Design and analysis of the prototype of boiler for steam pressure control

[1]Akanksha Bhoursae, [2]Jalpa Shah, [3]Nishith Bhatt

Institute of Technology, Nirma University, SG highway, Ahmedabad-382481,India
[3]Essar steels limited,Hazira,Surat-394270,India

Abstract – To obtain high energy efficient power plant operation it is necessary to efficiently control the steam pressure. Hence an effort has been made in this paper to control such a critical parameter i.e. steam pressure by developing a prototype of boiler kit using PLC based PID controller which uses IMC technique for tuning the parameters of the PID. Presented work also includes the modeling of the process and simulation has been done with the appropriate transfer function using feed forward feedback control strategy. Further practical responses and theoretical response has been compared. Also open loop validation has been done to validate the prototype model.

Keyword – steam pressure control, IMC based PID tuning, modeling, simulation.

I. INTRODUCTION

The steam boiler is an energy conversion equipment or device which transforms chemical energy of fuel such as coal, oil, gas or nuclear energy into steam which in turn is used for mechanical energy. Boilers are also commonly used in industry to generate a utility steam used for various purposes like heating or drying chemical compounds in a reactor. The primary function of boiler is to maintain the steam energy in balance with the load demand while maintaining the internal variables such as pressure, level in a desired range [1].

Steam pressure is one of the most important parameters for power plants efficiency. High steam pressure and temperature are the necessary conditions for achieving high (designed) efficiency of the power plant. However, both parameters are limited with the constraints of the drum and evaporator materials. Especially, boiler drum is protected with safety valves, taking out excess of the steam on high pressure. Activating this kind of protection makes large disturbance to the drum level control and leading to the plant fallout and loss of several thousand. At the other side, stable and smooth steam pressure is a necessary condition for high quality steam temperature and turbine control. Accordingly, steam pressure control is to provide low pressure variations enabling in that way pressure to be close to the safety limit, and not to exceed it [2].

A number of mathematical models have been developed to describe the dynamics of steam boiler pressure. The goal of this paper is to design a prototype of boiler kit using PLC based PID controller which uses IMC technique for tuning the parameters of the PID. Besides this, work includes the modeling of the process and simulation has been done with the appropriate transfer function using feed forward feedback control strategy. Further practical responses and theoretical response has been compared. Also open loop validation has been done to validate the prototype model.

II. BOILER STEAM PRESSURE CONTROL

The steam drum pressure is an indication of the balance between the inflow and the outflow of heat. Therefore, by controlling the steam supply pressure, one can establish a balance between the demand for steam (process load) and the supply of steam (firing rate). A change in steam pressure will result from a change in firing rate only after a delay of a few seconds to a minute, depending on the boiler and the load level. Therefore, as will be seen later, feed-forward control can improve pressure control by adjusting fuel as soon as a load change is detected, instead of waiting for pressure to change first.

The actual steam flow at any point is not necessarily a true indication of demand. For example, an increase in steam flow caused by increased firing should not be interpreted as a load increase; this would create a positive feedback loop, capable of destabilizing the boiler. According to Shinskey, the true load on a boiler can be approximated by $\sqrt{h/p}$ where, h is the differential developed by a flow element and p is the steam pressure.

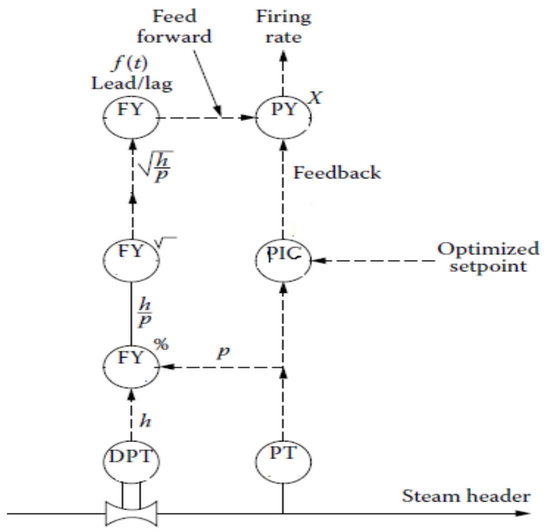

Fig.1. Firing rate determination using feed-forward loop with feedback trim

Fig 1 describes a boiler pressure control system using this type of feed-forward model. Fuel is set proportionally to the estimate of load √h/p. Dynamic compensation is applied in the form of a lead-lag function to help overcome the heat capacity in the boiler.

The pressure controller adjusts the ratio of firing rate to estimate load, to correct for inaccuracy of the model and variations in heat of combustion of the fuel [3].

III. METHODOLOGY

A. Model development

Since steam is taken out of a boiler, it is necessary that the boiler maintain a weight balance of the water inside the drum; next, since the heat is carried out by the steam, it is also necessary to maintain an energy balance.

The control of the water weight balance is done as a feed water flow control; with a drum boiler, since the ratio between the amount of steam and feed water flow appears as a drum water level variation, the controlling is done indirectly as a drum water level control.

The energy balance control, to maintain the pressure of generated steam constant, is done as a combustion output control and a boiler’s output is determined by this combustion output [4].

The mathematical model of the boiler system is described in this section where two main equations has been obtained i.e. the drum level and pressure equations. Both equations consider the level and pressure as state variables, and are obtained using mass and energy balances of the boiler system considering both liquid and steam phases [5].

The following assumptions are made for this model:

- The drum is a perfect cylinder.
- The heat exchange surface between vapor and liquid is planar.
- The water in both phases (liquid and vapor) at the drum is at the saturated conditions.

MASS FLOW RATE BALANCE

Based on mass flow rate balance, the equations are as follows:

$$W_{sf} - W_{fwf} = \frac{\partial}{\partial t}[\rho_v.V_v+\rho_1.v_1] \quad [1]$$

$$W_{sf} - W_{fwf} = v_v \frac{\partial}{\partial t}\rho_v + \rho_v \frac{\partial}{\partial t} v_v + v_1 \frac{\partial}{\partial t}\rho_1 + \rho_1 \frac{\partial}{\partial t} v_1 \quad [2]$$

Where, W_{sf} is the steam flow, W_{fwf} is the feed water flow, V_v is the drum vapor volume, V_1 is the drum liquid density, ρ_v is the drum vapor density, ρ_1 is the drum liquid density.

The drum water density is considered a function only upon the drum pressure due to saturation conditions, for both the liquid and vapor phases. Two quadratic interpolation functions provide an excellent fit to calculate these properties:

$$\rho_v = a_0+a_1 p+a_2 p^2 \quad [3]$$

$$\rho_1 = b_0+b_1 p+b_2 p^2 \quad [4]$$

The partial derivatives of densities with respect to drum pressure, p, are

$$\frac{\partial}{\partial t}\rho_v = a_1 + 2a_2 p = k_1 \quad [5]$$

$$\frac{\partial}{\partial t}\rho_1 = b_1 + 2b_2 p = k_2 \quad [6]$$

Now applying the chain rule in equation (2) to the density terms with respect to time, and considering that the variation of the vapor volume is the same as the variation of liquid volume in the drum, we have

$$W_{sf} - W_{fwf} = v_v \frac{\partial}{\partial P}\rho_v \frac{\partial P}{\partial t} + v_1 \frac{\partial}{\partial P}\rho_1 \frac{\partial P}{\partial t} + \rho_1 \frac{\partial}{\partial t} v_1 + \rho_v \frac{\partial}{\partial t} v_v \quad [7]$$

Substituting values from equations (3)-(6) into equation (7), we obtain

$$W_{sf} - W_{fwf} = v_v k_1 \frac{\partial P}{\partial t} + v_1 k_2 \frac{\partial P}{\partial t} + [\rho_1 - \rho_v] \frac{\partial}{\partial t} v_1 \quad [8]$$

Since the drum is a perfect cylinder so the liquid volume at the drum tank is given as

$$V_1 = \pi R^2 D \quad [9]$$

Where, R is the drum radius, D is the drum liquid level from the internal wall to the center of the drum cylinder.

Taking the derivative of equation (9) with respect to time, we have

$$\frac{\partial}{\partial t} v_1 = \pi R^2 \frac{\partial D}{\partial t} \quad [10]$$

Hence from equation (8), we have

$$W_{sf} - W_{fwf} = v_v k_1 \frac{\partial P}{\partial t} + v_1 k_2 \frac{\partial P}{\partial t} + [\rho_1 - \rho_v] \pi r^2 \frac{\partial D}{\partial t} \quad [11]$$

ENERGY BALANCE

Based on energy balance, the equations are as follows:

$$W_{sf}\, h_v - W_{fwf}\, h_{eo} + Q_{fl} = \frac{\partial}{\partial t}[\rho_1 h_1 v_1 + \rho_v h_v v_v] \quad [12]$$

Where, h_{eo} is the enthalpy of water entering the drum tank, h_v is the drum vapor enthalpy, Q_{fl} is the heat flow rate between the furnace metal and the liquid, h_1 is the drum liquid enthalpy, h_v is the drum vapor enthalpy.

The enthalpies at the drum saturation conditions are also calculated using two quadratic equations of the drum steam pressure:

$$h_v = e_0 + e_1 p + e_2 p^2 \quad [13]$$

$$h_v = f_0 + f_1 p + f_2 p^2 \quad [14]$$

Therefore, the derivatives of enthalpy with respect to drum pressure are:

$$\frac{\partial}{\partial t} h_v = e_1 + 2e_2\, p = k_3 \quad [15]$$

$$\frac{\partial}{\partial t} h_1 = f_1 + 2f_2\, p = k_4 \quad [16]$$

From equation (12), we have

$$W_{sf}\, h_v - W_{fwf}\, h_{eo} + Q_{fl} = v_1 h_1 \frac{\partial}{\partial t}\rho_1 + v_1 \rho_1 \frac{\partial}{\partial t} h_1 + \rho_1 h_1 \frac{\partial}{\partial t} v_1 + v_v h_v \frac{\partial}{\partial t}\rho_v + v_v \rho_v \frac{\partial}{\partial t} h_v + \rho_v h_v \frac{\partial}{\partial t} v_v \quad [17]$$

$$W_{sf}\, h_v - W_{fwf}\, h_{eo} + Q_{fl} = v_1 h_1 \frac{\partial \rho_1}{\partial p}\frac{\partial p}{\partial t} + v_1 \rho_1 \frac{\partial h_1}{\partial t}\frac{\partial p}{\partial t} + \rho_1 h_1 \frac{\partial v_1}{\partial t}\frac{\partial p}{\partial t} + v_v h_v \frac{\partial \rho_v}{\partial t}\frac{\partial p}{\partial t} + v_v \rho_v \frac{\partial h_v}{\partial t}\frac{\partial p}{\partial t} + \rho_v h_v \frac{\partial v_v}{\partial t}\frac{\partial p}{\partial t} \quad [18]$$

Since $v_v = - v_1$ and $v_1 = \pi R^2 D$,

Negative sign in the term corresponding to the drum vapor volume is because the increase or decreases in drum liquid volume.

So from equation (5), (6), (15), (16) and (17), we have

$$W_{sf}\, h_v - W_{fwf}\, h_{eo} + Q_{fl} = \pi r^2 D\, h_1\, k_2 \frac{\partial p}{\partial t} + \pi r^2 D\, \rho_1\, k_4 \frac{\partial p}{\partial t} - \pi r^2 D\, h_v\, k_1 \frac{\partial p}{\partial t} - \pi r^2 D\, \rho_v\, k_3 \frac{\partial p}{\partial t} + \frac{\partial D}{\partial t}[\rho_1\, h_1\, \pi r^2 - \rho_v\, h_v\, \pi r^2] \quad [19]$$

$$W_{sf}\, h_v - W_{fwf}\, h_{eo} + Q_{fl} = [\pi r^2 D\, h_1\, k_2 + \pi r^2 D\, \rho_1\, k_4 - \pi r^2 D\, h_v\, k_1 - \pi r^2 D\, \rho_v\, k_3]\frac{\partial p}{\partial t} + [\rho_1\, h_1\, \pi r^2 - \rho_v\, h_v\, \pi r^2]\frac{\partial D}{\partial t} \quad [20]$$

From equation (11) and (20), we have

$$W_{sf} - W_{fwf} = [v_1\, k_2 - v_1\, k_1]\frac{\partial P}{\partial t} + [\rho_1 - \rho_v]\,\pi r^2 \frac{\partial D}{\partial t} \quad [A]$$

$$W_{sf}\, h_v - W_{fwf}\, h_{eo} + Q_{fl} = [h_1\, k_2 + \rho_1\, k_4 - h_v\, k_1 - \rho_v\, k_3]\,\pi r^2 D \frac{\partial p}{\partial t} + [\rho_1\, h_1 - \rho_v\, h_v]\,\pi r^2 \frac{\partial D}{\partial t} \quad [B]$$

Let

$A_1 = v_1\, k_2 - v_1\, k_1,\ A_2 = \rho_1 - \rho_v$

$A_3 = [h_1\, k_2 + \rho_1\, k_4 - h_v\, k_1 - \rho_v\, k_3]\ \pi r^2 D$

$A_4 = \rho_1\, h_1 - \rho_v\, h_v$

Above equations A and B are as follows

$$W_{sf} - W_{fwf} = A_1 \frac{\partial P}{\partial t} + A_2\, \pi r^2 \frac{\partial D}{\partial t} \quad [C]$$

$$W_{sf}\, h_v - W_{fwf}\, h_{eo} + Q_{fl} = A_3 \frac{\partial p}{\partial t} + A_4\, \pi r^2 \frac{\partial D}{\partial t} \quad [D]$$

On solving equation C and D, we get the derivatives of drum level and pressure, we obtain

$$\frac{\partial p}{\partial t} = \frac{A4(Wsf - Wfwf) - A2(Wsf\ hv - Wfwf\ heo + Qfl)}{A4A1 - A2A3} \quad [E]$$

On substituting all the required values, we obtain

$$\frac{\partial p}{\partial t} = 3.1334084 \times 10^{-3}$$

Applying Laplace transform to above equation, we have

$$P(S) = \frac{3.1334084 \times 10^{-3}}{S}$$

B. Controller

Control strategies are necessary for any system to perform accurately. PID controller is used in this paper whose parameter is tuned by IMC technique and whose description is as shown below [6] :

A Proportional-Integral-Derivative (PID) controller is a general feedback control loop mechanism widely used in industrial process control systems. A PID controller corrects the error between a measured process variable and the desired set point by calculating the value of error. The corrective action can adjust the process rapidly to keep the error minimal [7].

The PID controller separately calculate the three parameters i.e. the proportional, the integral, the derivative values. The proportional value determines the reaction to the current error. The integral value determines the reaction based on the sum of recent errors as past error. The derivative value determines the reaction based on the rate at which the error has been changing as a future error. By tuning these three constants in the PID controller algorithm, the controller can provide control action designed for specific process control requirements [7].

Some applications may require only one or two parameters of the PID controller to provide the appropriate control on system. A PID controller will be called a PI, PD, P or I controller in the absence of the respective control actions. This is achieved by setting the gain of undesired control outputs to zero. PI controllers are very common, since derivative action is very sensitive to measurement noise and the absence of an integral value may prevent the system from reaching its target value due to control action.

Following are the process used to determine the PID gain parameter:

The IMC based PID structure uses the process model as in IMC design. In the IMC procedure, the controller

Qc(s) is directly based on the invertible part of the process transfer function. The IMC results in only one tuning parameter which is filter tuning factor but the IMC based PID tuning parameters are the functions of this tuning factor. The selection of the filter parameter is directly related to the robustness. IMC based PID procedures uses an approximation for the dead time.

IMC technique for determining the tuning parameters:

Process transfer function is given by,

$$G_p = \frac{4.445 \times 10^{-3}}{s}$$

Add filter and its transfer function is as follows

$$q_s = G_p^{-1} f_s$$

Where, $f_s = \frac{1}{\lambda s+1}$

Then, $q_c = \frac{q_s}{1-q_s G_p}$

Hence,

$$q_c = \frac{s \times 1000}{3.1334 \times (\lambda s + 1) - 1000 \times 3.1334 \times 10^{-3}}$$

$$q_c = \frac{1}{3.1334 \times 10^{-3} \times \lambda}$$

IV. SIMULATION

When steam disturbance is added in the system, than single feedback controller is not enough to control the whole process. So, a feed forward controller is added which removes this disturbances before it enter into the boiler plant. Feed-forward control avoids the slowness of feedback control system. Fig. 2 shows the simulink block diagram of the boiler steam pressure control using feedback and feed forward controller. In this block diagram, the feed forward controller control the steam disturbances present in the boiler. To calculate the parameter of controller for steam disturbances, following calculation have been taken.

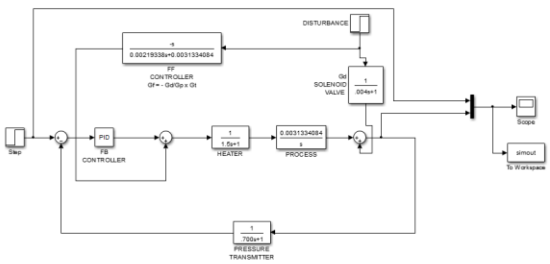

Fig.2. Simulink block diagram of boiler steam pressure control

V. RESPONSES

A. Open loop responses

In the modeling, we get the pure integration process for boiler steam pressure. Through experiment, we came to know that for even a small change in step to integrator in open loop strategy, process value will go to infinity so we have implemented it in close loop so to control the unstable process. Open loop responses of simulink and prototype model have been compared to validate the practical model with the theoretical one and are shown below as:

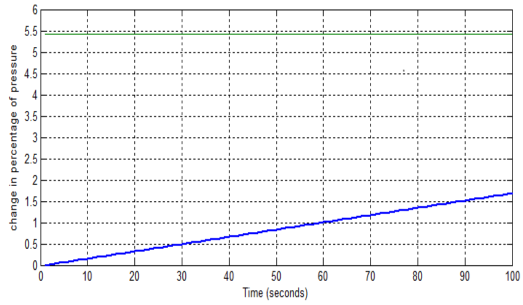

Fig.3. Simulink open loop response

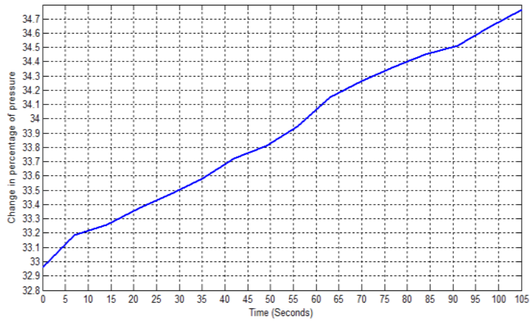

Fig.4. Practical open loop response

B. Closed loop response

For λ = 5, Kp = 45.59, Kd = 478.71, we get the practical response which is as shown below:

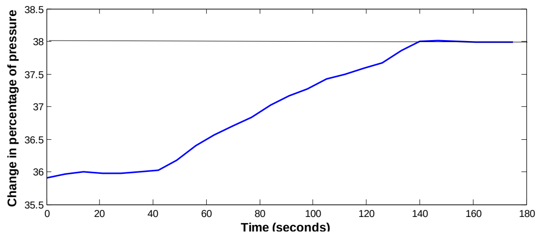

Fig.5. Practical closed loop response

Ideally, for λ = 5, Kp = 45.59, Kd = 478.71, we get the theoretical stable curve which is as shown below:

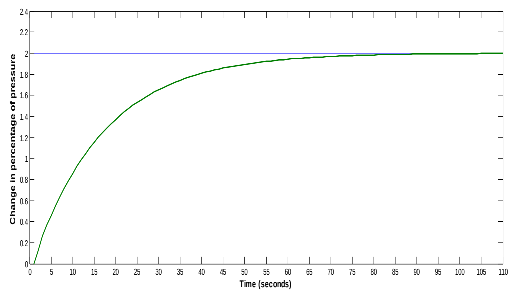

Fig.6. Simulink closed loop response

And the table shown below contains the compared value of simulink and practical response of closed loop,

Table1. Comparison of closed loop response parameters

	Delay time(td)	Rise time(tr)	Max peak overshoot	Settling time(ts)
Ideal	13 sec	40 sec	0	92 sec
Practical	33 sec	77 sec	0	98 sec

VI. CONCLUSION

IMC based PID tuning for $\lambda = 10$ is implemented as it gives zero overshoot. In above table parameters like delay time, rise time, settling time and maximum peak overshoot has shown. As we seen from the table, the practical model gives some deviation in delay time and rise time values. Also, open loop validation has been done successfully to validate the model with the experimental one.

VIII. REFERENCES

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