas turbine and the compare to the actual heat flux.

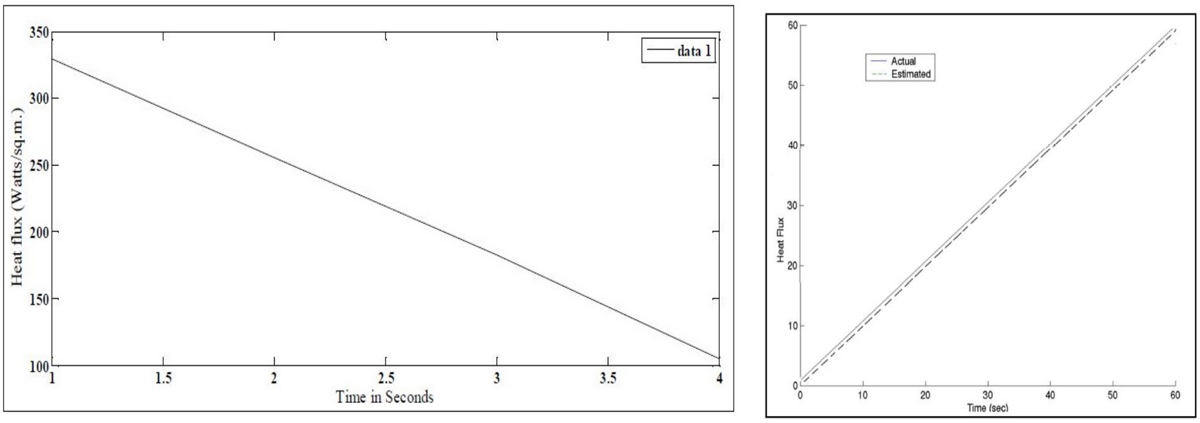


Figure 2.3: Change of heat flux for gas turbine and comparison of actual and estimated

Madajski et al. [16] developed a numerical model of steam superheater using control volume based finite element method. Due to high value of pressure and temperature in boiler, the superheater stages and individual passes are made of different law alloy steel grade. To select appropriate steel, the maximum wall temperature of tube need to be determined by the mathematical model in which, to determine wall temperature distribution in superheater tube, the control volume FEM were used. The methodology of CVFEM formulation is based on heat conduction equation.The energy conservation equation is solved at different nodes to calculate tube wall temperature by gauss seidal approach. The pressure distribution is calculated by mass conservation and momentum conservation equation using FDM. Then the model is applied to three different load of boiler 100,60 and 35% are carried out for steady state calculation and result obtained were compare with result of CFD simulation using star CCM+ to estimate accuracy of model based on FVM.

Haq et al. [17] presented mathematical model and simulation of industrial steam boiler which describing the thermodynamic process, which take place in furnace using mass, energy and momentum balance and simulate this model using SIMULINK which concludes that pressure inside furnace increase linearly and reaches to steady state due to variation of fuel flow inside boiler and due to law resistance of boiler. Figure 2.4 gives variation of drum pressure with respect to the time.

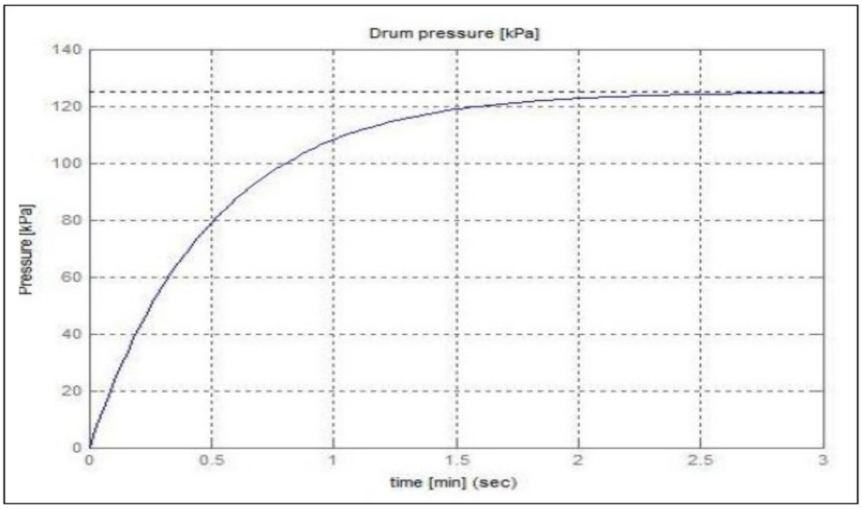


Figure 2.4: Variation of drum pressure with time

Chaibaksh and Ghaffari [18] carried out the mathematical model on steam turbine based on energy balance, thermodynamic principles and semi empirical equations to characterize transient dynamic of steam turbine. Then the relevant parameter of developed model is either determined by empirical relation or they are adjusted by applying genetic algorithm based on experimental data obtained from complete set field experiment. Dynamic high pressur turbine model is developed for which pressure, mass flow rate and temperature of steam at input and output of each section is developed. Reheater model is also developed based on thermodynamic principles and energy balance. Then the developed model of turbine is simulate using MATLAB-SIMULINK and comparison between responses of turbine-generator model with response of real system validate the accuracy of proposed model in steady state and transient condition.

Dulau and Bica [19] developed mathematical modeling for behaviour of steam turbine in which many intermediate variables are neglected and focus only on input variables to output variables, the model is based on continuity equation. Model is used to determine simulation diagram for steam turbine with high, medium and law pressure stages. Also using MATLAB-SIMULINK, the behaviour of output variables are determined like shaft torque and depending on control valve position high pressur valve and low pressure valve as input variable with uncertain parameters of process.

Bracco [20] prepared simulation model of steam drum based on heat transfer equation for calculating the temperature and thermomechanical stress distribution inside the metal and insulation of high pressure steam drum by the help of Fourier heat conduction equation in cylindrical symmetry.

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} = \frac{1}{\alpha}\frac{\partial T}{\partial r} \tag{2.5}$$

Analytically solve steam drum equation under steady state and transient condition using laplace equation and separation variable technique respectively and obtained numerical solution by finite difference backward and forward method, so resultant discretized equation is,

$$\frac{T_i^{t+\triangle t}}{\partial r} = \frac{a}{\triangle r^2}\left[T_{i+1}^t\left(1+\frac{\triangle r}{r_1}\right) - T_i^t\left(2+\frac{\triangle r}{r_i}\right) + T_{i-1}^t\right] \tag{2.6}$$

Then two simulation model are present for steam drum, both are based on discretized fourier equation. Both model differ in structure in SIMULINK i.e. first model SIM_CC has static block structure while SIM_CC_m is more flexible because it has been developed the discretized fourier equation in vectorial form and validate under steady state and transient condition. Figure 2.5 shows variation of metal temperature with respect to time at transient condition.

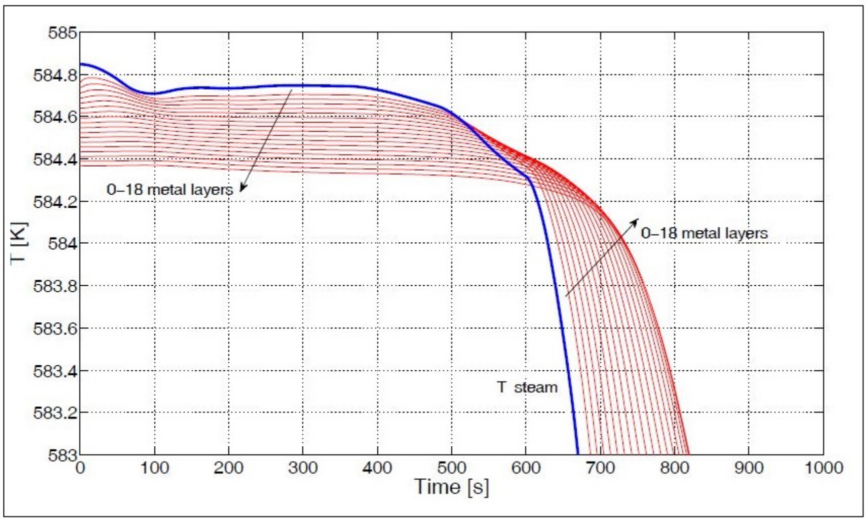


Figure 2.5: Variation of metal temperature at transient state.

Basu [21] investigated modeling approach to superheater of boiler using mass, energy and momentum balance equation. The nonlinearity in the equation is eliminated using Taylor series expansion and dynamic model based on obtained parameters have been made using MATLAB. Figure 2.6 gives time response of specific enthalpy at superheter outlet and for flue gas whereas figure 2.7 shows time response of superheater mean temperature.

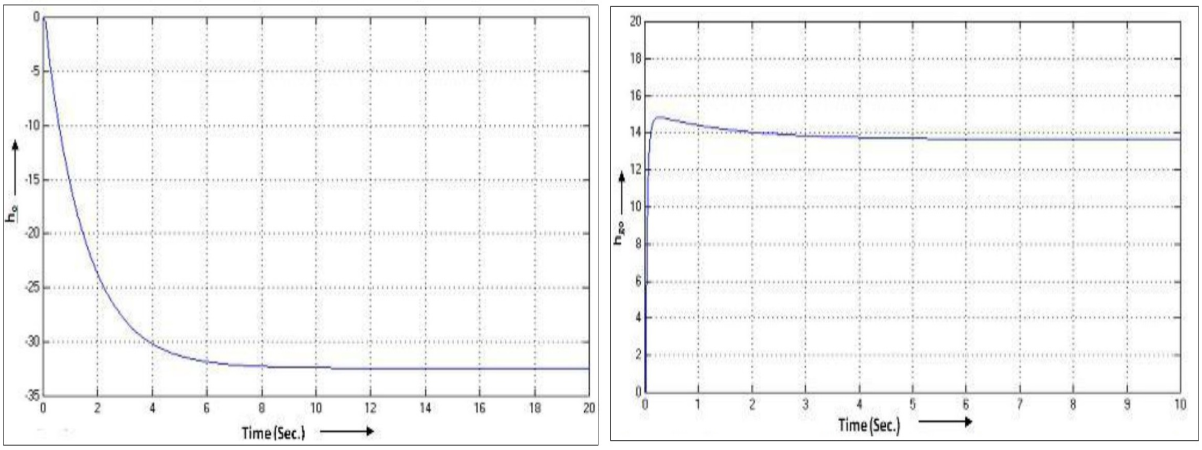


Figure 2.6: Time response of specific enthalpy at superheater outlet and for flue gas

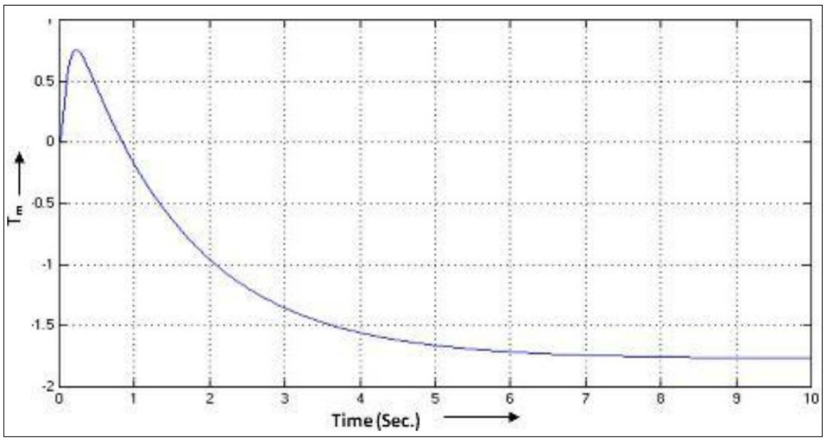


Figure 2.7: Time response of superheater mean temperature

Zima et al. [22] have done mathematical model with distributed parameters based on mass, momentum and energy conservation equation for simulation of heat flow phenomena occurring in water wall tubes of boiler for super critical parameters and for solving these equations, two methods are proposed, one is by implicit difference method and other is Runga Kutta method. By first one, on the basis of determined pressure and enthalpy distribution, fluid density and temperature are evaluated. The second approach to solve conservation equations by Runga Kutta method. Both method should follow this condition,

$$\triangle\tau \leq \frac{\triangle z}{w} \tag{2.7}$$

To verify these methods, the calculation is presented in a lignite coal fired boiler. The distribution of water wall pipe temperature was determined by the help of CFD and by comparing the result obtained from two methods and CFD analysis, a satisfactory convergence was found.

PurbolakSono et al. [23] suggested method for estimating heat flux in superheater and reheater tubes because it is important parameter to find out efficiency of steam generating tubes, safety of tubes, boiler tube failures. So by using empirical formula correlating scale thickness with Larson-miller parameter, the finite element modeling is proposed.

$$\log x = 0.00022P - 7.25 \tag{2.8}$$

$$T\,(C + \log t) = P \tag{2.9}$$

Where, $x = scale\,thickness\,in\,mils$, $t = rupture\,time\,(hr)$, $P = Larson\,miller\,parameter$, $T = absolute\,temperature\,in\,degree\,Rankine$ and $C = constant = 20$. Heat flux as both temperature and scale thickness increases over period of time was determine using iterative procedure. Figure 2.8 shows the heat flux and temperature variation with respect to the time for different tube tube geometries and mass flow rate.

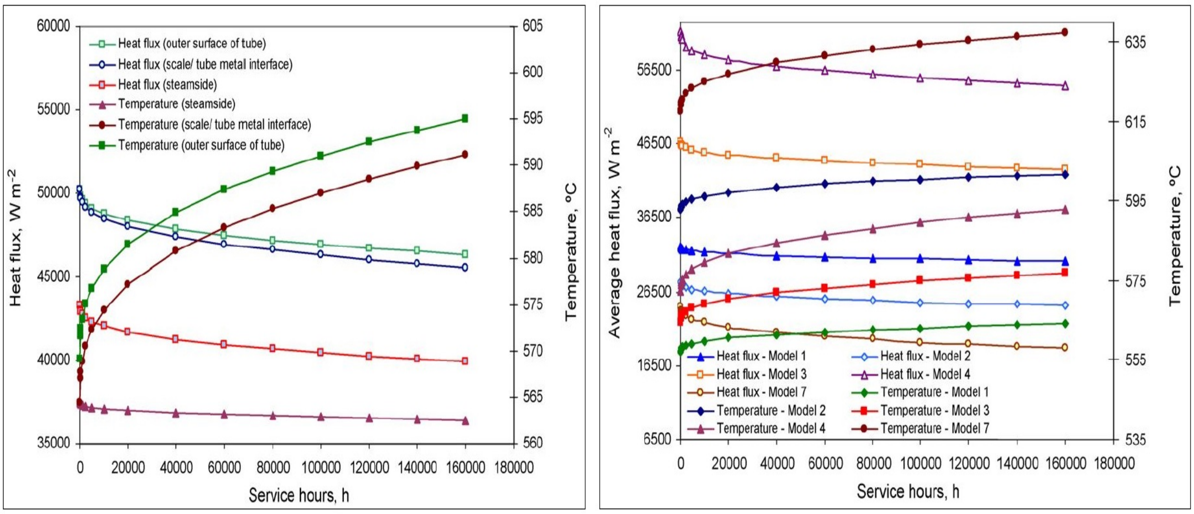


Figure 2.8: Heat flux and temperature variation with time for different tube geometries and mass flow rate.

Tzolakis et al. [24] have done mathematical modeling for steady state simulation and overall thermal efficiency of 300 MW lignite fired power plant in Greece. In flue gas circuit, mass and energy balance equation were applied for all components and concerning the furnace and water wall, radiation and convection are the heat transfer mechanism. Similar equations are used for superheater and reheater. Since the input fluid is a two phase fluid and temperature is also known, hence by using saturated enthalpies, the two mass flow rates can be calculated with combination of mass and energy balance. In the economizer, to calculate outlet condition for flue gases and water the energy balance, mass balance and heat transfer equation are used. The overall efficiency calculated as,

$$\eta_{total} = \eta_{thermal} * \eta \tag{2.10}$$

Then by using foreign objects ($gPROMS$) number of variables, equations and parameters of model were reduced. As a result, runtime of simulation was reduced. By the coupling of two circuit i.e. water-steam circuit and flue gas circuit gave satisfactory result with error not exceeding 1% and with minimal impact to efficiency of plant and we optimize its operation to improve the efficiency while maintaining same electric power production, which shows absolute improvement of 0.55% of overall thermal efficiency. Figure 2.9 compares the efficiencies of various circuits with overall efficiency for 300 MW plant.

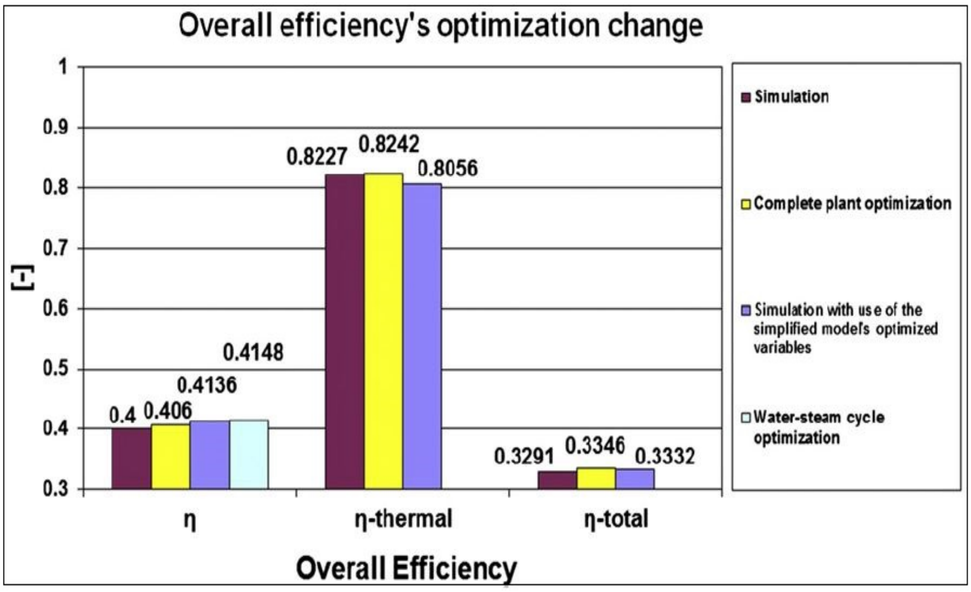


Figure 2.9: Comparison of the efficiencies of various circuits and complete plant efficiency

Makovicka et al. [25] presented mathematical modelling of steam and flue gas flow in heat exchanger of steam boiler by the help of Euler equation in conservative form in which we consider thermodynamic relation for enthalpy, continuity, energy and momentum equation. Then numerical solution of system, pipe wall model and thermal transfer and model of injection cooler suggest variant of finite volume scheme which is cell centred in mass flow rate and then comparison with theoretically computed temperature profile for stationary state in heat exchanger by solving system of ordinary differential equation.

Ludowski et al. [26] carried out CFD simulation and numerical model to determine the temperature distribution of steam and tube wall of superheater tubes in which equation of conservation mass, momentum and energy are solved using element based FVM and combination of Wilcox, $k-\omega$ and $k-\varepsilon$ turbulence models. Using ANSYS/CFX velocity, pressure and temperature of steam and tube wall is computed in which finite element number ranged from 790,179 to 3789,699 and mesh consisting of 1233,999 elements. Inlet velocities,$U_{in} = 16.7m/s$,where gas side heat transfer coefficient α_g= $200w/(m^2k)$ and for mesh consisting of 1233,999, following values of heat flow rate were obtained,$Q_{out} = Q_{in} = 57.266KW$, Relative difference obtained is (-0.656), So if number of element increases to 3789,699, the relative difference is almost the same and equal to -0.656. After it two inverse problems are described. In first one heat transfer coefficient on flue gas side was determine based on measured steam temperature at inlet and outlet of three pass steam superheater and solve using secant method. Whereas in second inverse problem, it is solved by Lavenberg-Marquardt method. And at every iteration step, a direct conjugate heat transfer problem was solved using ANSYS/CFX software. Figures 2.10 shows temperature distribution in the tube wall and steam in two cross section for $U_{in} = 16.7m/s$ and $\alpha_g = 200w/(m^2k)$.

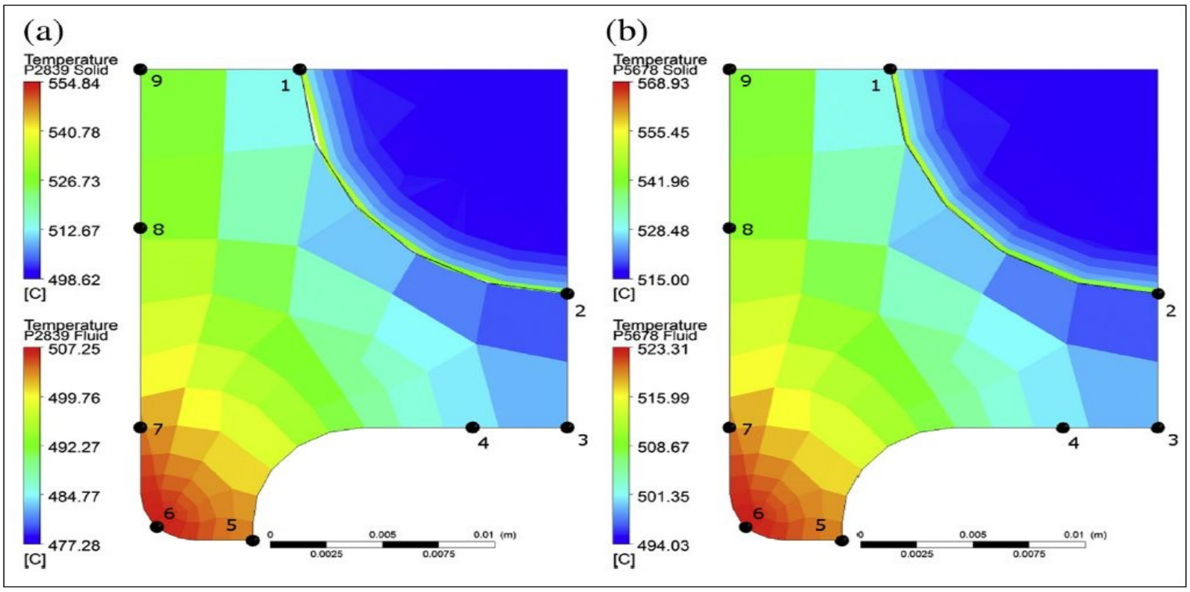


Figure 2.10: Temperature distribution at distance of 2.8 m and 5.6 m from inlet.

2.3 Conclusions from literature review

The conclusions made from the literature review are as under:

- For the analysis of thermal power plant components, many researcher have done the mathematical modeling and deveoped many equations.
- These equations are based on the mass, momentum and energy conservation phenomena for calculating relative behaviour of temperatures, heat transfer coefficients inside and outside surfaces of the components.

- For simulation, these differential equations are solved using different tools like finite volume method, finite element method, MATLAB, computational fluid dynamics (CFD) by different researches.
- These methods when applied in actual power plant take lot of time for analysing heat transfer processes.
- The present dynamic simulation methodology has capability to give results in real time mode.

2.4 Objectives of the project

The objectives of the present study are as under:

- To carry out heat transfer analysis and wall temperature estimation of various components of thermal power plant like superheater, reheater, economizer tubes, boiler drum and turbine casing.
- To estimate the heat losses for generic thick walled components.
- To validate the simulation results with the actual data.

2.5 Methodology adopted

The methodology applied to achieve required goals is described below:

- Convective (forced) heat transfer coefficients, non-luminous heat transfer coefficients and heat transfer will be calculated using correlations available in the literature. Appropriate amount of radiation as input was also considered for the applicable sections.
- Required fluid and material property library was used to evaluate the properties like thermal conductivity, dynamic viscosity, density etc.
- The dynamic and algebraic equations was formulated for a given problem. It is based on mass balance, heat balance and heat transfer models.
- The steady state and dynamic equations thus formed are solved using Excel based solver or ACSL-X solver.
- For all the components listed in the scope of work, one each case study was carried out.

Chapter 3

Mathematical Modeling of Boiler Components and Turbine Casing

3.1 General

This boiler section mainly consists of superheaters, reheaters, economiser and boiler drum. These components are constructed by a suitable sub models in the area of non-luminous gas radiation and convective heat transfer. In this chapter, the correlation are used for calculating all the parameters like convective and non luminous heat transfer coefficients, inside heat transfer coefficient through steam side, heat transfer rate through steam and gas side, calculation of wall/metal temperature and also LMTD calculation are shown from open soures. Similarly the same approach is applied for turbine casing also.

3.2 Tube banks

Inside the superheater, reheater and economizer, there are several assymblies and in each assymbly, there are several cylinderical tubes arranged in parallel arrangement. These tubes are as important heat transfer surface used in boiler for steam generation in which fluid flow normal to them so that they are heated. There arrangement either in-line or staggered in the direction of flow. In in-line arrangement, without displacement in cross flow direction, the column of tubes is placed exactly behind the next adjacent column along the stream wise direction whereas in case of staggered every second column of tube is displaced in the direction of flow. In our work, we consider the arrangement of tubes is in-line. The arrangement is characterized by the help of transverse and longitudinal pitches. Heat transfer related with tube is reliant on its position in the bank. Hence, for determining the required design, performance and operating parameters, heat transfer and flow characteristics must be required at every tube position[10]. At outside the tube, the gas flow will take place whereas inside there is flow of steam. Due to gas film there is some temperature drop at outside the tube wall, after it the temperature drop will take place due to the tube wall. Inside the tube wall there is a film formation due steam hence again temperature drop will take due to steam film. Figure 3.1 gives heat transfer philosophy through heat exchangers tubes.

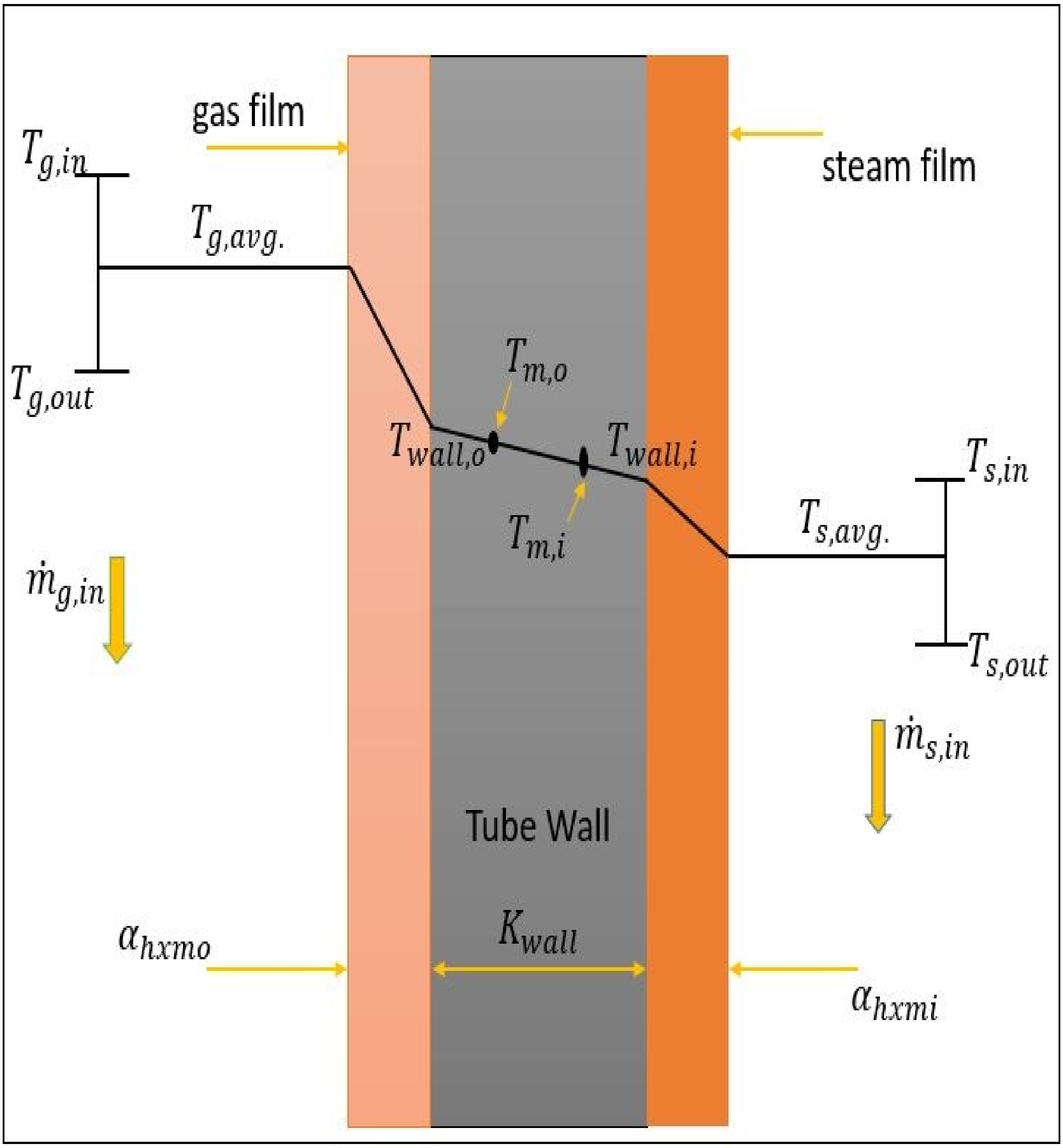


Figure 3.1: Illustration of heat transfer through tube wall

3.3 Conduction through tube wall

The implementation approach and mathematical equation used for conduction process occurs in the tube wall are described below:

3.3.1 Implementation approach

Calculation steps involved in conduction through tube wall, is illustrated in figure 3.2.

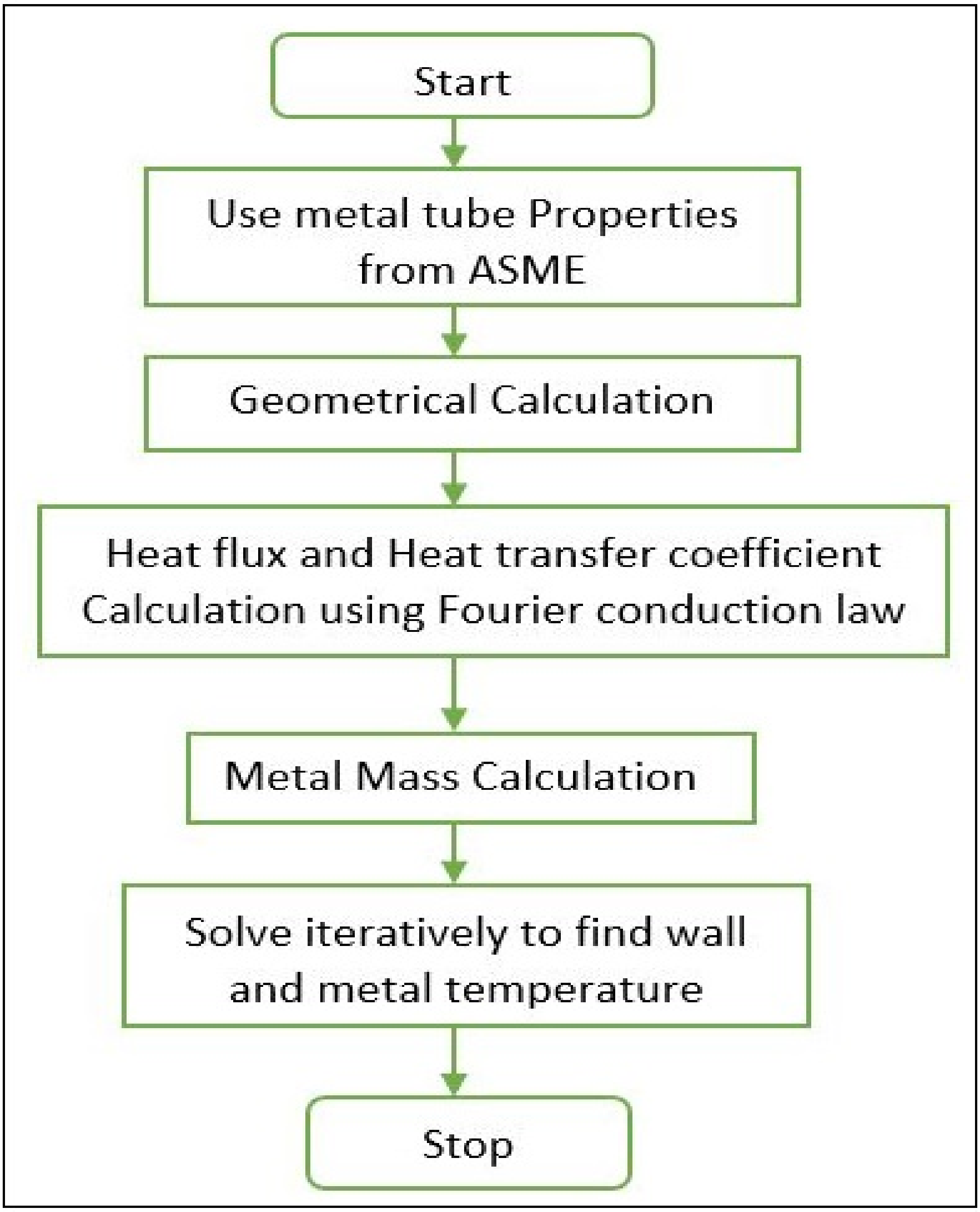


Figure 3.2: Implementation approach for conduction

According to fourier law of heat conduction in the tube wall,

$$Q_{metal} = -KA\frac{dT}{dx}$$

$$Q_{metal} = -KA\frac{(T_o - T_i)}{thickness/1000} \tag{3.1}$$

For tubes, overall heat transfer correlation is given by:

$$Q = UA\triangle T_m \tag{3.2}$$

$$U = \frac{1}{h_o} + F_{f,o} + \frac{D}{24K} ln\frac{D}{d} + \frac{1}{h_i}\frac{D}{d} + F_{f,i}\frac{D}{d}$$

Where,

$D = Outer\ tube\ diameter.$

$d = Inside\ tube\ diameter.$
$F_{f,o} = Outside\ film\ fouling.$
$F_{f,i} = Inside\ film\ fouling.$
$K = Thermal\ conductivity.$
Total overall heat transfer coefficien$'U$' is the summation of inside and outside heat transfer coefficient

3.4 Convective heat transfer modeling

Convection is a process which is happening due to motion of fluid. In which, cold fluid takes heat from the hot surfaces. When fluid motion occurs naturally, called free natural convection and when fluid motion occurs due to external force called forced convection heat transfer. Both processes occur at different rates, the forced convection is being the more common and rapid process. In the boiler, the forced convection heat transfer mode is predominant. Inside the tube, convective heat transfer taking place between the steam and the tube wall and calculated by required properties of steam whereas outside the tube, convective heat transfer taking place between the gas and tube wall. The gas properties are formulated based on the information available from open sources.

3.4.1 Implementation approach

The calculation steps involved in both convective heat transfer and non-luminous heat transfer modelling, are illustrated in figure 3.3.

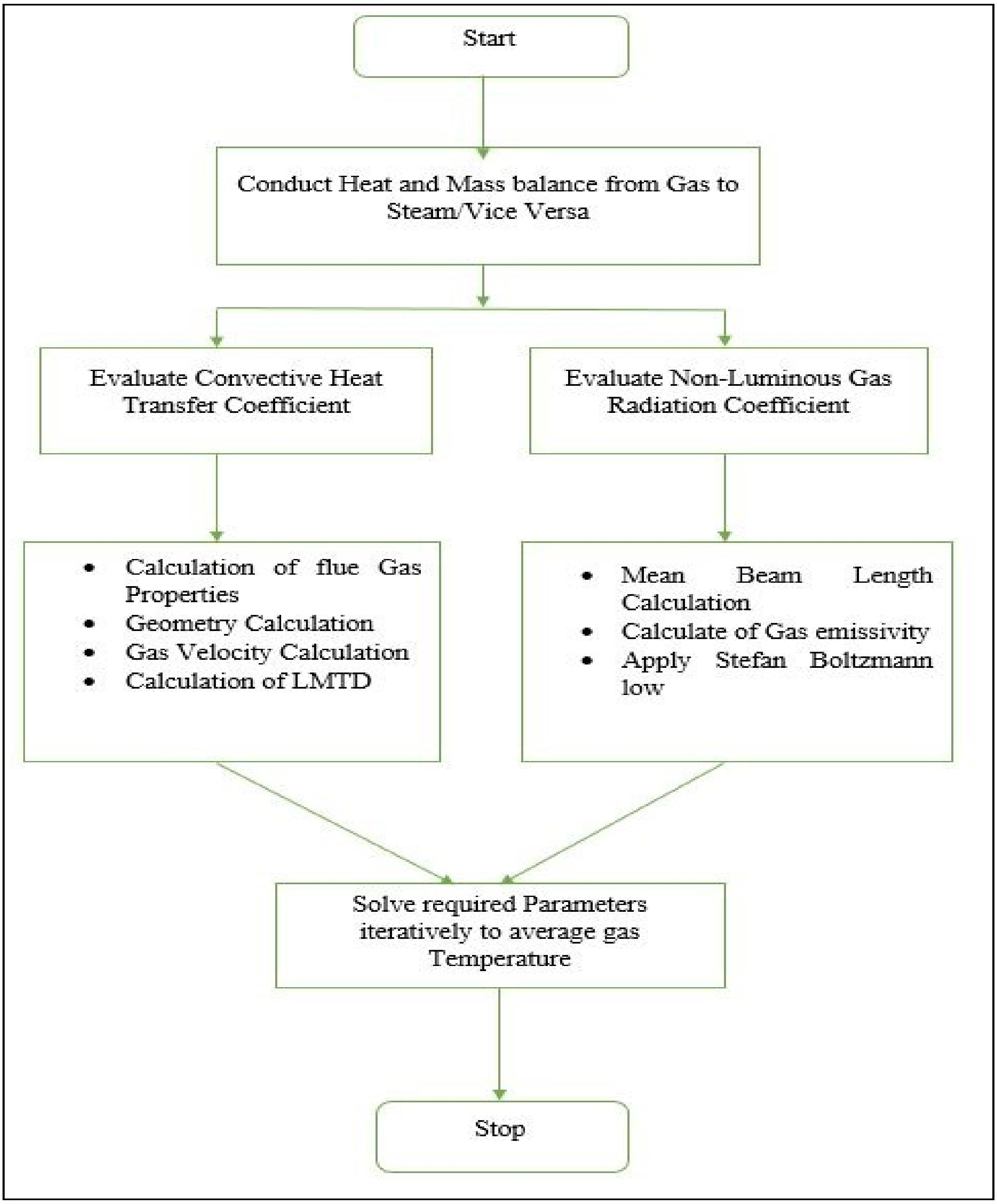


Figure 3.3: Implementation approach for convective and non-luminous heat transfer

3.4.2 Convective heat transfer coefficient through gas side

We know that, total heat transfer coefficient changes with the depth of tube bank hence if banks' having ten or more rows in depth and flow is normal to tubes then [10],

$$Nu = 0.33 F_H Re^{0.6} Pr^{0.33} \tag{3.3}$$

Where, $F_H$is factor which is depend on the arrangement of tubes in the bank.
If the banks are less than ten rows in depth then another correction factor $F_N$is added. So that,

$$Nu = 0.33 F_H F_N Re^{0.6} Pr^{0.33} \tag{3.4}$$

We know that,

$$h_c^{'} = \frac{Nu.K}{D}$$

Or,

$$h_c^{'} = 0.33 Re^{0.6} Pr^{0.33} \frac{K}{D} \tag{3.5}$$

Therefore,

$$h_c = 0.33 F_H F_N Re^{0.6} Pr^{0.33} \frac{K}{D} \tag{3.6}$$

Hence,

$$h_c = h_c^{'} F_H F_N \tag{3.7}$$

Rate of heat transfer is also depend on the angle of gas flow with the tubes so another factor $'F_A'$ is takes place. Hence,

$$h_c = h_c^{'} F_H F_N F_A \tag{3.8}$$

Where,
$F_H = factor\, depends\; on\; tube\; arrangement.$
$F_N = factor\; depends\; on\; number\; of\; rows.$
$F_A = factor\; depends\; on\; angle\; of\; gas\; flow.$
$h_c^{'} = Basic\; convection\; heat\; transfer\; coefficient.$
$h_c = overall\; convection\; heat\; transfer\; coefficient\; for\; clean\; tubes.$

3.4.3 Non-luminous heat transfer coefficient through gas side

Radiation through gas side is take place due to luminosity and radiant properties of flue gas. Hence, luminous radiation means direct contact of flue gas to heat exchanger or boiler components whereas non-luminous gas radiation means, there is no direct contact of luminosity of flame gas to the tube wall of heat exchangers. It happens due to radiant properties of flue gas due to carbon dioxide, water and sulphur dioxide present in the combustion of gas. Also, emissivity is the major factor of radiation. Partial pressure exerted by individual gases and the gas layer thickness or we can say beam length are other factors which affect properties of radiation and gas temperature, tube surface and tube wall emissivity are also important factors which included in our calculation. Hottel and Sharan proposed their method to calculate non-luminous radiation heat transfer coefficient[10].

$Hottel's\ method,$

$$h_n = \sigma\varepsilon\left({\left(T_g^4 - T_{w,o}^4\right)}/{\left(T_g - T_{w,o}\right)}\right) \quad {W}/{m^2k^2} \tag{3.9}$$

$$\sigma = 5.67 * 10^{-8} \quad {W}/{m^2k^2}$$

And also

$$\varepsilon = 0.9\left(1 - exp\left(-kL\right)\right)$$

$$L = 1.08 * \left({\left(S_T S_L - 0.785D^2\right)}/{D}\right) \quad 'm'$$

$$k = {(0.8+1.6P_w)(1-0.38T_g/1000)(P_C+P_W)}/{\sqrt{(P_C+P_W)*L}}$$

Where,
$h_n = Basic\ non-luminous\ radiation\ heat\ transfer\ coefficient.$
$\sigma = Stephan\ Boltzmann\ constant.$
$L = Radiant\ beam\ length.$
$S_T = Transverse\ pitch.$
$S_L = Longitudinal\ pitch.$
$P_C = Partial\ pressure\ exerted\ by\ CO_2.$
$P_W = Partial\ pressure\ exerted\ by\ tube\ wall.$

$Sharan\ method,$
According to this method[10], non-luminous heat transfer coefficient through $H_2O + CO_2$ at different temperature, beam length and partial pressure is calculated as:

Case 1: When the total pressure is one atmospheric, tube emissivity is 1 and temperature is 300°C.

$$h_n' = P_x\left[1 + \frac{T_g - 2192}{2068.8 - 170.4\log_{10}X}\right] \tag{3.10}$$

Where,

$$X = L\left[\left(P_C + P_H\right)P_H\right]^{0.5}$$

And

$$L = ((1.082 * {}^{S_T}/_{D} * {}^{S_L}/_{D}) - 0.85) * ({}^{D\ in\ inch}/_{12})$$

$$P_x = a_o + a_1 x + a_2 x^2 a_3 x^3 + a_4 x^4 + a_5 x^5$$

Where,

$$a_o = 2.1007079 \qquad a_3 = 3.39128$$

$$a_1 = 16.185383 \qquad a_4 = -0.538107$$

$$a_2 = -9.934423 \qquad a_5 = 0.031373$$

By substituting values, we obtained P_x is $11.44x^{0.50762} Btu/ft^2 hr°$F for $'x'$ is 0.12 to 0.8 $ft.atm.$ with 0.2 % accuracy.

Case 2: If temperature of tube surface also not equal to $300°C$, then

$$K_t = \left[\frac{T^4_{g,abs.} - T^4_{M,abs.}}{T^4_{g,abs.} - 1031^4}\right] * \left[\frac{T_g - 572}{T_g - T_M}\right] \quad for\ T_M \leq 300°C \tag{3.11}$$

$$K_t = \left[\frac{(4.90 * K_t \leq 572°\text{F}) - 4.82}{T_g^{0.25}}\right] + 1 \qquad for\ 300°C < T_M < 600°\text{C} \tag{3.12}$$

Also, if tube emissivity not equal to 1, then

$$K_e = \frac{E_w}{E_w + A_g + E_w A_g} \tag{3.13}$$

Wall emissivity $'E_w'$ for oxidized steel is 0.79 and for stainless steel is 0.9. So that, the non-luminous heat transfer coefficient from radiating gases to the clean surface of tube is,

$$h_n = h_n^{'} K_t K_e \tag{3.14}$$

Where,

$P_H = Partial\ pressure\ exerted\ by\ H_2O.$

$T_(g, abs) = Absolute\ mean\ gas\ temperature.$

$T_(M, abs) = Absolute\ outside\ metal\ temperature.$

$T_M = Outside\ metal\ temperature.$

$K_t = Factor\ associated\ in\ correction\ of\ h_N\ for\ wall\ temperature.$

$K_e = Factor\ associated\ in\ correction\ of\ h_N\ for\ gas\ absorptivity.$

$E_w = Tube\ wall\ emissivity.$
$A_g = CO_2\ and\ H_2O\ mixture\ absorptivity.$
$h_n^{'} = Non - luminous\ heat\ transfer\ coefficient\ clean\ tubes.$

3.4.4 Total heat transfer coefficient through gas side

Total heat transfer coefficient outside the tube is the summation of convective heat transfer coefficient and non-luminous gas radiation heat transfer coefficient through gas side[10], hence,

$$h_t = h_c + h_n \tag{3.15}$$

Where,
$h_c = convective\ heat\ transfer\ coefficient.$
$h_n = non - luminous\ heat\ transfer\ coefficient.$

3.4.5 Convective heat transfer coefficient through steam side flowing in a tube

Convective heat transfer taking place between the steam and the tube wall. Where convective heat transfer coefficient is calculated using the required properties of steam [10].

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \tag{3.16}$$

$$Re = \frac{\rho V D}{\mu}$$

$$Pr = \frac{\mu C_p}{K}$$

$$h_i = \frac{Nu.K}{D}\ in\ Kcal/hr.m^2k \tag{3.17}$$

Where Nu, Re, Pr are the Nusselt number, Reynolds number and Prandtl number.
And,
$\rho = density\ of\ fluid$
$V = velocity\ of\ fluid$
$\mu = viscosity\ of\ fluid$
$C_p = specific\ heat\ of\ fluid$
$h_i = heat\ transfer\ coefficient\ of\ fluid$

3.5 Heat transfer rate calculation

Due to convection, heat transfer take place through gas and steam side[27]. To see heat transfer philosophy refer figure 3.1.
Heat transfer through gas side,

$$Q_g = \dot{m}_g C_p (T_{g,in} - T_{g,out})$$

$$Q_g = \alpha_{hxmo} A_o (T_{g,avg.} - T_{w,o}) \tag{3.18}$$

Heat transfer through steam side,

$$Q_s = \dot{m}_s C_p (T_{s,out} - T_{s,in})$$

$$Q_s = \alpha_{hxmi} A_i (T_{w,i} - T_{s,avg.}) \tag{3.19}$$

Where, α_{hxmi} is the total heat transfer coefficient inside the tube which include combined heat transfer coefficient due to steam, inside scale over inside surface area. Whereas α_{hxmo} is the total heat transfer coefficient outside the tube which include combined heat transfer coefficient due to gas, outside scale over outside surface area.

$$\alpha_{hxmi} = f_s \alpha_{I,tube} A_i$$

$$\alpha_{hxmo} = f_f \alpha_{O,tube} A_o$$

Where,
combined heat transfer coefficient at inside ($\alpha_{I,tube}$) and outside ($\alpha_{O,tube}$) are as follows:

$$\frac{1}{\alpha_{I,tube}} = \frac{1}{h_i} + \frac{\delta_{r,i}}{K}$$

$$\frac{1}{\alpha_{O,tube}} = \frac{1}{h_t} + \frac{\delta_{r,o}}{K}$$

Also,
Inside surface area,

$$A_i = \pi(2 * r_i) * heated\ tube\ length$$

Outside surface area,

$$A_o = \pi(2 * r_o) * heated\ tube\ length$$

or,

$$Heated\ tube\ length = \left(\frac{heating\ surface\ area}{\pi D}\right) \tag{3.20}$$

Note: The calculation for $\delta_{r,i}$ and $\delta_{r,o}$ are in tube wall and metal temperatue calculation section.

Where,

$A_i = Inside\ heating\ surface\ area.$

$A_o = Outside\ heating\ surface\ area.$

$f_s = 1\ (detoriation\ of\ heat\ transfer\ because\ of\ inside\ scaling)$

$f_f = 1\ (detoriation\ of\ heat\ transfer\ because\ of\ outside\ caling)$

$\dot{m}_g = mass\ flow\ rate\ of\ gas.$

$\dot{m}_s = mass\ flow\ rate\ of\ steam.$

$K = thermal\ conductivity\ of\ wall\ material.$

3.6 Tube wall and metal temperature calculation

For inner and outer tube wall and metal temperature calculation, following procedure is adopted[27], where cross section and temperatures of the tube are illustrated in the following figure 3.4.

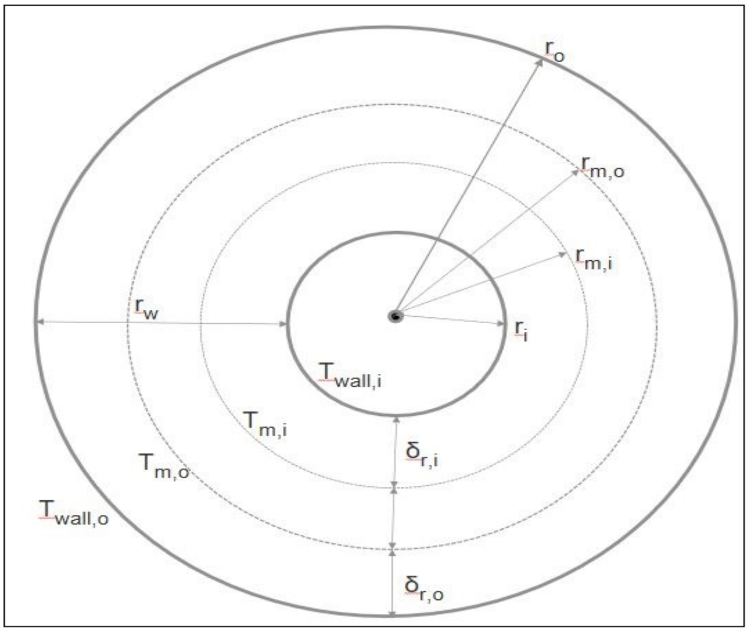


Figure 3.4: Illustration of cross section and wall/metal temperatures of tube[27]

Depth of inner zone,

$$\delta_{r,i} = r_{m,i} - r_i$$

Depth of outer zone,

$$\delta_{r,o} = r_o - r_{m,o}$$

Inner zone centre radius,

$$r_{m,i} = r_i + \left[\theta * \left(\frac{r_o - r_i}{2}\right)\right] \quad 0 < \theta < 1 \tag{3.21}$$

Outer zone centre radius,

$$r_{m,o} = r_o - \left[\theta * \left(\frac{r_o - r_i}{2}\right)\right] \tag{3.22}$$

Also, Outer radius,

$$r_o = (\tfrac{D}{2})/1000$$

Inner radius,

$$r_i = r_o - \left(\frac{thickness}{1000}\right)$$

Wall thickness,

$$r_w = r_o - r_i \tag{3.23}$$

Where,
Boundary between inner and outer zone,

$$r_b = r_i + (\theta * r_w) \tag{3.24}$$

Mass of tube metal for inner zone,

$$M_{m,i} = \rho_{steel} * \pi * \left(r_b^2 - r_i^2\right) * (heated\ tube\ length) \tag{3.25}$$

Mass of tube metal for outer zone,

$$M_{m,o} = \rho_{steel} * \pi * \left(r_o^2 - r_b^2\right) * (heated\ tube\ length) \tag{3.26}$$

Specific of metal, $C_m = 4200\ {}^{KCal}/_{KgC}$
$C_{m,i} = C_m * M_{m,i}$
$C_{m,o} = C_m * M_{m,o}$

Using iterative method,$T_{m,i}$and $T_{m,o}$ are calculated. Hence, Rate of change of temperature for a given time steps,

$$\frac{dT_{m,i}}{dt} = \frac{C_{m,i} * \left({dT_{m,i}}/{dt}\right)}{C_m * M_{m,o}} \tag{3.27}$$

$$\frac{dT_{m,o}}{dt} = \frac{C_{m,o} * \left({dT_{m,o}}/{dt}\right)}{C_m * M_{m,o}} \tag{3.28}$$

Average mid wall temperature,

$$T_{wall,avg.} = \frac{T_{m,i(iterated)} + T_{m,o(iterated)}}{2} \tag{3.29}$$

Inner wall temperature,

$$T_{wall,i} = T_{m,i} + (r_i - r_{m,i}) * \frac{(T_{m,o} - T_{m,i})}{(r_{m,o} - r_{m,i})} \tag{3.30}$$

Outer wall temperature,

$$T_{wall,o} = T_{m,i} + (r_o - r_{m,i}) * \frac{(T_{m,o} - T_{m,i})}{(r_{m,o} - r_{m,i})} \tag{3.31}$$

3.7 Log-mean temperature difference

From the reference [28], LMTD equation is given as,

$$LMTD = \triangle T_m = \frac{\triangle T_1 - \triangle T_2}{ln\frac{\triangle T_1}{\triangle T_2}} \tag{3.32}$$

Where, $\triangle T_1$and $\triangle T_2$ are the temperature differences between two fluids at each end of counter flow and parallel flow heat exchangers.
For counter flow,

$$\triangle T_{1=}T_{h,i} - T_{c,o}$$

$$\triangle T_2 = T_{h,o} - T_{c,i}$$

For Parallel flow,

$$\triangle T_{1=}T_{h,i} - T_{c,i}$$

$$\triangle T_2 = T_{h,o} - T_{c,o}$$

Figure 3.5 and 3.6 gives temperature distribution in counter flow and parallel flow heat exchangers.

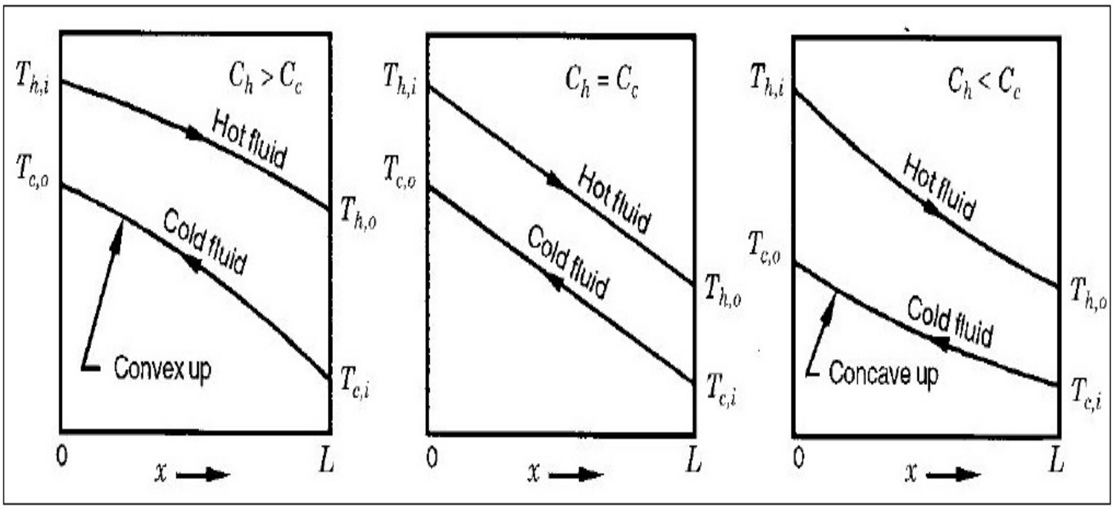


Figure 3.5: Temperature distribution in counter flow heat exchanger [29].

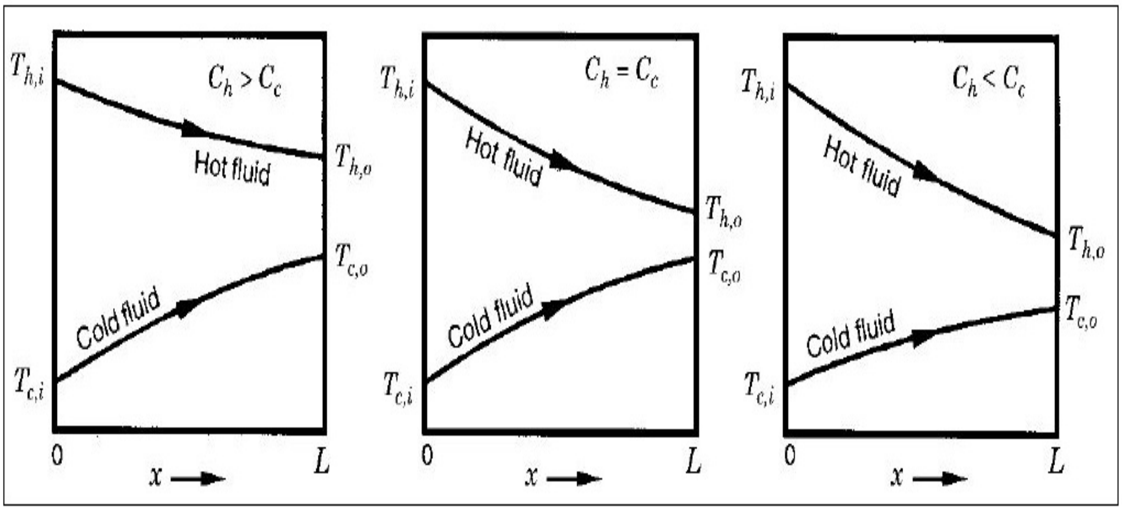


Figure 3.6: Temperature distribution in parallel flow heat exchanger [29].

Chapter 4

Simulation of Boiler Componets

4.1 General

The steady state and dynamic equation (Refer chapter 3) formed based on conductive, convective and non-luminous heat transfer processes which are formulated in the Excel based programming tool called Visual Basic Application (VBA) are solved using Advanced Continuous Simulation Language (ACSL-X). All the minor components have less heating surface area, that's why they have low effect on wall and metal temperature hence here desciption of only major componets are described in detail. But in actual, modeling of all the minor componets are also done in the present study. Simulation of all the sub-models of boiler components is described below:

4.2 Superheater

In the superheater section, there are several components like SH-Roof, SH-Cavity, SH-Screen, SH-Hanger, LTSH, LTSH-Terminal, Panel-SH and Platen-SH (For typical arrangement of superheater, refer figure 4.2). From all of these components, large temperature rise takes place in the LTSH, Panel SH and Platen SH as there heating surface area is large around $4750\,m^2$ for LTSH and $1695\,m^2$ for platen SH while other components have less heating surface area so temperature rise in steam is less. The approximate outside tube diameter and thickness are $0.04\,m$ and $0.006\,m$ for LTSH whereas for platen SH $0.05\,m$ and $0.007\,m$ respectively, similarly longitudinal and transverse pitches are $0.09\,m$ and $0.15\,m$ for LTSH while for platen SH $0.06\,m$ and $0.76\,m$ respectively. The number of assymblies are around 95 in LTSH and 25 in platen SH. Detailed simulated work of LTSH sub-model was presented below.

LTSH is a low temperature superheater or we can say that it is a primary superheater. The saturated steam from drum is first comes to this component where steam become superheated according to its heating surface area so that steam temperature increases in small range and small pressure drop will take place. Due to some amount of gas heat coming from the furnace is used by the steam to become superheated, the drop in temperature and pressure for gas side. The method used for calculate convective and non luminous heat transfer coefficient, heat transfer rate and to calculate wall/metal temperatures is described in chapter 3. Figure 4.1 gives detailed analysis for LTSH, where gas path is shown by pink color and steam path is shown by brown color.

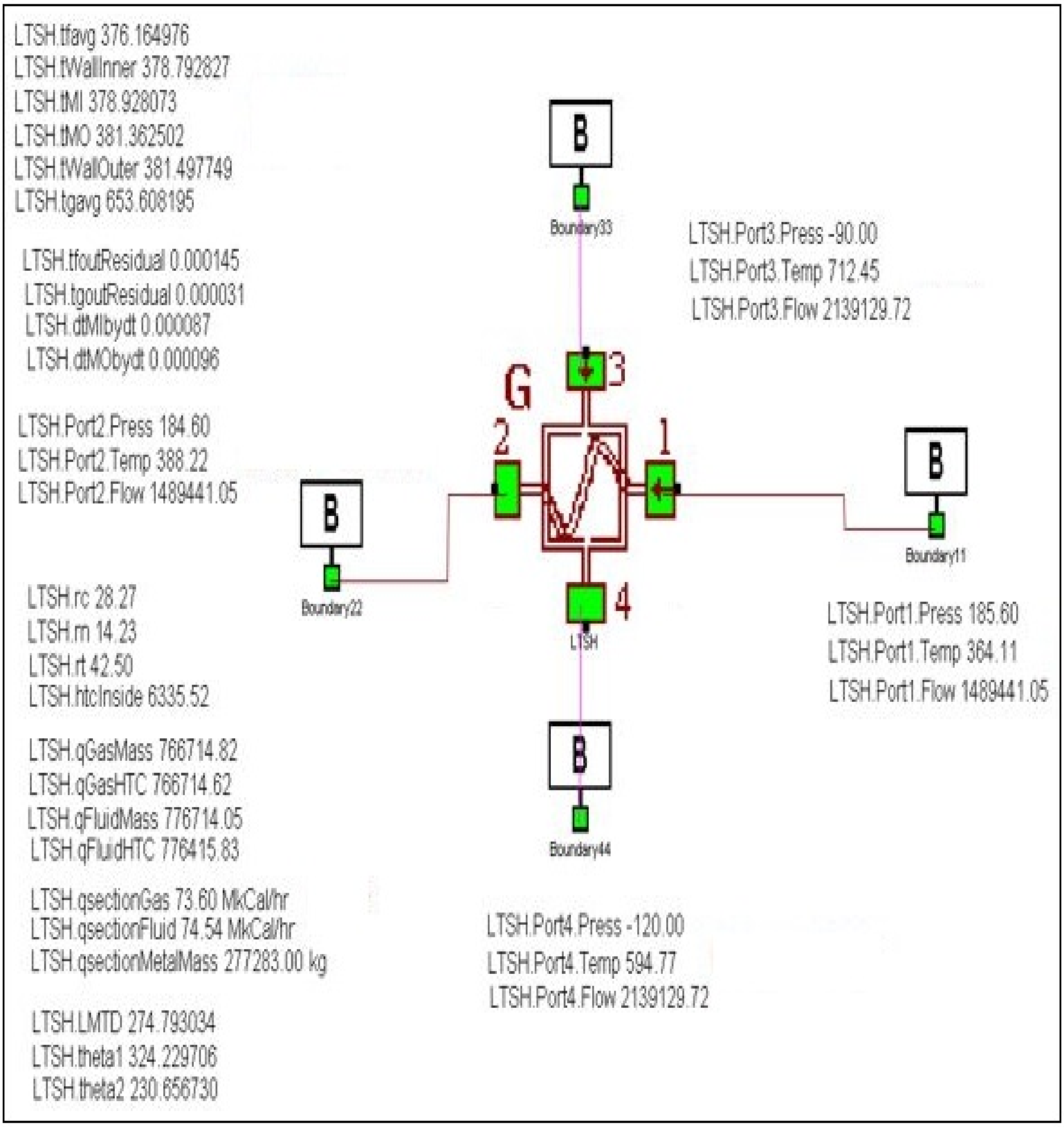


Figure 4.1: Simulation of LTSH

Similarly, sub-models for all superheater sections were build and simulated, then linked all together. The core model of superheater is shown in given figure 4.2.

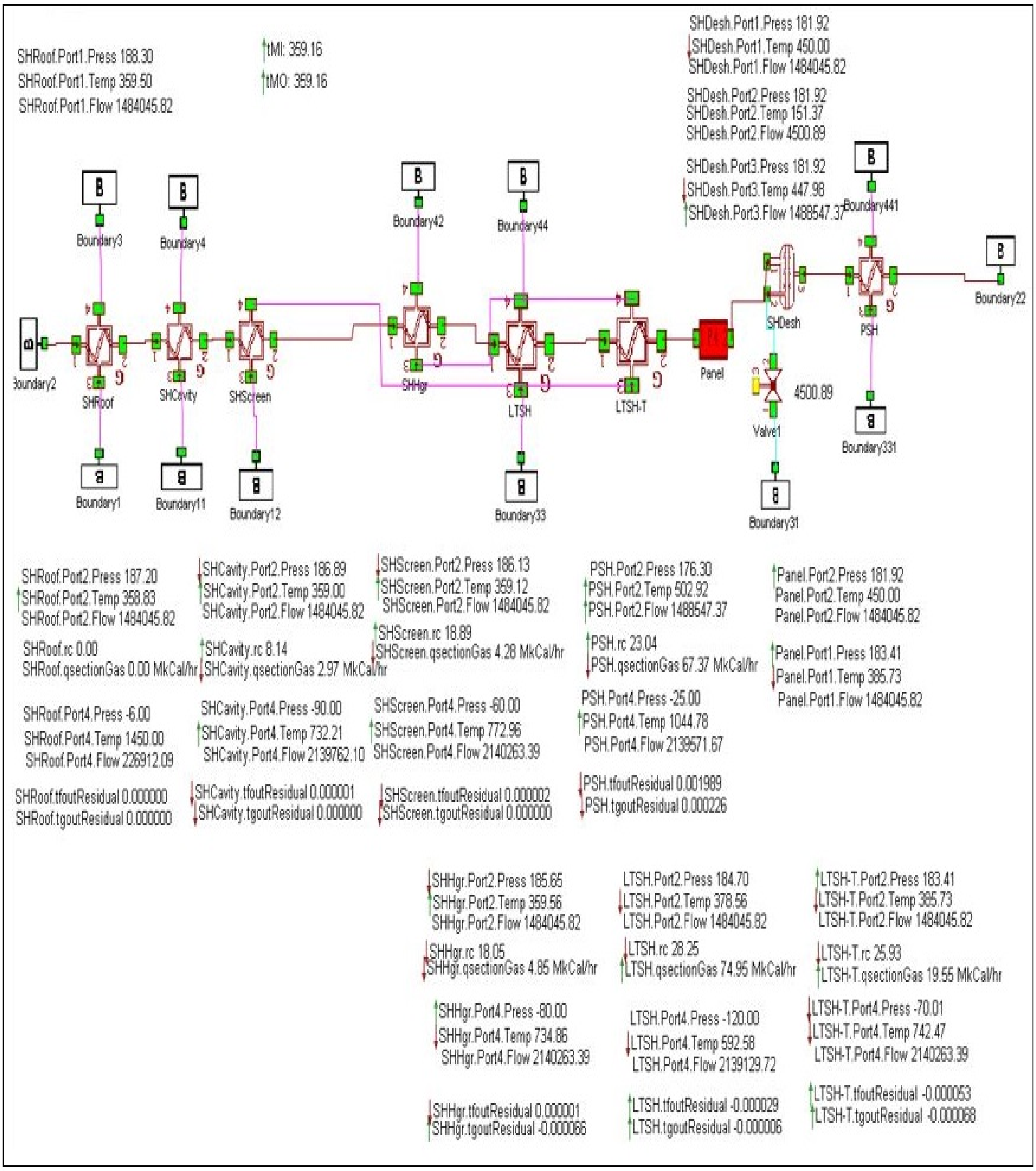


Figure 4.2: Simulation of superheater section

4.3 Reheater

There is large temperature rise in this section as there high heating surface area around $3460\,m^2$ for RH Front and $3853\,m^2$ for RH Rear. The approximate outside tube diameter and thickness for both RH Front and RH Rear are $0.05\,m$ and $0.004\,m$ respectively, similarly longitudinal pitch is $0.06\,m$ for RH Front and $0.09\,m$ for RH Rear. The transverse pitch is around $0.25\,m$ for both componets. The approximate number of assymblies are also same in both components i.e. 75. This reheater section consists of RH-Front, WW-Hanger and RH-Rear (For typical arrangement refer figure 4.4.).

From the high pressure turbine the superheated steam comes into the reheater section where temperature of steam increases in large amount. In the case of reheater front, there is large amount of radiation effect is takes place due to direct luminosity of flame gas, emmisivity or combustion of gas where carbondioxide and water present in gas. Hence, radiation is given externally. The method used for calculate convective and non luminous heat transfer coefficient, heat transfer rate and to calculate wall/metal temperatures is described in chapter 3. Gas path is shown by pink color and steam path is shown by brown color. Detailed simulated work of RH Front sub-model was presented in figure 4.3.

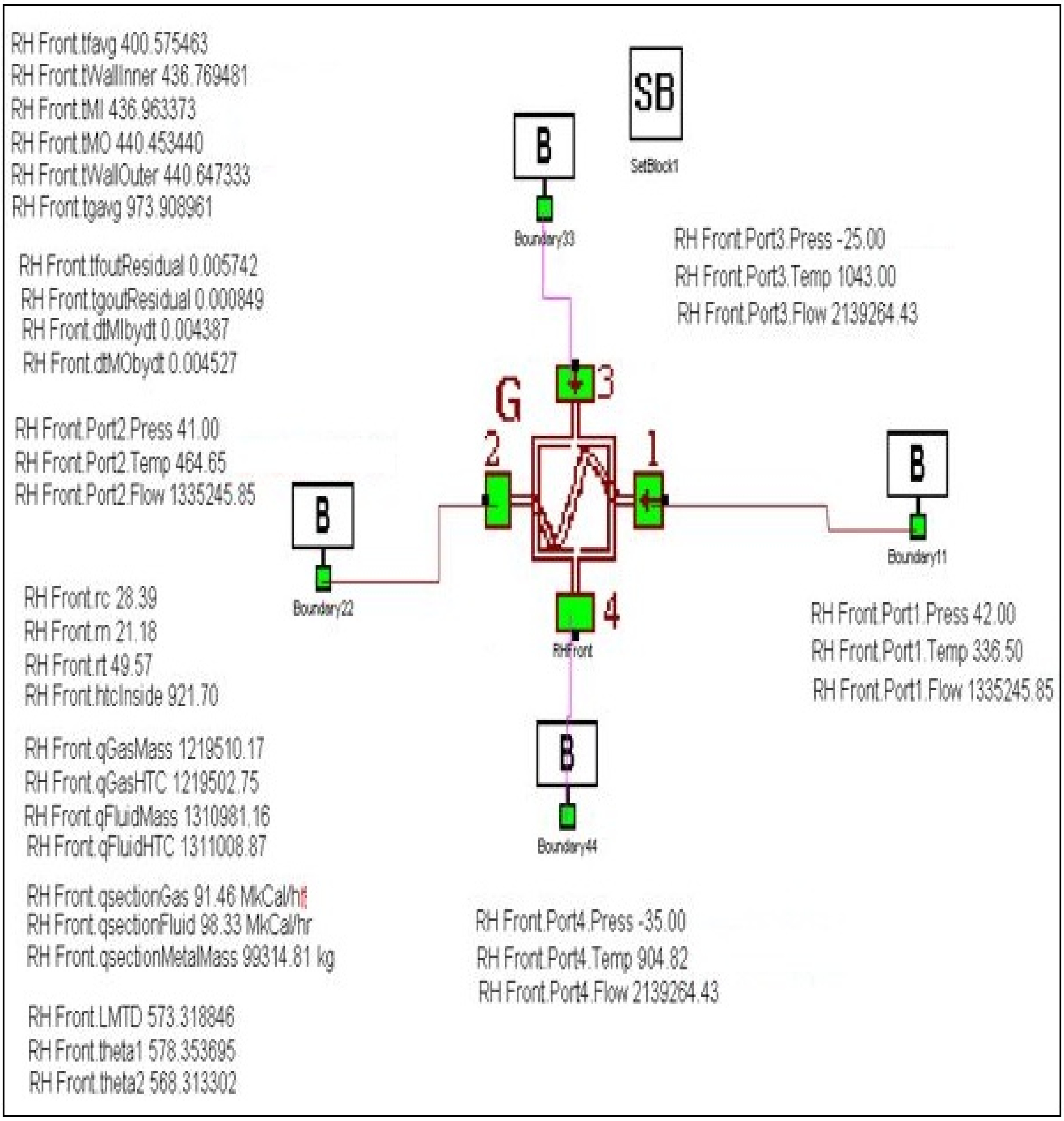


Figure 4.3: Simulation of reheater front

Similarly, sub-models for reheater rear and waterwall hanger sections were also build and simulated, then linked together. The core model of reheater is shown in given figure 4.4.

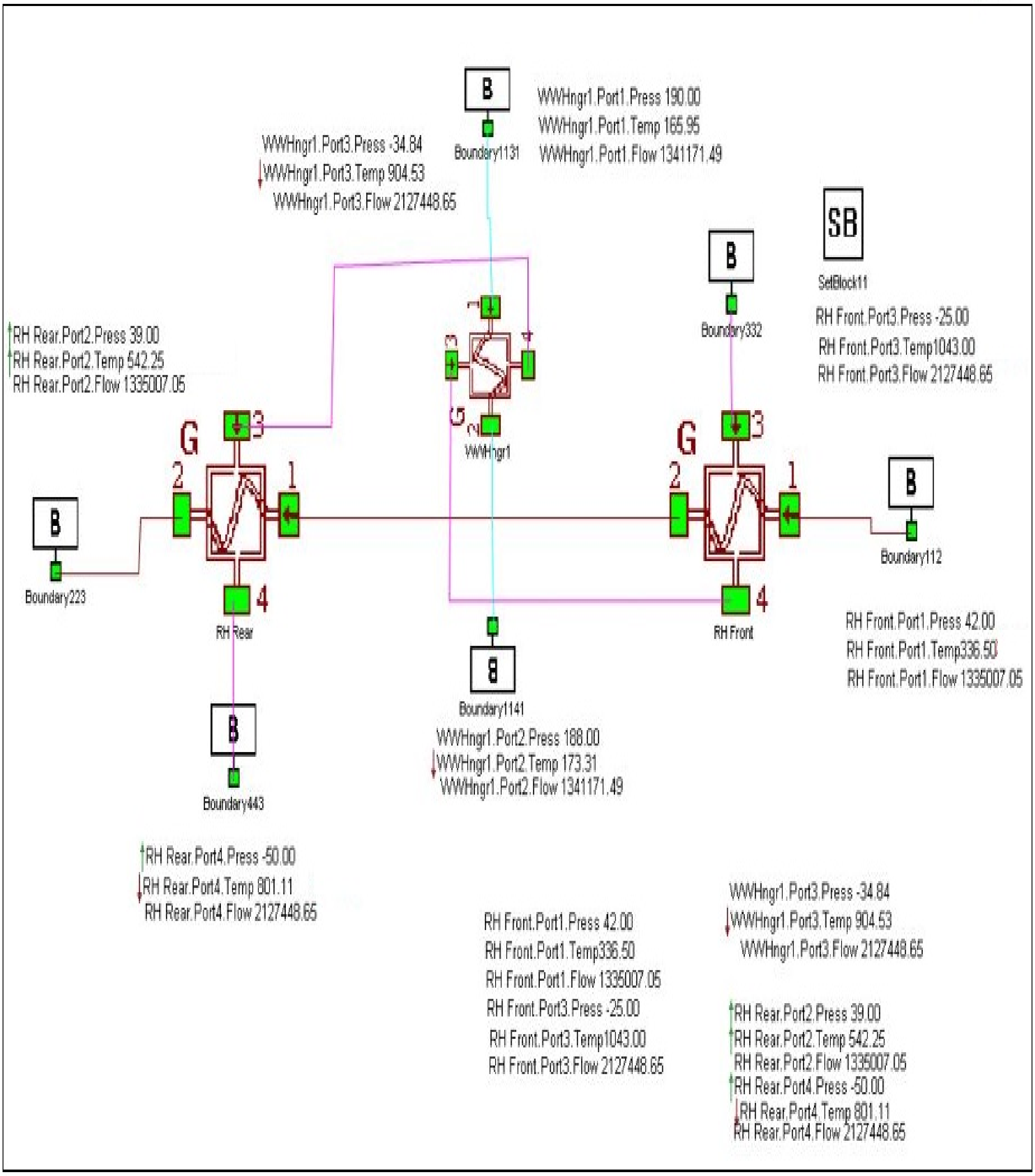


Figure 4.4: Simulation of reheater section

4.4 Economizer

From the condenser, the saturated water is comes through the various low pressure heaters, deareater and pump into the economizer. Economizer is used to heat the water so that required amount of heated water is gone to the drum hence it required high heating surface area around 23650 m^2 hence, gas gives heat to the water in large amount, that's why large temperature and pressure drop take place in gas side. The approximate outside tube diameter and thickness are 0.04 m and 0.005 m, similarly longitudinal and transverse pitches are 0.07 m and 0.10 m. The number of assymblies are around 95 in economizer. The heat transfer is only due to convection, there is no radiation effect in the case of economizer.The gas path is shown by pink color and water path is shown by blue color. Detailed simulated work of economizer sub-model was presented in Figure 4.5.

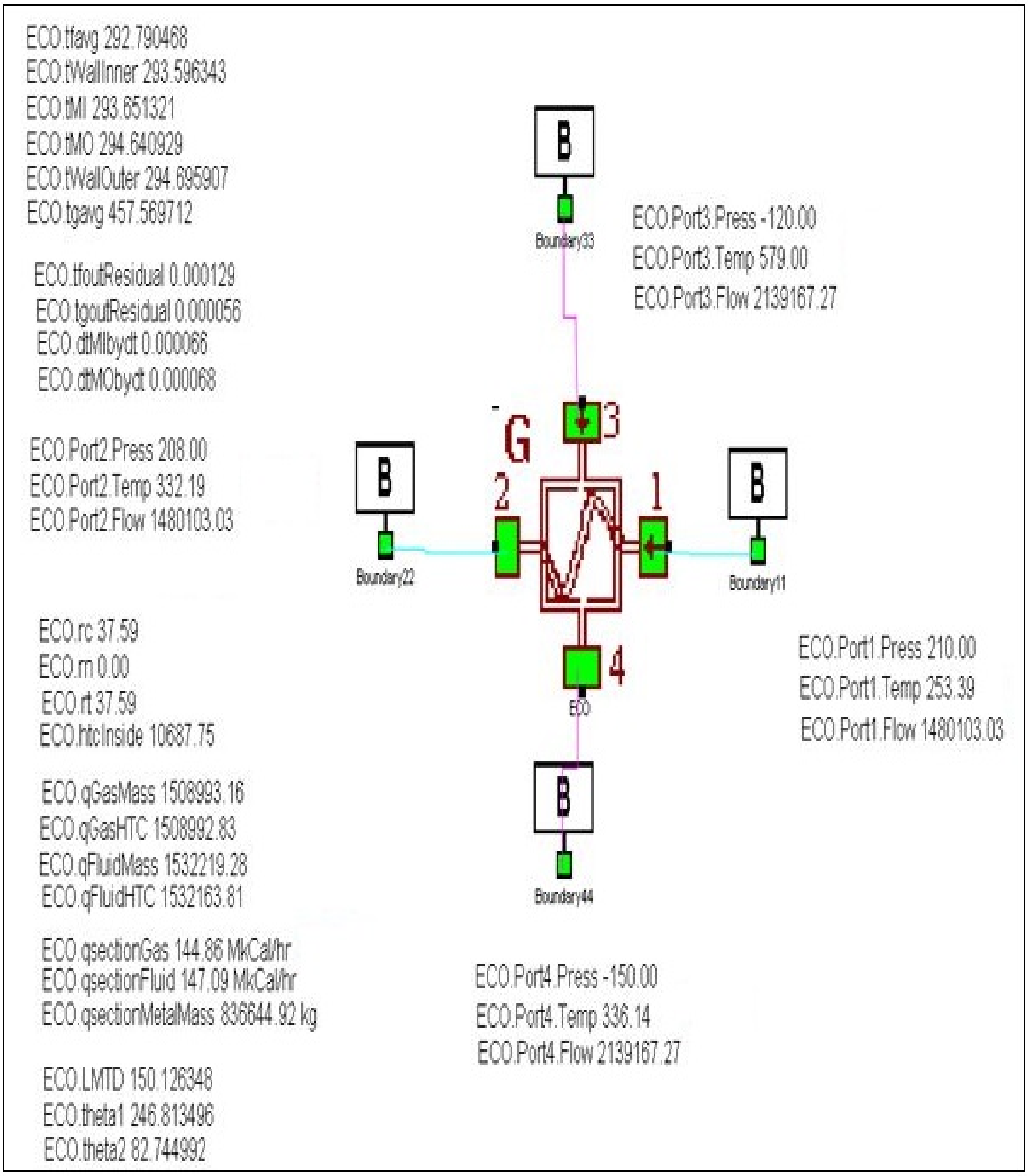


Figure 4.5: Simulation of economizer

4.5 Boiler drum

Drum having outer diameter $2.2\,m$, thickness $0.2\,m$, length $0.02\,m$ is situated at upper front of boiler where steam and water separation will take place, so steam goes to superheated in the superheater and saturated water comes to downcomer having length $70\,m$. To minimize heat loss, insulation is provided on the surface of drum having thickness $0.1\,m$. The water walls surrounded in furnace having outer diameter, thickness, length and tube pitching are $0.05\,m$, $0.005\,m$, $80m$ and $0.06\,m$ respectively. The boiler used for 500MW is forced circulation boiler so pump is used having volume $4\,m^2$. It is a centrifugal type pump whose rate flow 16554 and rated pressure rise were $20\,Kg/cm^2$

4.6 Simulation of combined model of boiler

After testing all the heating sections of boiler individually and their results are found to be in agreement with the expected process trends. Now, all the sections are linked together.

In the combined model, minor componets like SH-Roof, SH-Cavity, SH-Screen, SH-Hanger, LTSH-Terminal, SH-Panel, WW-Hanger, WW-Screen are also included. Economizer is connected to the boiler drum through referencer1 in the given below figures. The typical arrangement of overall analysis of all the sections of boiler are shown in figure 4.6 and 4.7.

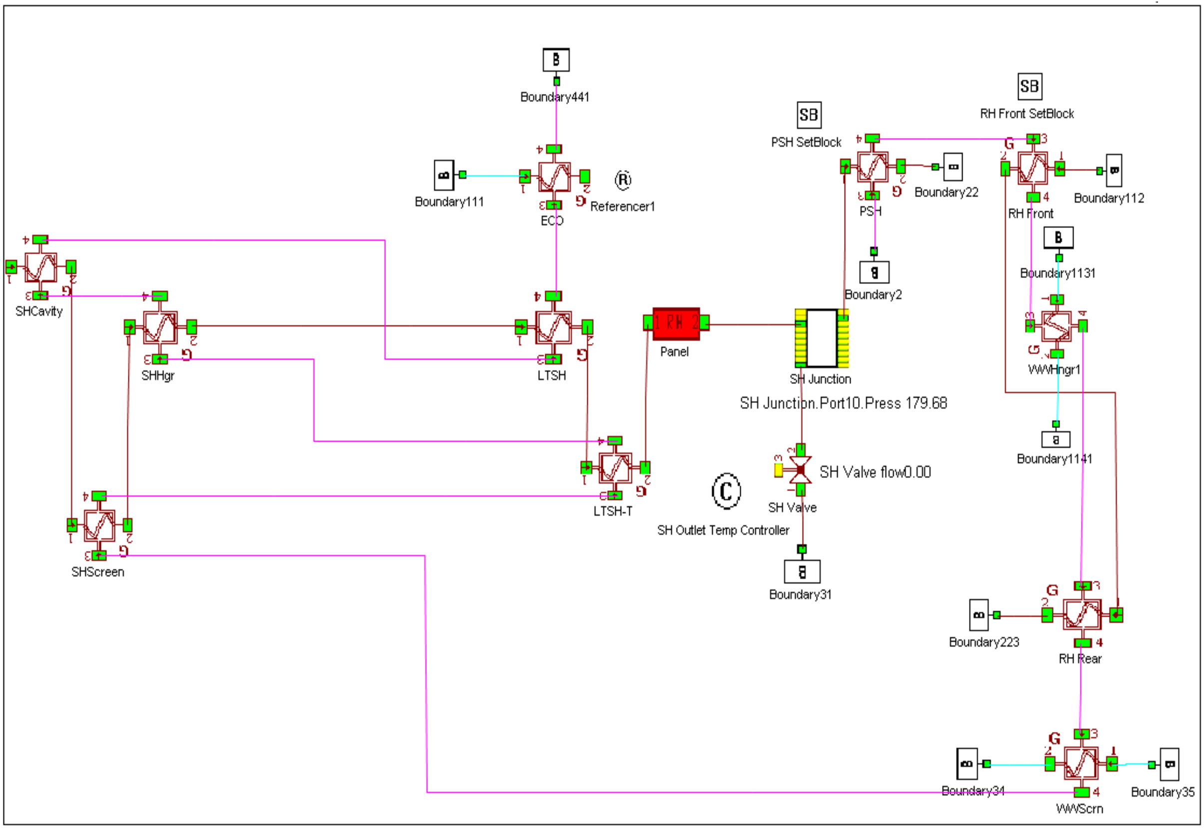


Figure 4.6: Simulation of all heating sections of boiler

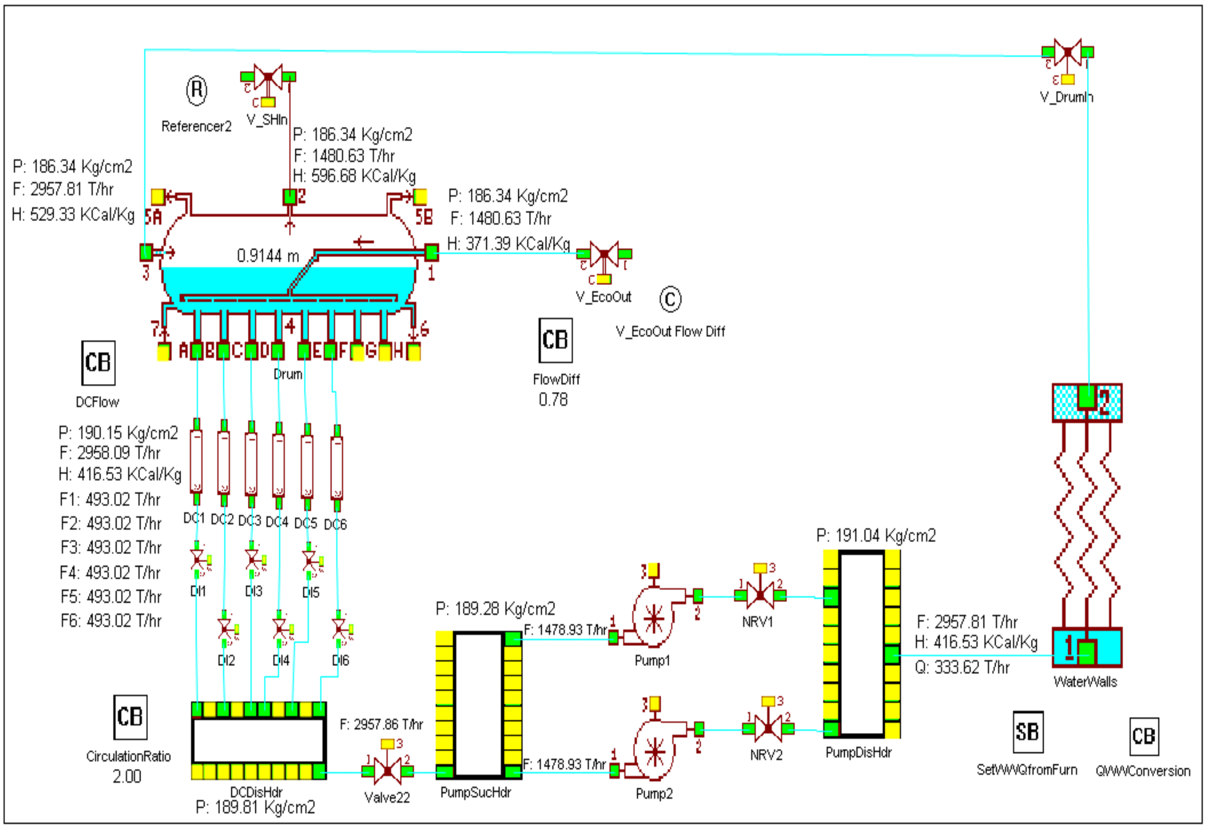


Figure 4.7: Simulation of boiler drum

From the condensor, deareater and low pressure heaters, the saturated water comes to the economizer at *port* 1. From the *port* 2 of economizer water goes to *port* 1 of the drum through valve (V_EcoOut). Whereas water present in the water wall having flow $2957.8 TPH$. is heated due to flue gas present in the furnace, hence this water in the form of vapour comes to the drum from the *port* 3. The amount of heat absorbed by the waterwall from the furnace is $333.62 TPH$. Now, the seperation of water and steam takes place so that form the *port* 2 of drum the steam goes to superheater section through valve (V_SHIn) and remaining water from the port ($4A, 4B, 4C, 4D, 4E, 4F$) comes to the downcomer ($DC1, DC2, DC3, DC4, DC5, DC6$). Through the valves and multi junction (downcomer discharge header and pump suction header), water comes into the pump. And through the pump discharge header the water goes to the waterwalls. We should maintained the circulation ratio of formation of steam through water is around 2.

4.7 Results and discussion

In this section, the detailed study of results obtained from simulated model are compared with testing data. And also study, the behaviour of heat transfer coefficient, heat transfer rate and wall/metal temperatures is carried out under running condition of model and also by vary the gas flow and steam temperature at outlet of superheater, reheater and economizer.

4.7.1 Simulated results and their deviation with respect to testing data

Based on above case study, the following results are obtained in terms of convective and non-luminous radiation heat transfer coefficient, total heat transfer coefficient for fluid and gas side, wall temperature at inlet and outlet and metal temperature at inlet and outlet. Table 4.1 shows the gas and steam flow in superheater, reheater and economizer.

Table 4.1: Steam flow and gas flow in boiler

Components	Superheaters	Reheater	Economizer
Steam / water flow($Kg/hr.$)	1480625	1335000	1480626
Gas flow($Kg/hr.$)	2139385	2139385	2139385

Pressure(*in* Kg/cm^2) and temperature (*in* $°C$) at inlet and outlet of gas side for major components of boiler and also deviation of outlet gas temperature with testing data are shown in table 4.2.

Table 4.2: Pressure and temperature for gas side

Components	$(P_{g,in})$	$(P_{g,out})$	$(T_{g,in})$	$(T_{g,out})_{obt.}$	$D\,(T_{g,out})$
PSH	-15	-25.69	1145	1048.32	0.5 %
RH Front	-25.69	-34.64	1048.3	899.34	0.3 %
RH Rear	-42.59	-50.51	887.8	801.54	0.9 %
LTSH	-85.70	-116.39	698	582.64	0.6 %
ECO	-116.39	-150	582.64	341.88	0.3 %

Table 4.3 shows the deviation of simulated data with testing data for pressure ($in\ Kg/cm^2$) and temperature ($in\ °C$) at inlet and outlet of steam side for major components of boiler.

Table 4.3: Pressure and temperature for steam side

Components	$(P_{s,in})_{obt.}$	$D\,(P_{s,in})$	$(P_{s,out})_{obt.}$	$D\,(P_{s,out})$	$(T_{s,in})_{obt.}$	$D\,(T_{s,in})$	$(T_{s,out})_{obt.}$	$D\,(T_{s,out})$
LTSH	183.46	1.1 %	182.77	1.15 %	361.54	0.7 %	386.97	0.7 %
PSH	179.68	0.06 %	176.3	0 %	460.89	0.9 %	539.14	0.1 %
RH Front	43	0 %	42.26	0.37 %	337.48	0.2 %	461.32	0.5 %
RH Rear	42.26	0.37 %	41.30	0 %	461.32	0.5 %	540.97	0.1 %
ECO	193.20	0 %	188.29	0.005 %	253.4	0 %	330.87	0.7 %

Table 4.4 shows the deviation of simulated data with testing data for convective and non-luminous heat transfer coefficient ($in\ Kcal/hr.m^2.k$) and also total heat transfer rate ($in\ MKcal/hr.$) for gas and fluid (steam/water) side for major components of boiler.

Table 4.4: Heat transfer coefficients and heat transfer rate

Components	$(h_c)_{obt.}$	$D\,(h_c)$	$(h_n)_{obt.}$	$D\,(h_n)$	h_i	$(Q_g)_{obt.}$	$D\,(Q_g)$	$(Q_f)_{obt.}$	$D\,(Q_f)$
PSH	23.05	0.2 %	39.2	0.3 %	3547.61	66.4	3 %	87.56	4 %
RH Front	25.93	1 %	21.32	0.8 %	924.51	92.77	4 %	95.83	2 %
RH Rear	21.31	0.8 %	21.71	0.8 %	949.71	59.55	0.9 %	59.58	6 %
LTSH	28.47	0.2 %	12.65	0.4 %	6285.08	73.54	1.8 %	73.54	7 %
ECO	37.82	0.02 %	0	0 %	10752.7	150.47	6 %	148.47	5 %

Note: The testing data for inside heat transfer coefficient ($in\ W/m^2.k$) is not available. Hence, deviation for inside heat transfer coefficient is not carried out in the above table.

Table 4.5 shows simulated results of wall and metal temperature ($in\ °C$) at inside and outside of tube for major components of boiler.

Table 4.5: Simulated wall and metal temperature

Components	$T_{w,i}$	$T_{m,i}$	$T_{m,o}$	$T_{w,o}$
PSH	520.18	521.04	536.39	537.24
RH Front	434.63	434.82	438.21	438.40
RH Rear	520.81	520.93	522.93	523.04
LTSH	377.08	377.23	379.82	379.97
ECO	296.02	296.08	297.16	297.22

Table 4.6 shows pressure ($in\ Kg/cm^2$), temperature ($in\ ^\circ C$) and flow ($Kg/hr.$) of water inside the boiler drum through economizer, waterwall and downcomer. And also for steam going to the superheater from $port\,2$.

Table 4.6: Simulated parameters of drum

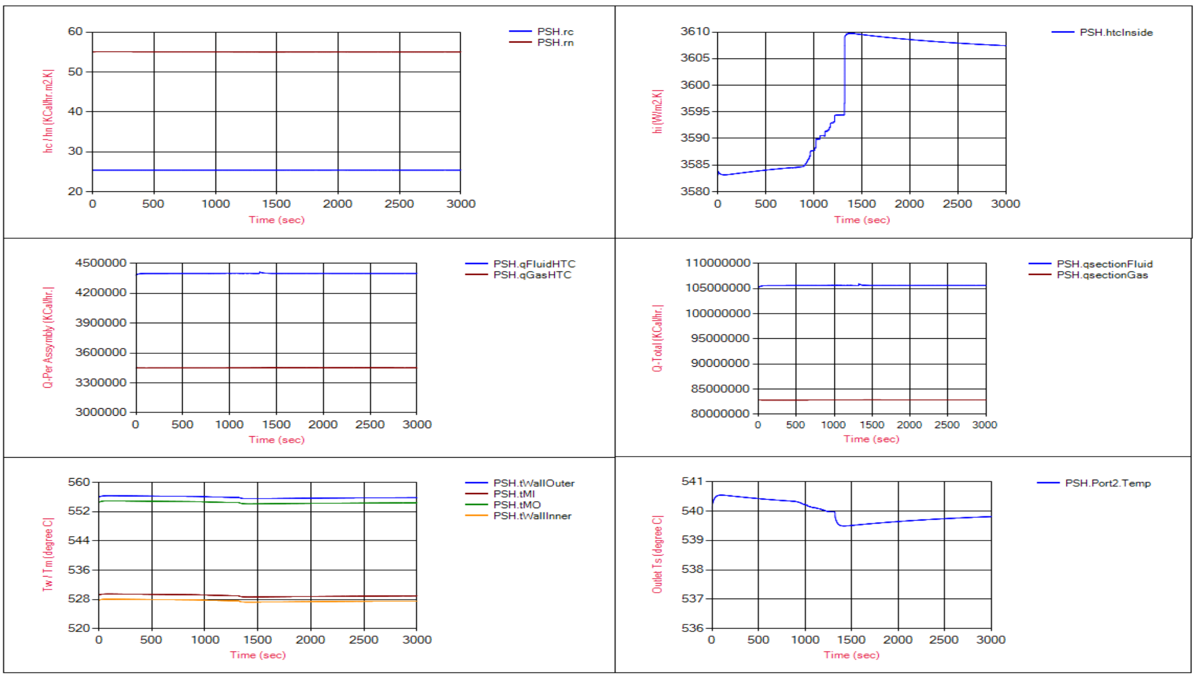
Port	1	Port	3
$[(P_w)_{in}]_{ECO}$	186.34	$[(P_w)_{in}]_{WW}$	186.34
$[(T_w)_{in}]_{ECO}$	330.79	$[(T_w)_{in}]_{WW}$	358.23
$Flow$	1480626	$Flow$	2957824

Port	2	Port	4
$(P_s)_o$	186.34	$[(P_w)_o]_{DC}$	190.14
$(T_s)_o$	358.23	$[(T_w)_o]_{DC}$	358.71
$Flow$	1480625	$Flow$	493021

4.7.2 Behaviour of parameters at superheater outlet

In the combined model, behaviour of convective, non-luminous and inside heat transfer coefficient, heat transfer rate, wall and metal temperature, outlet steam temperature, SH valve flow and its position and also steam flow for superheater are shown in these graphs.

The steam temperature at outlet of superheater should be $540°C$, hence when the outlet temperature is coming more then $540°C$, the valve supply more water to coming down temperature and become $540°C$ by oppening its position for supplying water. It means valve is used as controller to control the steam temperature at outlet by spraying water. Hence, Figure 4.8 shows behaviour of all the above parameters at superheater outlet in combined model at real time



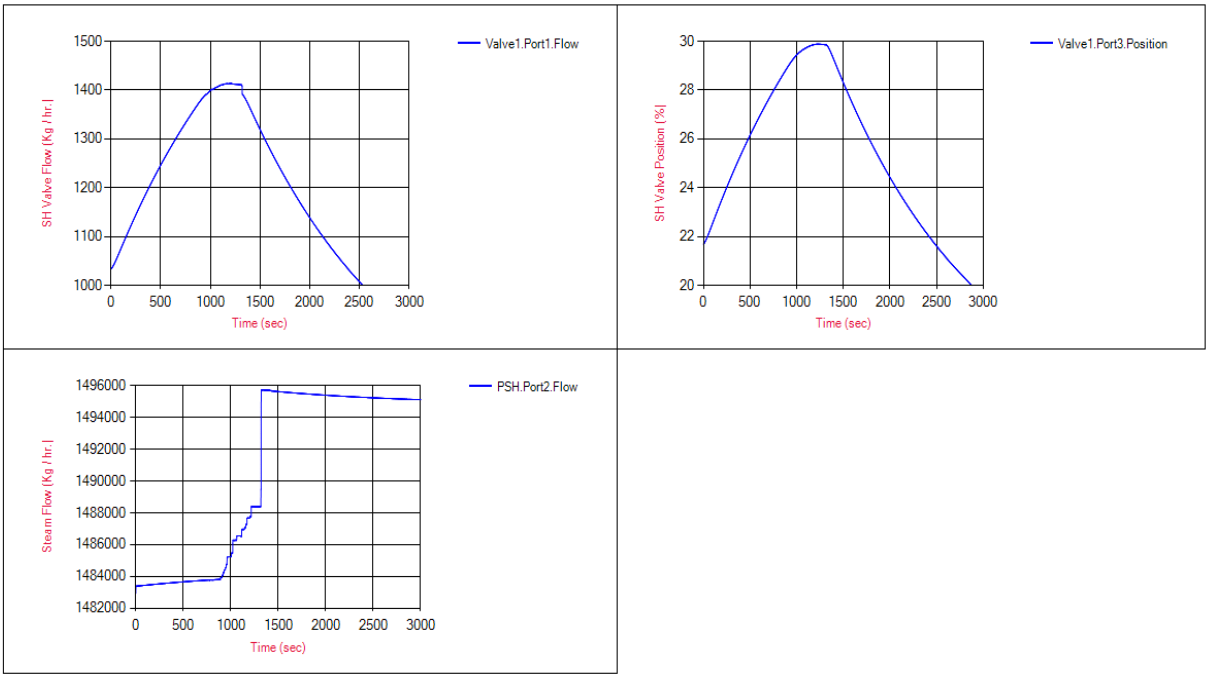


Figure 4.8: Behaviour of various parameters at superheater outlet in combined model at real time

4.7.3 Behavior of parameters by changing gas flow and steam temperature

Now, in the case of sub-system or individual model of superheater, reheater and economizer by changing the gas flow ($Kg/hr.$) and steam temperature ($^\circ C$), the behaviour of all the parameters like outside ($in\ Kcal/hr.m^2.k$) and inside ($in\ W/m^2.k$) heat transfer coefficient, heat transfer rate per assymbly ($in\ Kcal/hr.$), total heat transfer rate ($in\ Kcal/hr.$), wall and metal temperature($^\circ C$) at outlet are shown in these graphs.

4.7.3.1 Behaviour of platen superheater by varying gas flow and steam temperature

Keep inlet gas temperature equal to $1450^\circ C$ and gradually vary gas flow from zero to $2139000 Kg/hr.$, behaviour of all the parameters are shown in these graphs. As far as steady state comes at zero flow, the graphs shows behaviour of all parameters due to previously enthalpy present in the component. In the case of outside heat transfer coefficient graph, the convective heat transfer coefficient 'r_c' is zero at zero flow which is gradually increases with increasing gas flow whereas the non-luminous heat transfer coefficient 'r_n' is around $40 Kcal/hr.m^2.k$ at zero flow due to direct radiation or direct luminosity to the component. Inside the tube, heat transfer coefficient decreases as gas flow increases. 'h_i' depends on thermal conductivity, Nusselt, Reynolds and Prandtl's number. The heat transfer through the gas to the metal and then metal to the steam is take place until both gas side and steam side heat transfer rate become equal or up to steady state. Due to radiation effect, the heat transfer rate through steam side is may be slightly more compare to gas side. In Platen superheater, there is high radiation effect so the difference between heat transfer rate through steam and gas sides is large. As gas flow increases, wall and metal temperature also increases. Figure 4.9 shows behaviour of all the parameters at superheater outlet by varying gas flow.

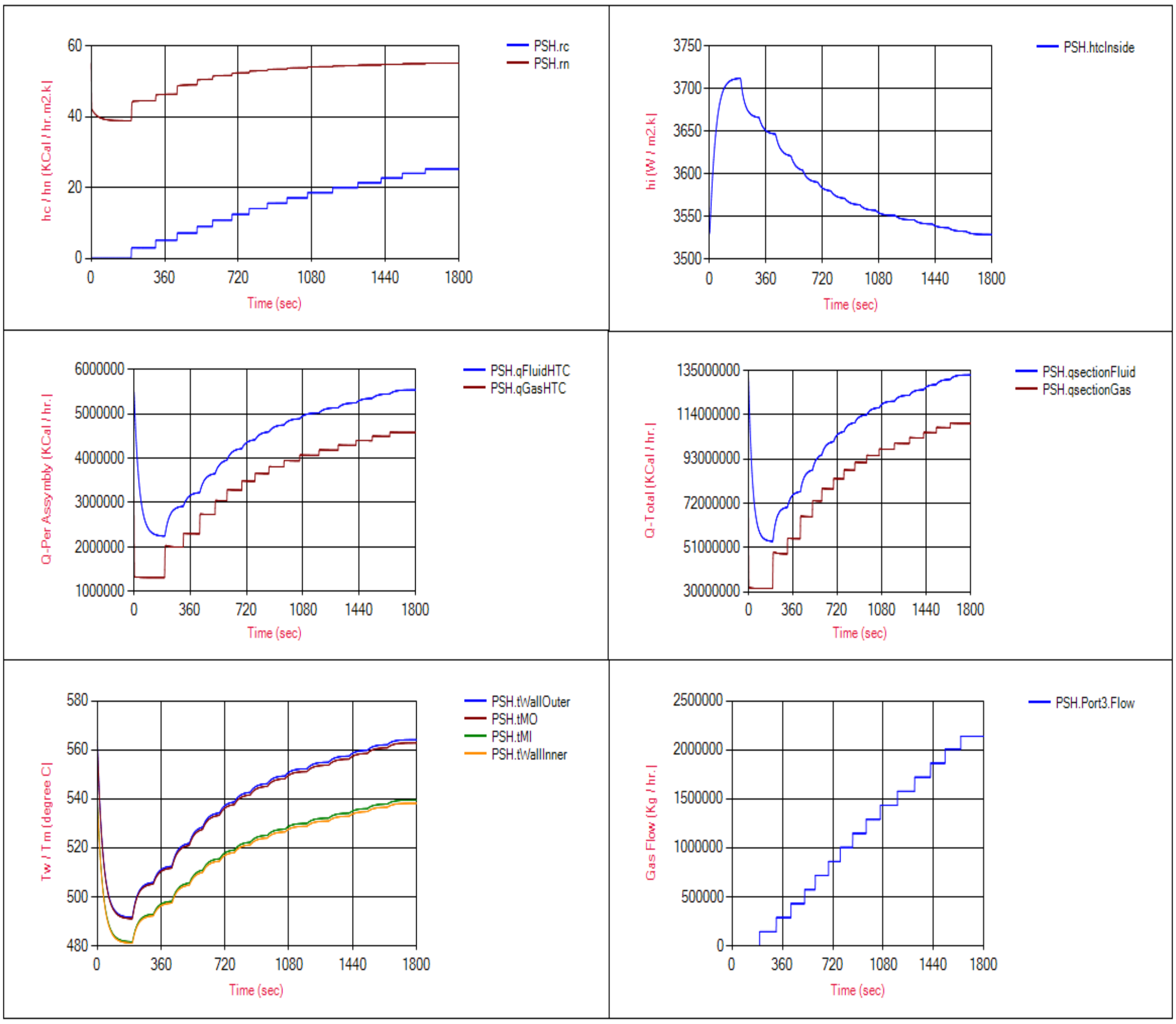


Figure 4.9: Behaviour of various parameters at superheater outlet by varying gas flow

Now, keep gas flow equal to $2139000 Kg/hr.$ and vary the inlet steam temperature from $420°C$ to $480°C$ gradually then behaviour of all the parameters are shown in these graphs. In the case of outside heat transfer coefficient, the effect of variation in steam temperature is very less so very less change in r_c and r_n . Whereas inside the tube, due to increase in steam temperature, steam pressure increases so its specific volume and density decrease also in the case of superheater, steam pressure is high around $185 Kg/cm^2$ hence inside heat transfer coefficient decreases. Initially heat transfer rate shows behaviour due to previous enthalpy present but after steady state, it vary in small propartion with respect to increase in steam temperature but difference between heat transfer rate due to steam and gas is much more compare to reheater and economizer due to high radiation effect. Wall and metal temperatures are increases with increase in steam temperature. Figure 4.10 shows behaviour of all the parameters at superheater outlet by varying steam temperature.

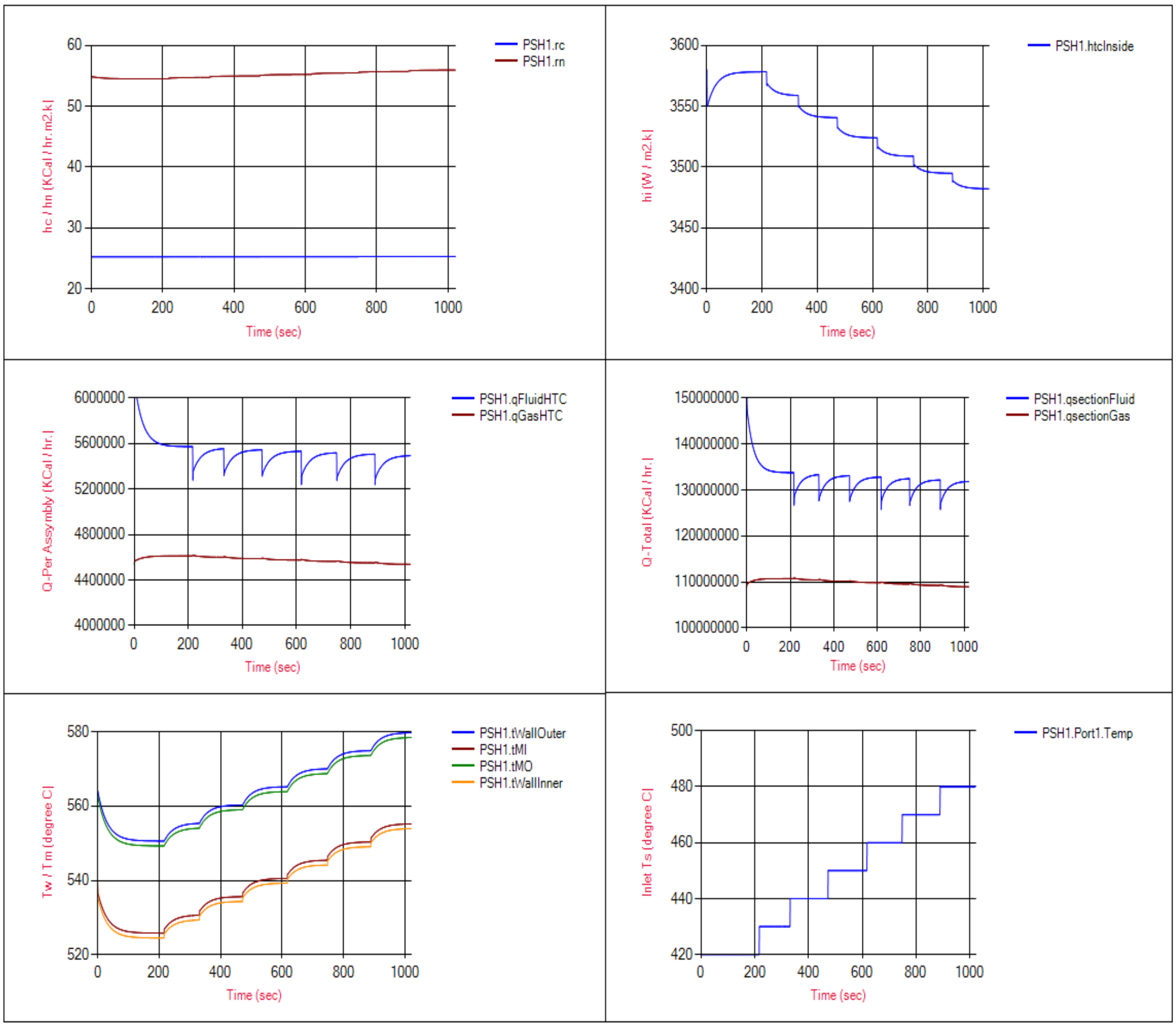


Figure 4.10: Behaviour of various parameters at superheater outlet by varying steam temperature

4.7.3.2 Behaviour of reheater rear by varying gas flow and steam temperature

Keep inlet gas temperature equal to $1068.64°C$ and gradually increases the gas flow from zero to $2139000 Kg/hr.$, the behaviour of all the parameters are shown in these graphs. Initially, as far as steady state comes at zero flow, all the graphs shows behaviour due to previously enthalpy present in the component. In the case of outside heat transfer coefficient graph, At zero flow, convective heat transfer coefficient 'r_c' is zero which is gradually increases with increase in gas flow whereas the non-luminous heat transfer coefficient 'r_n' is around $27 Kcal/hr.m^2.k$ due to radiation effect at zero flow. Inside the tube, heat transfer coefficient increases with increase in gas flow. The heat transfer rate through the gas to the metal and then metal to the steam and also wall and metal temperature are increases with increase in gas flow. Figure 4.11 shows behaviour of all the parameters at reheater outlet by varying gas flow.

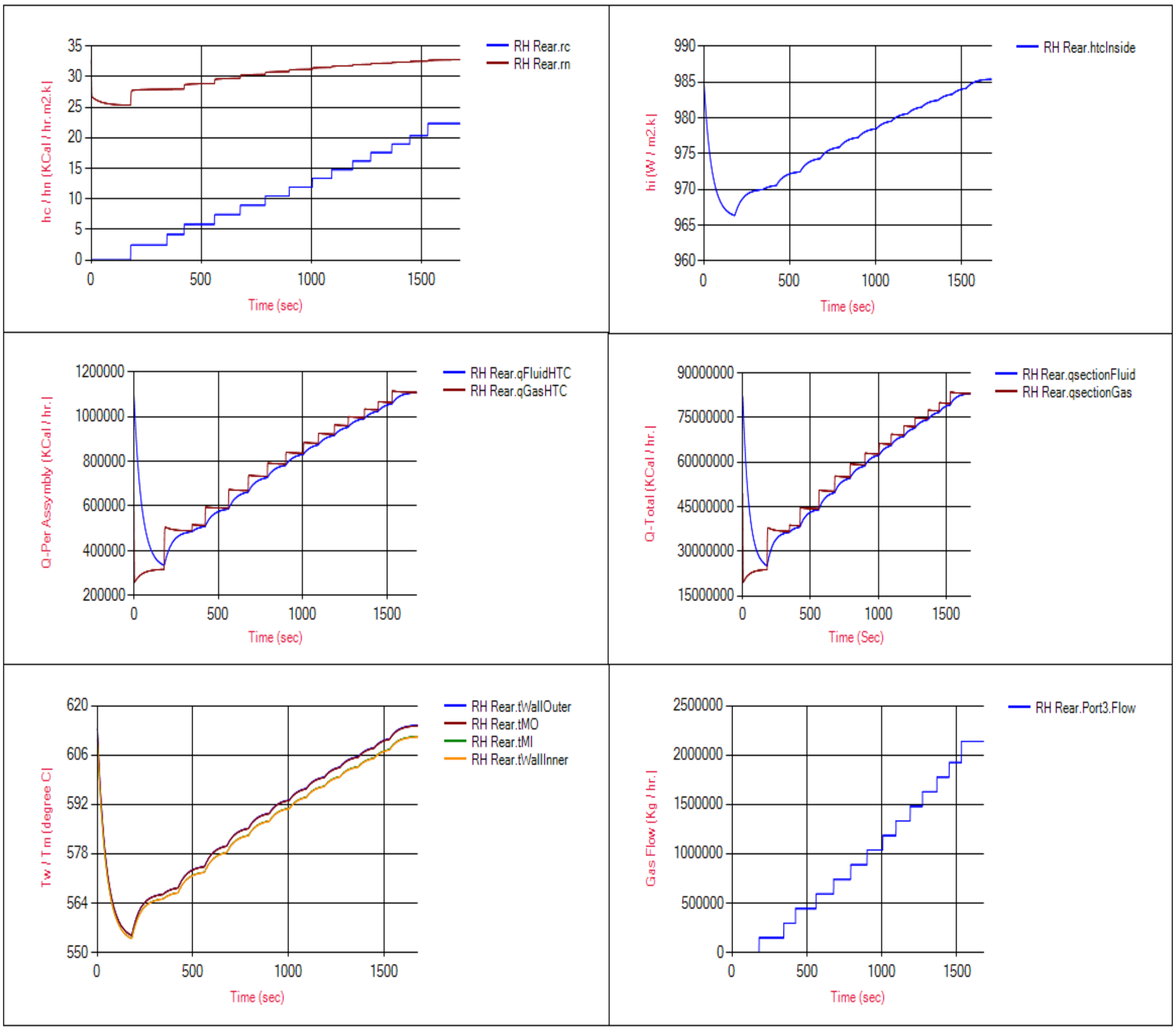


Figure 4.11: Behaviour of various parameters at reheater outlet by varying gas flow

Now, keep gas flow equal to $2139000Kg/hr$.and vary the inlet steam temperature from $500°C$ to $560°C$ then variation in outside heat transfer coefficient is very less, Whereas inside the tube, due to increase in steam temperature, steam pressure increases so its specific volume and density decrease and also in the case of reheater, steam pressure is very less compare to superheater around $40Kg/cm^2$,so that inside heat transfer coefficient increases. Initially heat transfer rate shows behaviour due to previous enthalpy present but after steady state, it vary in small propartion with respect to increase in steam temperature but difference between heat transfer rate due to steam and gas is very less compare to superheater due to low radiation effect. Wall and metal temperatures are increases with increase in steam temperature. Figure 4.12 shows behaviour of all the parameters at reheater outlet by varying steam temperature.

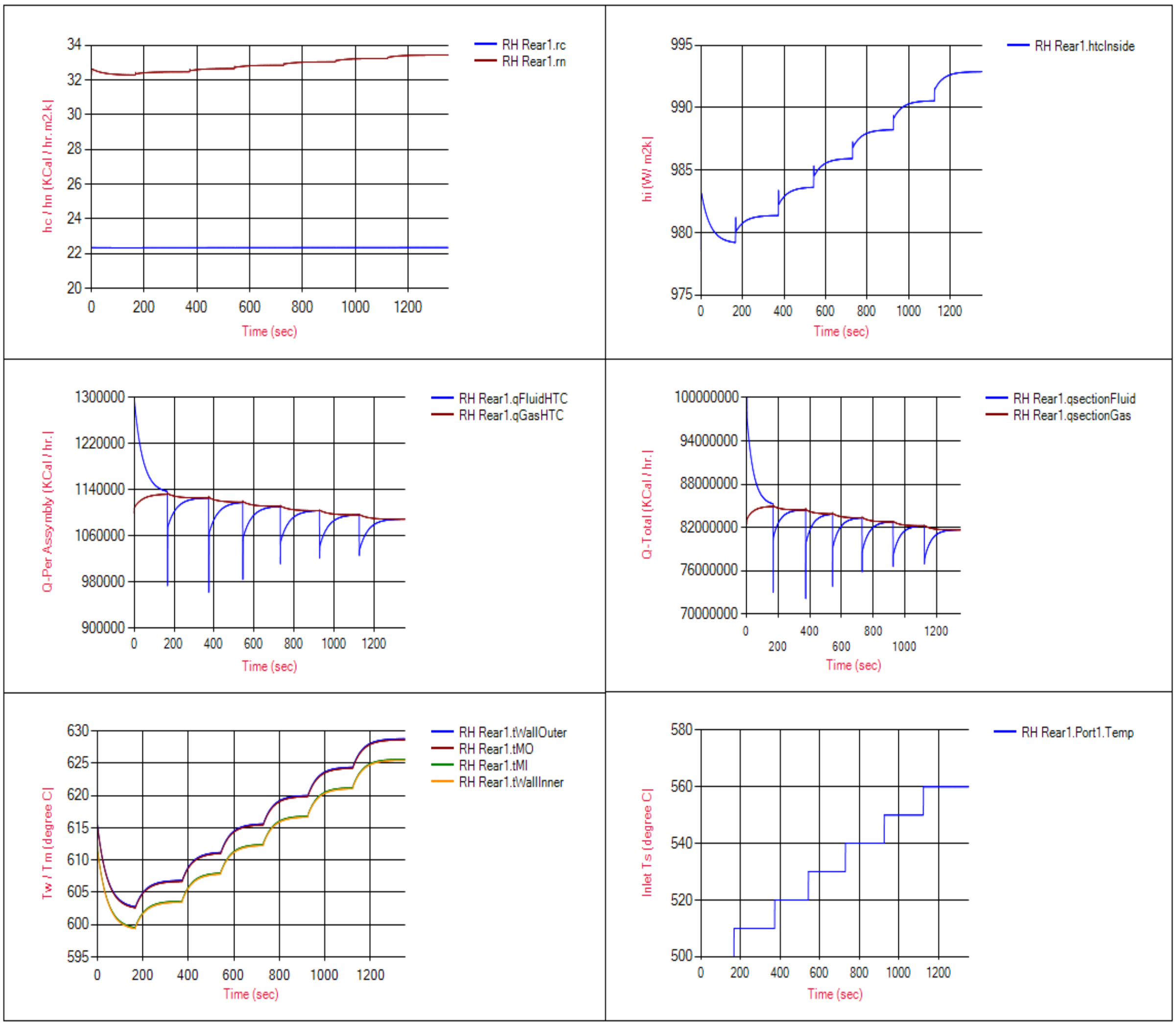


Figure 4.12: Behaviour of various parameters at reheater outlet by varying steam temperature

4.7.3.3 Behaviour of economizer by varying gas flow and steam temperature

Keep inlet gas temperature equal to 685°C for economizer and gradually increases the gas flow from zero to 2139000$Kg/hr.$, the behaviour of all the parameters are shown in these graphs. As far as steady state comes at zero flow, the graphs shows behaviour of all parameters due to previously enthalpy present in the component. In the case of outside heat transfer coefficient graph, the convective heat transfer coefficient 'r_c' is zero at zero flow which is gradually increases with increasing gas flow whereas the non-luminous heat transfer coefficient 'r_n' is zero due to there is no radiation effect takes place in economizer. Inside the tube, heat transfer coefficient is gradually increases with gas flow. The heat transfer rate wall temperatures also increase with increase in gas flow. Figure 4.13 shows behaviour of all the parameters at economizer outlet by varying gas flow.

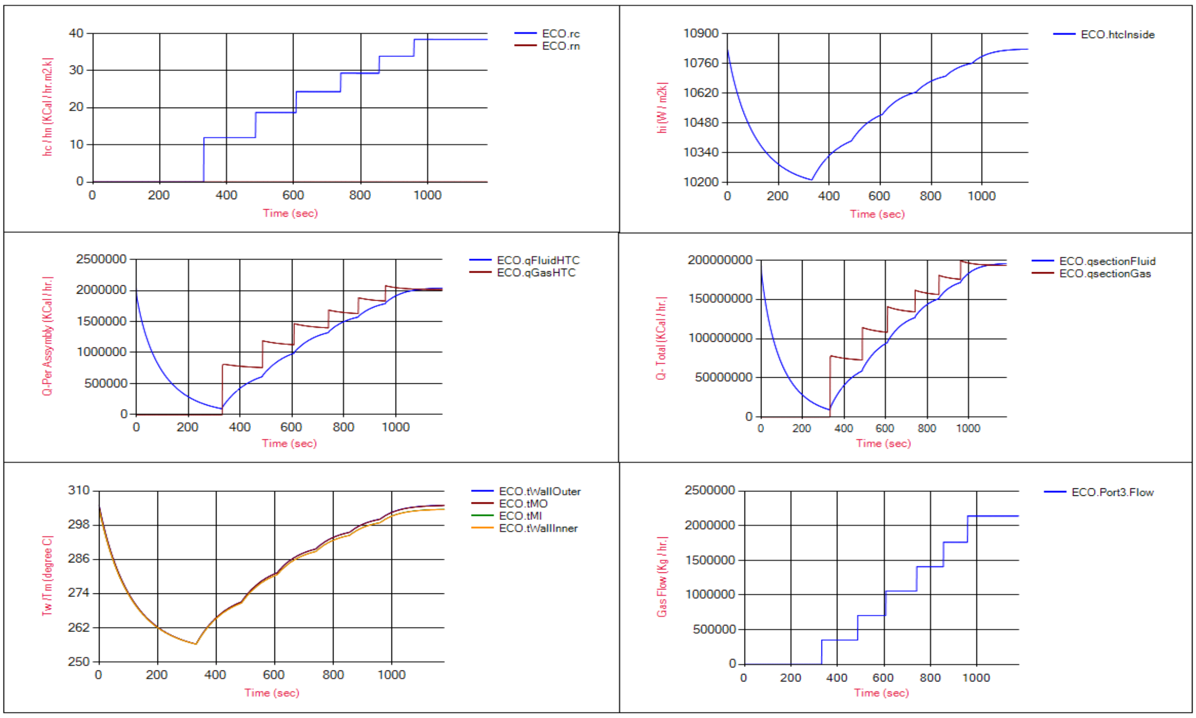


Figure 4.13: Behaviour of various parameters of economizer at varying gas flow

Similary in the economizer by keeping gas flow equal to $2139000Kg/hr.$ and vary the inlet steam temperature from $230°C$ to $280°C$. All the parameters increases with increase in steam temperature, where non luminous heat transfer coefficient is zero. Figure 4.14 shows behaviour of all the parameters at economizer outlet by varying water temperature.

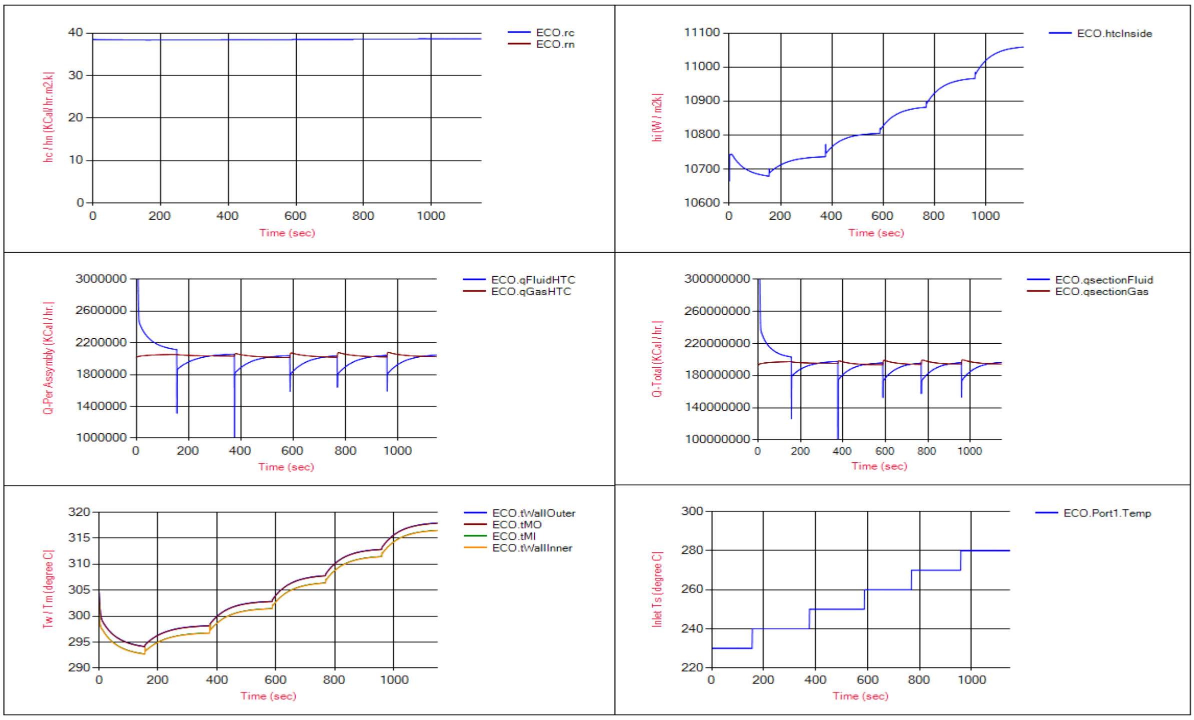


Figure 4.14: Behaviour of various parameters of economizer by varying water temperature

4.8 Turbine casing

For modeling of casing, It is take as cylinderical pipe which is made up of carbon steel inside which steam is flowing. The pipe having outside diameter, length and thickness are approximately $3\,m$,$5\,m$ and $0.05\,m$ respectively. The pipe is surrounded by rockwool insulation having thickness around $0.05\,m$ to minimize the heat loss. The given steam flow, pressure and temperature are shown in the given table 4.7.

Table 4.7: Flow, pressure and temperature inside casing

Inside parameters	Data
Steam flow$(Kg/hr.)$	1480000
Steam pressure(Kg/cm^2)	180
steam temperature$(°C)$	540

4.8.1 Simulation and results obtained for turbine casing

The graph shows the results and behaviour of wall and metal temperatures of casing and insulation applied under steady state. The simulation of turbine casing is shown in given figure 4.15.

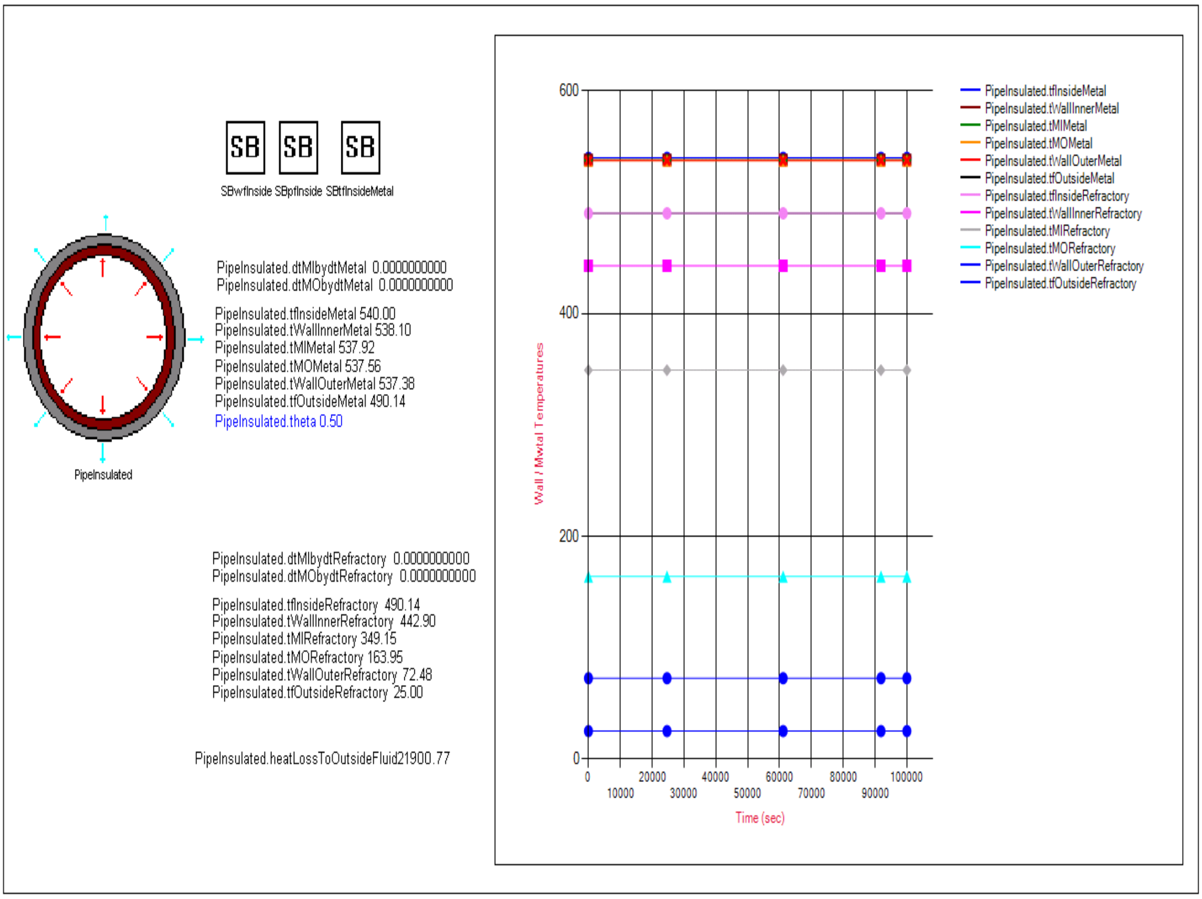


Figure 4.15: Turbine casing model

Note: Here refractory means insulation over casing.

The simulated results of wall/metal temprature of casing and insulation are shown in given table 4.8.

Table 4.8: Simulated temperature of casing and insulation

Temperatures	(*in* $°C$)
Steam temperature	540
Inside wall temperature	538.10
Inside metal temperature	537.92
Outside metal temperature	537.56
Outside wall temperature	537.38
Inside insulation wall temperature	442.90
Inside insulation metal temperature	349.15
Outside insulation metal temperature	163.95
Outside insulation wall temperature	72.48
Ambient temperature	25
Heat loss to surrounding from casing($Kcal/hr.$)	21900.77

Now, We give inside the pipe steam temperature is $480°C$ at same pressure $180Kg/cm^2$ and flow $1480000Kg/hr.$ as previous, so that due to decrease in inside temperature, the wall and metal temperature start comes down and after almost $5009sec.(1hr. and\,22min.)$ the wall temperature also equals to inside steam temperature i.e. $480°C$. Hence, insulation temperatures also comes down. The simulated behaviour of temperatures is shown in the given figure 4.16.

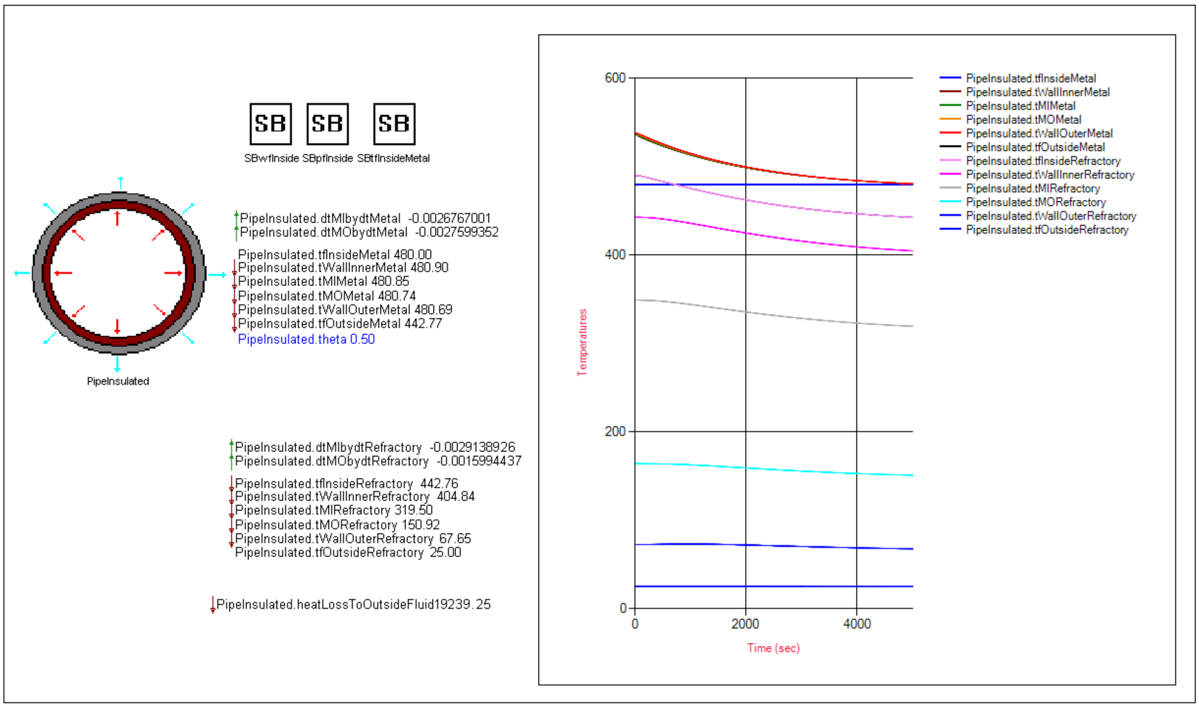


Figure 4.16: Behaviour of casing by changing steam temperature

Chapter 5

Conclusions and Future Work

5.1 Conclusions

In the present study, the dynamic behavior of flue gases to water/steam heat exchangers of a typical 500 MW Thermal power plant components are studied using the mathematical modeling and computer simulation. Various boiler heating sections like superheater, reheater, economizer, boier drum and also turbine casing are modeled. The dynamic and algebraic equations are formulated based on mass balance, heat balance and heat transfer models.The steady state and dynamic equations thus formed are solved using Advanced Continuous Simulation Language (ACSL-X) solver. From the simulation, the obtained wall/ metal temperature was also found good agreement with expected trends. The deviation between simulation results and tested data was found within 0% to 1 % in the case of heat transfer coefficients, 2% to 7% in heat transfer rate and 0% to 0.7% deviation in steam temperature at outlet of superheater and reheater; which shows very good agreement. Inspite boiler heating sections, we also done modeling and simulation for turbine casing where similar methodology applied as heating sections of boiler. The obtained results of turbine casing was also found good agreement with expected trends.

5.2 Future work

The future work is as under:

- To analyse the flow and temperature distribution in the gas side .
- To analyse the flow and temperature distribution in the steam side .
- To analyse the axial conduction between both the elements of a components.
- To simulate the heat transfer analysis and metal load temperature calculations for turbine casing of varying thickness.

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