Temperature Control of Shell and Tube Heat Exchanger System Using Internal Model Controllers

Sarabeevi G.M., Laila Beebi M.
TKM College of Engineering
Department of Electrical and Electronics Engineering
(Industrial Instrumentation and Control)
Kollam-691005, Kerala, India
Sarabv555@gmail.com

Abstract— **In many industrial processes and operations heat exchangers are one of the simplest and important unit for the transfer of thermal energy. There are different types of heat exchangers used in industries; the Shell and Tube heat exchanger system being the most common since it can sustain wide range of temperature and pressure. The main purpose of heat exchanger is to maintain specific temperature conditions, which is achieved by controlling the exit temperature of the process fluid in response to variations of the operating conditions. In this paper, Model based control technique is employed ;Internal Model Controller (IMC) combined with disturbance rejection function and Internal Model based Proportional-Integral-Derivative(PID) controller combined with feed-forward controllers are used to regulate the temperature of outlet fluid of Shell and Tube heat exchanger system to overcome the disturbance occurred due to deviation in the input fluid flow. Simulations are done in the MatLab/Simulink environment.**

Keywords Internal Model Controller,Internal Model based PID controller,Feed-forward controller

I. INTRODUCTION

Heat exchangers are commonly used equipments in industrial chemical processes used to transfer heat between two process streams. They are indispensable part of process industries. Hence researchers are focusing on the system for many years.The setpoint tracking and the disturbance rejection using intelligent control strategies are investigated by Anna Vasickaninova and Monika Bakosov in 2015.Simulations of control of the tubular heat exchanger are done in the MatLab/Simulink environment using complex control structure with two controllers, is compared with the conventional PID control, fuzzy control and NNPC[1]. Robust control of heat exchangers are done by coefficient diagram method [4]. An artificial neural network aided fuzzy logic controller for simultaneous control of indoor air temperature and humidity was developed and is reported in [9]. In 2012, Simon van Mourik, Dirk Vries, Johan P.M. Ploegaert, Hans Zwart, Karel J. Keesman [10], combined methods for model reduction and parameter estimation, such that the frst-order dynamics of a system are in analytical form and are adjusted using experimental data. In 2010, Naseer A. Habobi [6], analyzed the dynamics of cross flow Shell and Tube heat exchanger and is modeled from step changes in cold water flow rate. In 2002, Tony Kealy, Aidan O'Dwyer [11], discussed the estimation of the parameters of a first order plus dead-time process model using the closed-loop step response data of the process under Proportional plus Integral (PI) control. The proportional gain and the integral time, in the PI controller, are chosen such that the closed-loop step response exhibits an under-damped response.

From the literature review we can understand that the design of a practical effective controller is a difficult task, since heat exchangers are highly nonlinear complex systems with time delays that alter the operating conditions. The other factor which adversely affect the operation of heat exchanger system is the presence of disturbances. There can be two types of disturbances in this process, one is the flow variation of input fluid and the second is the temperature variation of input fluid. But in practice the flow variation of input fluid is a more prominent disturbance than the temperature variation in input fluid which cause the output temperature to deviate from the desired value[2]. This factor produce divergence from nominal operating conditions. Also the presence of time delay causes unwanted impacts on the system under consideration which imposes strict limitation on achievable or targeted performance. The objective of this work is to design suitable controllers to maintain the temperature of the outgoing fluid of a Shell and Tube heat exchanger system to a desired setpoint, despite of disturbance like deviation in input fluid flow.

Our paper presents an advanced control strategy that uses the Internal Model Controller and Internal Model based PID controller. This paper considers a single-input single-output model of Shell and Tube heat exchanger.

II. MATHEMATICAL MODELING OF HEAT EXCHANGER SYSTEMS

A. Modeling of Shell and Tube Heat Exchanger

From material balance equation,

$$\frac{dV\rho}{dt} = F_i\rho_i - F\rho \qquad (1)$$

$$\frac{dV}{dt} = F_i - F \qquad (2)$$

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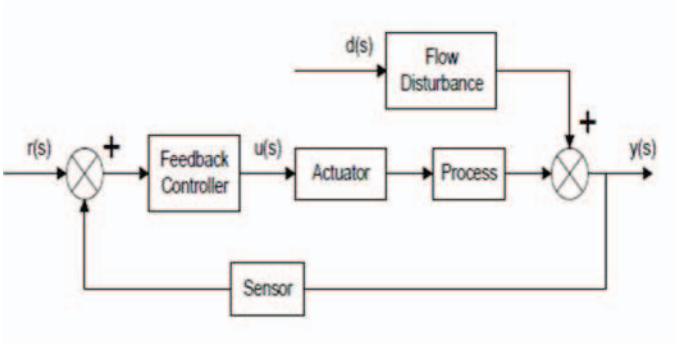

Fig 1: Block diagram of temperature control loop

From energy balance equation by neglecting the kinetic and potential energy contributions .

$$\frac{dU}{dt} = F_i\rho_i\bar{U}_i\text{-}\ F\rho\bar{U} + Q + W_T \qquad (3)$$

Total work done on the system is a combination of the shaft work (W_S) and the energy that the system performs on the surroundings to force the fluid out.

$$W_T = W_S + F_i\rho_i - F\rho \qquad (4)$$

This allows us to write (4) as

$$\frac{dU}{dt} = F_i\rho_i(\bar{U}_i + \frac{p_i}{\rho_i})\text{ - }F\rho(\bar{U} + \frac{p}{\rho}) + Q + W_S \qquad (5)$$

And since H=U+ pV, and$\bar{H} = \bar{U} + p\bar{V} = \bar{U} + \frac{p}{\rho}$,
we can write (5) as

$$\frac{dH}{dt} - \frac{dpV}{dt} = F_i\rho_i\bar{H}_i\text{-}\ F\rho\bar{H} + Q + W_S \qquad (6)$$

Neglecting pressure-volume changes, we find

$$\frac{dH}{dt} = F_i\rho_i\bar{H}_i\text{-}\ F\rho\bar{H} + Q + W_S \qquad (7)$$

The total enthalpy term is

$$H = V\rho\bar{H}$$

And assuming no phase change, we select an arbitrary reference temperature (T_{ref}) for enthalpy

$$\bar{H}(T) = \int_{T_{ref}}^{T} c_p\,dT \qquad (8)$$

Often we assume that the heat capacity is constant, or calculated at an average temperature

$$\bar{H}(T) = c_p(T - T_{ref} \qquad \underline{(9a)}$$

$$\bar{H}_i(T) = c_p(T_i - T_{ref}) \qquad (9b)$$

Hence energy balance (7) in the following fashion,

$$\frac{dV\rho c_p(T - T_{ref})}{dt} = F_i\rho_i c_p(T_i - T_{ref})\text{-}\ F\rho c_p(T - T_{ref}) + Q + W_S$$

Expanding the derivative term and assuming that the density is constant, we have

$$V\rho c_p\frac{d(T - T_{ref})}{dt} + \rho c_p(T - T_{ref})\frac{dV}{dt}$$
$$= F_i\rho c_p(T_i - T_{ref}) - \ F\rho c_p(T - T_{ref}) + Q + W_S \qquad (10)$$

From (2), (10) can be written as

$$V\rho c_p\frac{d(T - T_{ref})}{dt} + \rho c_p(T - T_{ref})F_i - F = F_i\rho c_p(T_i - T_{ref})\text{-}$$
$$F\rho c_p(T - T_{ref}) + Q + W_S$$

Cancelling common terms gives

$$V\rho c_p\frac{d(T - T_{ref})}{dt} = F_i\rho c_p(T_i - T) + Q + W_S \qquad (11)$$

But T_{ref} is a constant, so $\frac{d(T - T_{ref})}{dt} = \frac{dT}{dt}$ .Also neglecting W_S (which is significant only for very viscous fluids),

we can write

$$V\rho c_p\frac{dT}{dt} = F_i\rho c_p(T_i - T) + Q \qquad (12)$$

Which yields the two modeling equations

$$\frac{dV}{dt} = F_i - F \qquad (13)$$

$$\frac{dT}{dt} = \frac{F_s}{V_s}(T_{is} - T) + \frac{Q}{V_s\rho c_p} \qquad (14)$$

Where the subscript 's' is used to indicate that a particular variable remains at its steady state value. Defining the following deviation variables

$$u = Q - Q_s$$
$$y = T - T_s \qquad (15)$$

Equation (14) can be written in the form

$$\frac{V_s}{F_s}\frac{dy}{dt} = -y + \frac{u}{F_s\rho c_p}$$

Or

$$\frac{dy}{dt} = -y + K_p u \qquad (16)$$

Where the parameters of the first-order model are

$$K_p = \frac{1}{F_s\rho c_p} = \text{process gain}$$

$$\tau_p = \frac{V_s}{F_s} = \text{time constant}$$

From (15), we can obtain the transfer function of the system as,

$$F(s) = \frac{Y(s)}{U(s)} = \frac{K_p}{\tau_p s+1}. \quad (17)$$

One of the major characteristics of heat exchanger process is the presence of time delay.

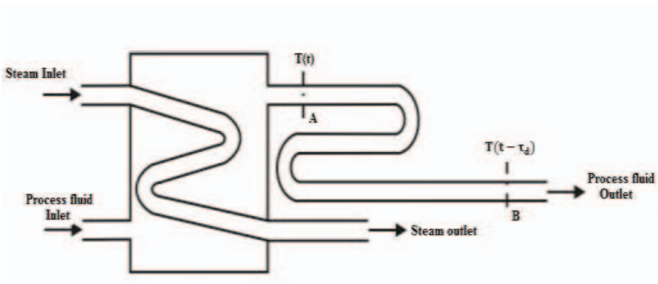

Fig 2.Time delay of heat exchanger system

The transducer should be placed at a location in the water outlet line just after the tank (location A in Fig .2). But suppose, due to the space constraint, the transducer was placed at location B, at a distance L from the tank. In that case, there would be a delay sensing this temperature. If *T(t)* is the temperature measured at location A, then the temperature measured at location B would be $T(t-\tau_d)$. The time delay term τ_d can be expressed in terms of the physical parameters as: $\tau_d = \frac{L}{V}$where L is the distance of the pipeline between locations A and B; and V is the velocity of fluid through the pipeline [7][8].By taking the Laplace Transformation,

$$\mathrm{L}f(t-\tau_d) = e^{-s\tau_d}F(s) \quad (18)$$

Thus the transfer function of the shell tube heat exchanger, from (17) can be written as

$$F(s) = \frac{Y(s)}{U(s)} = \frac{K_p}{\tau_p s+1} e^{-s\tau_d} \quad (19)$$

B. Modeling of Valve Top and Positioner

The position of the stem (or, equivalently, of the plug at the end of the stem) will determine the size of the opening for flow and consequently the size of the flow (flow rate) determined by the balance of all forces acting on it [6][8].

$$P_d\mathrm{A} = \mathrm{kx} + \mathrm{C}\frac{dx}{dt} \quad (20)$$

These forces are:

P_dA=Force exerted by the compressed air at the top of the diaphragm

kx =Force exerted by the spring attached to the stem and the diaphragm

$\mathrm{C}\frac{\mathrm{dx}}{\mathrm{dt}}$=Frictional force exerted upward and resulting from the close contact of the stem with valve packing

where: A – Area of the Diaphragm, P_d– Pressure acting on diaphragm, x – Displacement, k – Hook's constant

The Transfer Function Of the valve from (20) is

$$\frac{K_V}{1+TS} \quad (21)$$

Let the control valve has a maximum travel of 1.56mm, and time constant of 3 sec. The nominal pressure range of the valve is 3 to 15 psig.

$$\text{Control val vegain} = \frac{\text{Range of stem}}{\text{Pressure range}} = 0.13\text{mm/psig}$$

Hence the Control valve transfer function is given by,

$$G_v = \frac{0.13}{3S+1} \quad (22)$$

$$\text{Actuator gain} = \frac{\text{Range of pressure}}{\text{Range of temperature}} = 0.75\ \text{psig/mA}$$

C. Modeling of RTD sensor

The sensor and transmitter can be represented by a first order differential equation as shown below [8],

$$\tau\frac{dI}{dt} + I = kT \quad (23)$$

Where, I is the current signal from the temperature sensor/transmitter to the controller and T is the temperature of exit process fluid. Therefore transfer function of the sensor is,

$$G(S) = \frac{I(S)}{T(S)} = \frac{k_v}{\tau_s S+1} \quad (24)$$

In the system, 2-wire RTD which is calibrated to a range of 0^0C and 100^0C and a time-constant of 10 Sec. The steady state gain k_v can be obtained by the equation,

$$\mathrm{k_v} = \frac{\text{Range of temperature}}{\text{Range of pressure}} = 0.16\mathrm{mA}/^0\mathrm{C}$$

Hence transfer function of the sensor is,

$$G_s = \frac{0.16}{10S+1} \quad (25)$$

D. Modeling Disturbance Transfer Function

The heat exchanger is filled at a flow rate of $\mathrm{Q_{in}}\ \mathrm{m^3/s}$ which is the input to the system. The output is the discharge flow rate, $\mathrm{Q_{out}}\ \mathrm{m^3/s}$. If $\mathrm{Q_{in}} = \mathrm{Q_{out}}$, the level h, remains constant. If $\mathrm{Q_{in}} > \mathrm{Q_{out}}$, the level h, rises. If $\mathrm{Q_{in}} < \mathrm{Q_{out}}$, the level, h, falls. This much is obvious but what exactly is the relationship between the flow in, the flow out and the level.The following equation is a mass balance that can be applied to any system:

In - Out = Accumulation (26)

In this case, the accumulation manifests itself as an increase or a decrease in volume.Accumulation is the change in volume with time.

$$Q_{in} - Q_{out} = \frac{\Delta V}{\Delta t} \quad (27)$$

Volume, V = area x height = A x h. The diameter, d, is a constant. Therefore, h is the only variable which means that $\Delta V = \Delta h \times A$ .Writing the equation in differential form, we have:

$$Q_{in} - Q_{out} = A\frac{dh}{dt} \quad (28)$$

Next, consider the output flowrate, Q_{out}.The driving force for the discharge flow is the head of water in the tank which is given by ρgh. The restriction to the discharge flow is the presence of the valve (and to a lesser extent the pipe) and this can be represented by a resistance, R, i.e.

$$Q_{out} = \frac{\rho gh}{R} \quad (29)$$

Therefore, the above equation can be rewritten as follows:

$$\frac{AR}{\rho g}\frac{dh}{dt}+h=\frac{R}{\rho g}Q_{in} \quad (30)$$

Examination of this equation reveals that it has the following form:

$$\tau\frac{d\theta_o}{dt} + \theta_o = K\theta_i \quad (31)$$

i.e, It has the first order characteristic where the output, θ_o, is equivalent to height, h; the input, θ_i, is equivalent to the flow in, Q_{in}, and the time constant and gain are as follows:

$$\tau = \frac{AR}{\rho g} = \frac{\pi d^2 R}{4\rho g}$$

$$K = \frac{R}{\rho g}$$

Converting this equation to the frequency domain and rearranging in terms of the output over the input gives the following:

$$\frac{\theta_o(s)}{\theta_i(s)} = \frac{K}{1+\tau s} \quad (32)$$

III INTERNAL MODEL CONTROLLER

Internal Model Controller can be developed by a model-based procedure, where a process model is 'embedded' in the controller. The assumption we are making is that the model is perfect. Process transfer function can be obtained as,

$$G_p(s) = \frac{5e^{-s}}{90s^2+33s+1} \quad (33)$$

Assume that the model is perfect. Separating (33) into invertible and non-invertible parts, we obtain

$$\tilde{G}_p(s) = \left(\frac{5}{90s^2+33s+1}\right)e^{-s} \quad (34)$$

Form an idealized IMC controller and add a filter to make the controller proper so as to make it physically realizable. In order to achieve improved disturbance rejection we can use filter transfer function in the form of $f(S) = \frac{\gamma s+1}{(1+\lambda s)^n}$ and 'n' is chosen to make the controller proper. 'λ' should be taken in such a way that 1/2 to 1/3 of the dominant closed loop time constant.Where γ is selected to achieve good disturbance rejection.In practice γ will be selected to cancel a slow disturbance time constant.Consider the closed loop transfer function for disturbance rejection[8].

$$\frac{1-\tilde{G}_p(s)\tilde{q}(s)}{1+\tilde{q}(s)\left(G_p(s)-\tilde{G}_p(s)\right)}g_d(s) \quad (35)$$

In the case of a perfect model, this results in

$$y(s) = [1 - \tilde{G}_p(s)\tilde{q}(s)]\, g_d(s) \quad (36)$$

The controller using the new filter form,is

$$\tilde{q}(S) = \left[\tilde{G}_p^{\,-}(s)\right]^{-1}\frac{\gamma s+1}{(1+\lambda s)^n}$$

So the output response is

$$y(s) = \left[1 - \tilde{G}_p(s)\left[\tilde{G}_p^{\,-}(s)\right]^{-1}\frac{\gamma s+1}{(1+\lambda s)^n}\right]g_d(s)$$

$$= \left[\frac{(1+\lambda s)^n - \tilde{G}_p^{\,+}(s)(\gamma s+1)}{(1+\lambda s)^n}\right]g_d(s)$$

Neglecting e^{-s} term and let n=2,

$$y(s) = \frac{(2\lambda-\gamma)s\left[\frac{\lambda^2}{2\lambda-\gamma}s+1\right]}{(1+\lambda s)^2}\frac{K_p}{T_d s+1} \quad (37)$$

If we select $\lambda^2/2\lambda-\gamma$ to cancel the process model time constant, T_d, we find

$$\gamma = \frac{2\lambda T_d - \lambda^2}{T_d} \quad (38)$$

For $\lambda = 15, \gamma = 22.5$.

IMC controller cascaded with filter can be written as,

$$\tilde{q}(S) = \left(\frac{90s^2+33s+1}{5}\right)\left(\frac{\gamma s+1}{(1+\lambda s)^n}\right) \quad (39)$$

Hence, IMC controller can be obtained as

$$\tilde{q}(S) = \frac{2025s^3+832.5s^2+55.5s+1}{16875s^3+3375s^2+225s+5} \quad (40)$$

IV INTERNAL MODEL BASED PID CONTROLLER

The second order plus dead time process transfer function is reduced to a first order model. For a first order model $T_2 = 0$ and the parameter is given as;

$$T_I = T_{10} + \frac{T_{20}}{2} \quad (41)$$

$$\theta = \theta_0 + \frac{T_{20}}{2} + \sum_{i\geq 3} T_{i0} \sum_j T_{jo}^{inv} \quad (42)$$

By using half rule reduced model is given as

$$\tilde{G}_p(s) = \frac{K_p e^{-\theta s}}{T_I s+1} \quad (43)$$

Hence

$$\tilde{G}_p(s) = \frac{5e^{-2.5s}}{1+31.5s} \quad (44)$$

Using first order pade approximation technique

$$e^{-2.5s} = \frac{-1.25s+1}{1.25s+1} \quad (45)$$

Factor out noninvertible components,

$$\tilde{G}_p(s) = \frac{5}{(1+31.5s)(1+1.25s)} \quad (46)$$

Find the IMC controller transfer function,$q(S)$.Here we allow $q(S)$ to be improper because we wish to end up in an ideal PID controller.

$$q(S) = \left[\tilde{G}_p^{\,-}(s)\right]^{-1}f(S)$$

$$= \frac{(1+31.5S)(1+1.25S)}{5(\lambda S+1)} \tag{47}$$

Find the equivalent standard feedback controller using the transformation

$$g_c(S) = \frac{q(S)}{1-\tilde{g}_p(s)q(S)}$$

$$= \frac{(1+31.5S)(1+1.25S)}{5\lambda S} \tag{48}$$

In order to make the Internal Model Based PID controller shown in (48) to be physically realizable we have to cascade it with a filter in the form $\frac{1}{\tau_F S+1}$ .Hence, the controller becomes

$$g_c(S) = \frac{39.375s^2+32.75s+1}{31.2S(\tau_F S+1)} \tag{49}$$

'λ' should be taken in such a way that 1/3 to 1/5 of the dominant closed loop time constant.Choose $\lambda = 6.24$ and $\tau_F = 10$.

$$g_c(S) = \frac{39.375s^2+32.75s+1}{312s^2+31.2s} \tag{50}$$

V FEED-FORWARD CONTROLLER COMBINED WITH IMC BASED PID CONTROLLER

The transfer function of feed-forward controller can be represented as

$$G_{cf}(S) = -\frac{G_d(S)}{G_p(S)} \tag{51}$$

Here $G_{cf}(S)$ is the transfer function of feedback feed forward controller, $G_p(S)$ is the process transfer function and $G_d(S)$ is the disturbance transfer function [7].The control signal of feedback controller and feed-forward controller is summed up and provided to the process.

$$G_{cf}(S) = \frac{-18S^2-6.6S-0.2}{30S+1} \tag{52}$$

The feed forward controller in (52) is improper, hence it is physically not realizable. In order to make the controller proper,it should be cascaded with a low pass filter in the form $\frac{1}{(\lambda S+1)^n}$.,Hence (52) becomes,

$$G_{cf}(S) = \frac{-18S^2-6.6S-0.2}{300S^2+40S+1} \tag{53}$$

VI. SIMULATION RESULTS AND DISCUSSION

The simulations for the different control mechanisms discussed above were carried out in Simulink and the simulation results have been obtained.

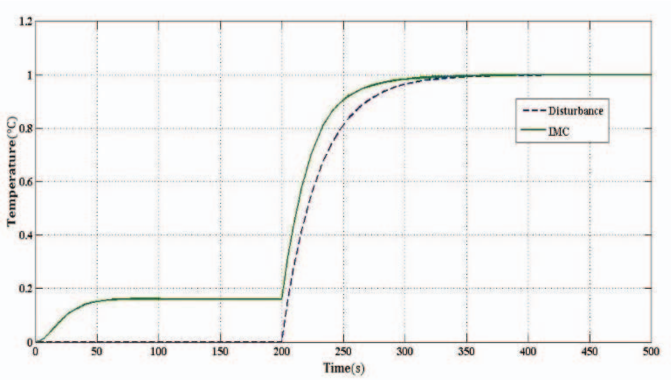

Fig 3. Step response of the system with Internal Model Controller combined with disturbance rejection function

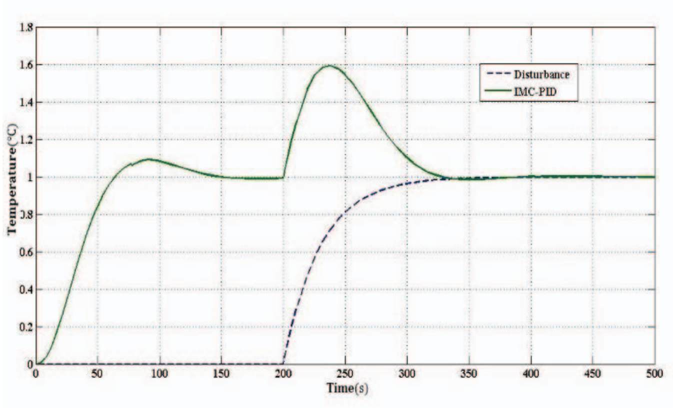

Fig 4. Step response of the system with Internal Model based PID Controller

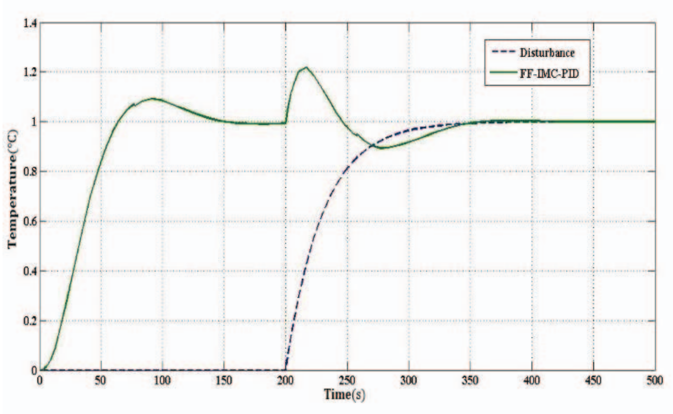

Fig 5. Step response of the system with Internal Model based PID Controller combined with Feed-forward controller

TABLE 1.COMPARISON OF PARAMETERS

	IMC		IMC-PID		FF-IMC-PID	
	Without Disturbance	With Disturbance	Without Disturbance	With Disturbance	Without Disturbance	With Disturbance
Peak Amplitude(^oC)	0.15	1	1.1	1.6	1.08	1.23
Peak Overshoot (%)	0	0	10	60	8	23
Settling Time (Sec)	60	125	150	125	140	125
IAE	38.4		13.72		9.229	
ISE	5.732		1.246		0.8653	

From the above observations we can see that with the controllers in the feedback loop, the heat exchanger produces a peak overshoot of 0% and 10% without disturbance for IMC and IMC-PID method respectively. When the response is settled, a disturbance is applied after 200s to understand how the disturbance is affecting the system under the application of suitable control action. In the presence of disturbance the heat exchanger produces a peak overshoot of 0% and 60% for IMC and IMC-PID method respectively. With IMC controller, setpoint tracking is not achieved; response attains steady state only at 0.15℃.But the effect of disturbance in the system is suppressed by incorporating disturbance function along with IMC. By using IMC-PID,setpoint tracking is improved and the response settled at the desired setpoint value. But it results in high values for time domain specifications and performance indices which is quite undesirable. When feed-forward controller is combined with IMC-PID controller,the parameter values are improved.

VII. CONCLUSIONS

This paper evaluates different methods to control the outlet fluid temperature of Shell and Tube heat exchanger. Three different kinds of controllers are designed and the performances of these controllers are evaluated. Even though the effect of disturbance is suppressed by incorporating disturbance rejection function, IMC fails to overcome the limitation of time delay. Setpoint tracking is also very poor with IMC controller. Hence IMC-PID controller was designed by including the time delay expressed using Pade approximation method. It shows improved setpoint tracking and reduced the effect of time delay. To further improve the efficiency of the system, a feed-forward controller is designed and placed in the forward path of the system;shows improved setpoint tracking and performance indices is observed which increase the efficiency and robustness of the system, gives superior performance in the analysis.

Many directions can be taken to continue this work. If it is desired to create a system model that could be connected to real equipment, then re-writing the code using a language more suited for data acquisition, such as LabVIEW by National Instruments, would be a better option. In this paper, only the outlet and inlet of shell and tube respectively are considered. In addition to this nozzle conditions, the system model can be modified by including the inlet and outlet of shell and tube respectively and thus it can be extended to a MIMO system. The information about hydrodynamic components such as the fan spacing,action of pumps,valves,flow meters can also be studied together to analyze the inuence of this devices in the behaviour of the heat exchanger.

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