

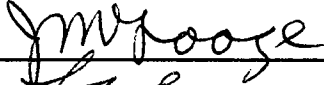
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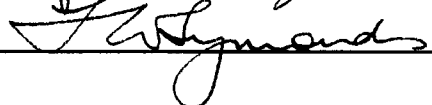
I am submitting herewith a thesis written by Kenneth Edward Swayne entitled "Analysis of Steam Drum Level Control Schemes for an 80,000 pph Industrialized Boiler Model." I have examined the final copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Electrical Engineering.



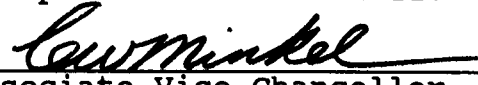
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Date July 24, 1992

ANALYSIS OF STEAM DRUM LEVEL CONTROL
SCHEMES FOR AN 80,000 PPH INDUSTRIALIZED
BOILER MODEL

A Thesis
Presented for the
Master of Science
Degree
The University of Tennessee, Knoxville

Kenneth E. Swayne

August 1992

ACKNOWLEDGEMENTS

In the course of thesis research, a number of scholars assisted me generously. In particular, I am indebted to Dr. James C. Hung, Dr. Frederick Symonds, and Dr. Joseph Googe, who served as my director and committee respectively. With their consultation, criticism, and encouragement, I was able to achieve results otherwise impossible. Whatever errors remain in my work, however, are solely my responsibility.

From my employment at J.M. Huber, I wish to recognize my Supervisor, Joe Crockett, as well as my co-workers, John Coulter, Paul Douglas, and Ken Scannapiego, who offered encouragement and support in the completion of this degree. I wish to thank my secretary, Carol Tolliver, who offered consultation on word processors which enabled the printing of this manuscript. I also wish to thank my parents, Doug and Shirley Swayne, for their encouragement and understanding for the entire duration. And, most of all, I wish to thank my wife, Faith Ann, who stood by my side from the beginning and made numerous sacrifices to make this paper possible.

ABSTRACT

Over the past 28 years, the chemical process defined in this paper has traditionally used a thermostatic regulator for steam drum level control. The plant steam load is cyclic in nature which has caused level variations of ± 4 inches, as determined by high and low alarm positions.

To resolve the poor control performance of the mechanical device, three types of improved control schemes were investigated, namely, feedback, feedforward-feedback, and Smith-Predictor model designs. The problem solving technique involved model development from field data, controller design, and computer simulation of the controllers. A comparison of the simulated controller performance was used to determine the design appropriate for field application.

The feedforward-feedback design was chosen to replace the mechanical system. The controller demonstrated fast response and tight control. The level variation improved to ± 1 inch during load upsets.

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LIST OF SYMBOLS

SYMBOL

A	Cross-sectional Area of Cylindrical Tank
DTC	Deadtime Compensator
d/dt	Rate of Change or Derivative
F_{in}	Volumetric Flow Rate Entering Tank
F_{out}	Volumetric Flow Rate Exiting Tank
$F_{in}(s)$	Laplace Transformation of Volumetric Flow Rate Entering Tank
$F_{out}(s)$	Laplace Transformation of Volumetric Flow Rate Exiting Tank
F_s	Steam Flow
ΔF	Difference of Water Flow and Steam Flow
$G(s)$	General Block Transfer Function
$G_{DTC}(s)$	Dead Time Compensator Transfer Function
$G_{ff}(s)$	Feedforward Compensator
$G_{IRC}(s)$	Inverse Response Compensator Transfer Function
$G_p(s)$	Plant Transfer Function
$G_{p1}(s)$	Plant Transfer Function Component 1
$G_{p2}(s)$	Plant Transfer Function Component 2
h	Level of Tank
Δh	Rectangular Slice Height of Cylindrical Tank
$H(s)$	Laplace Transformation of Tank Level
$\Delta H(s)$	Laplace Transformation of Rectangular Slice Height
H'	Level Control Loop Setpoint
IRC	Inverse Response Compensator
K	Gain Factor

SYMBOL

K_1	Inverse Response Integral Factor Gain
K_2	Inverse Response Lag Factor Gain
K_c	Controller Gain
K_p	Plant Gain
K_v	Control Valve Gain
L	Length of Horizontal Tank
m	Slope of Level Response
M	Mass Accumulated in Tank
M_{in}	Mass Flow Rate Entering Tank
M_{out}	Mass Flow Rate Exiting Tank
ρ	Density of Material in Tank
ρ_{in}	Density of Material Entering Tank
ρ_{out}	Density of Material Exiting Tank
r	Radius of Horizontal Tank
s	Laplace Transformation Operator
t	Time
T_2	Inverse Response Lag Time Constant
T_D	Level Response Deadtime
T_p	Level Response Peak Time
T_R	Level Response Reversing Time
$u(t)$	Unit Step
V	Volume Accumulated in Tank
W	Rectangular Slice Width of Tank
x	Horizontal Coordinate

CHAPTER 1

INTRODUCTION

Although an early form of energy conversion, steam generation has evolved into an essential process for many different industries. Today's technology applies steam as work energy to drive large turbo-generators or as heat energy to complete a chemical reaction. While there are variations in the applications of steam, the fundamentals of the process are essentially unchanging.

Basically, the steam generating process requires one of two vessel types: a firetube boiler, or a watertube boiler. Our discussion will be limited to the latter. A watertube boiler consists of a minimum of two large drums: a steam drum and a mud drum. (Dukelow, 17, 22) These drums are connected by rows of tubes commonly called downcomers and risers. Figure 1-1 illustrates a typical D-type watertube boiler.

Water enters the steam drum and through natural circulation passes through the downcomers to the mud drum. As heat is applied to the tubes, the water in the riser tubes begins to boil and steam is generated in these tubes. The steam travels to the steam drum and exits through the top.

The steam generating process, although simple in nature, demonstrates its true complexity when one is searching for the appropriate control scheme to manage the process. Basically, a boiler is a multivariable process in which the variables

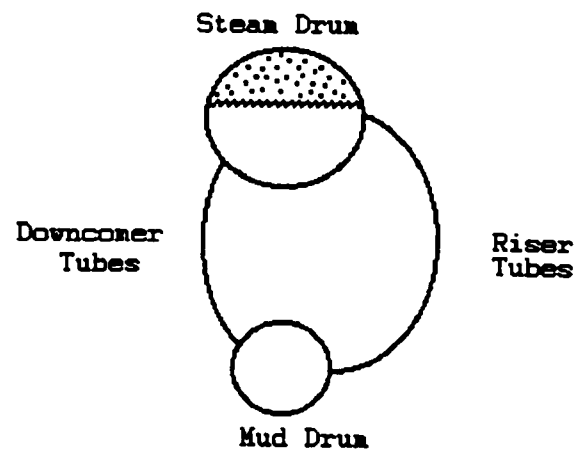


Figure 1-1. Typical D-type Watertube Boiler

typically interact with each other. Traditionally, the variables have been separated by defining a control boundary between the combustion controls and the water inventory controls.

The combustion controls require a steam property such as pressure, temperature, or mass flow to be the controlled variable and the fuel and air flows to be the manipulated variables. On the other hand, the water inventory controls require the steam drum level to be the controlled variable and the feedwater flow to be the manipulated variable.

As the combustion and water inventory controls each have an impact on the efficiency of the energy conversion process, most often the water controls are neglected. Too often, the water inventory controls are viewed as a requirement for protecting the boiler tubes from being damaged. This attitude has often led to a misapplication of traditional proportional level control to steam drum level. A process, such as a chemical batch process, in which steam is required to heat several reactors has a steam load that is typically cyclic in nature. These changes in steam load adversely affect the level of the steam drum and the response of a proportional feedback loop. A single line steam flow diagram of a chemical batch process is illustrated in Figure 1-2.

As noted the steam load is distributed among several reactors which operate asynchronously in an intermittent

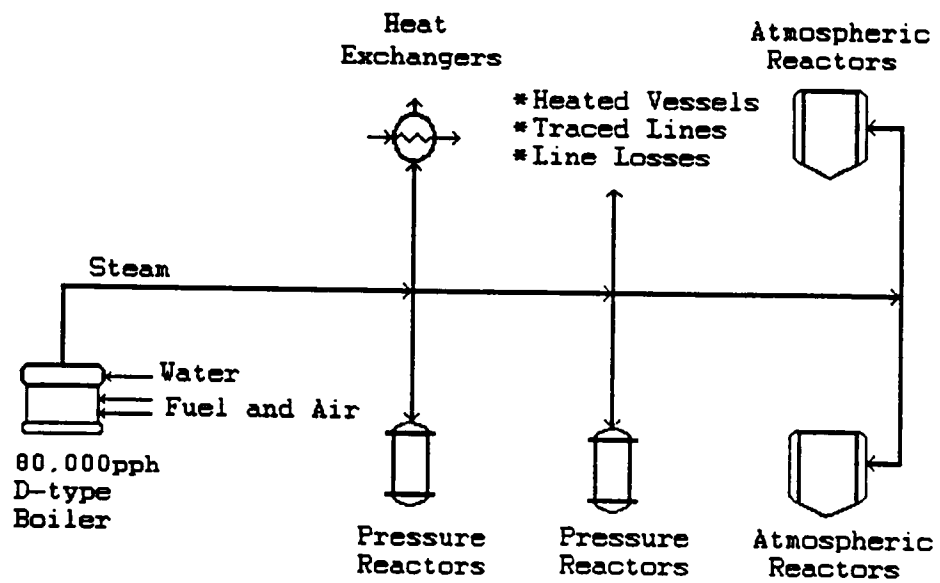


Figure 1-2. Plant Steam Load Distribution Diagram

fashion. The plant operator introduces a nondeterministic random variable, i.e. steam load, as he starts and stops a batch making process. Actually, the randomness could be minimized some by scheduling the on/off times; however, most production processes operate on a demand basis which makes scheduling very difficult. The steam load for the process illustrated in Figure 1-2 is shown in Figure 1-3.

The steam load shown in Figure 1-3 has a definite impact on the level in the steam drum. Developing a control scheme to maintain a relatively constant water level is a distinct challenge when the load disturbance is cyclic.

For 28 years, the plant shown in Figure 1-2 has traditionally used a mechanical thermostatic regulator (Figure 1-4) to control the steam drum level on all boilers.

A thermostatic regulator is a mechanical self-contained proportional control loop. The regulator consists of an inclined tube connected to the steam drum via two taps: a vapor space tap and a water space tap. The end of the tube is connected via linkages to a feedwater control valve. As the steam drum water level drops, the excess steam in the tube causes the tube to expand, which opens the feedwater valve. As the water level increases, the excess water causes the tube to contract, thus closing the feedwater valve. (Dukelow pp 121-122)

A thermostatic regulator has two basic problems when used in a cyclic steam load application:

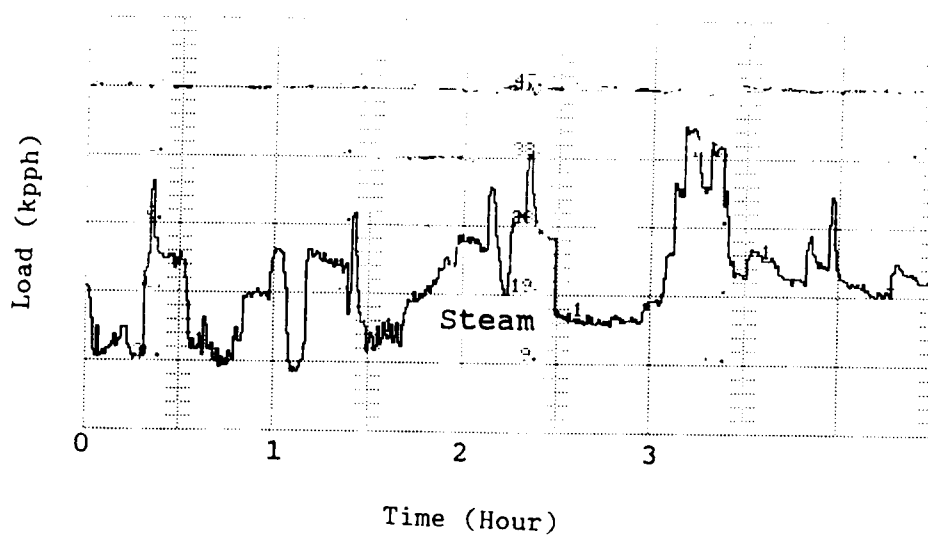


Figure 1-3. Plant Steam Load versus Time

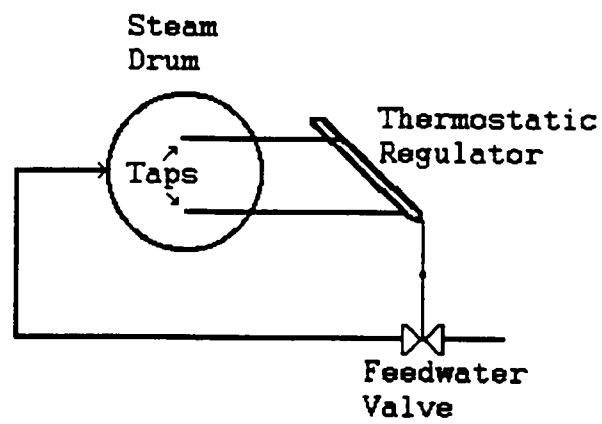


Figure 1-4. Typical Thermostatic Regulator

- (1) The mechanical nature is too slow to respond to the load disturbance.
- (2) Due to the proportional nature of the self-contained controller, the steam drum will always have an offset.

Using the thermostatic regulator for drum level control for the plant described above has resulted in a control band of ± 4 to 5 inches during a sudden load change.

This paper is an effort to demonstrate the operational difference of several drum level control schemes as applied to an 80,000 lb/hr boiler supplying the steam load illustrated in Figure 1-3. The study will investigate the use of a single element controller as a performance reference. An advanced scheme such as two element control will be studied. Finally, a technique known as "Smith-Predictor Model Control" will also be investigated. The problem solving technique will require model development, controller design, and computer simulation of the controllers.

CHAPTER 2

MODEL DEVELOPMENT

Control system design can never be completed unless the process under study is fully understood. There are many tools available to aid in understanding a system; however, a mathematical model is perhaps the most complete method of defining the behavior of any linear system. Linear systems are typically defined by a series of ordinary differential equations which clearly illustrate the process dynamic and steady state characteristics.

The level process, Figure 2-1, is typically a very simple process to define mathematically. This process can be modeled using the following mass balance equation:

$$M_{in} - M_{out} = dM/dt \quad (2-1)$$

where M_{in} = mass flow in, M_{out} = mass flow out, and M = accumulated mass. This equation states that the total mass flowing into the tank minus the total mass flowing out of the tank is equal to the rate of change of the accumulated mass in the tank. More practically, the equation could easily be written in the volumetric form,

$$pF_{in} - pF_{out} = d[pV]/dt \quad (2-2)$$

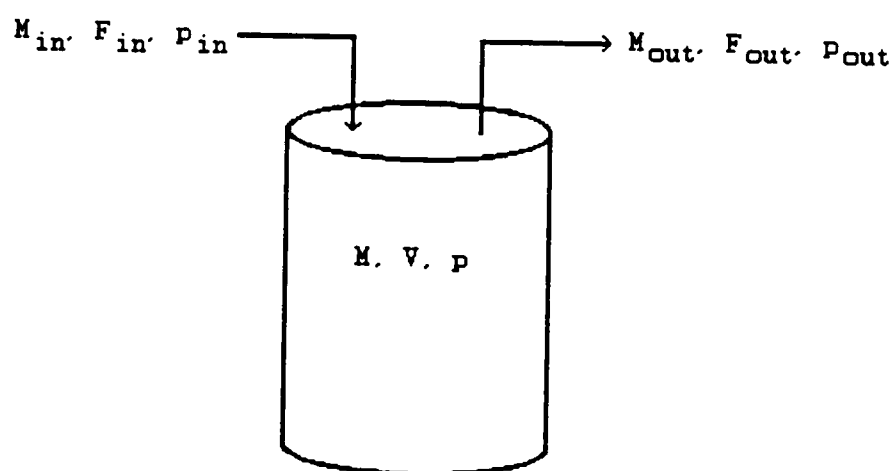


Figure 2-1. Level Process

where F_{in} = volumetric flow in, F_{out} = volumetric flow out, V = volume, and p = density. For an assumption of constant density, p ,

$$F_{in} - F_{out} = dV/dt \quad (2-3).$$

If the ordinary differential equation (ODE) given by (2-3) is linear and F_{in} and F_{out} are independent of tank level, then the level process reduces to an integrating process. A block diagram of this process is shown in Figure 2-2.

In a steam drum application, the tank geometry is typically a horizontal straight cylindrical tank. This type of geometry, Figure 2-3, complicates equation (2-3). Since the cylinder is lying on its side, the level of the tank becomes the tank diameter. This changes (2-3) to a nonlinear differential equation.

Referring to Figures 2-3 and 2-4, the change in volume of the tank is given by

$$dV = L(2x)(dh) \quad (2-4).$$

From Figure 2-4, the equation of a circle is given by

$$x^2 + (h-r)^2 = r^2 \quad (2-5).$$

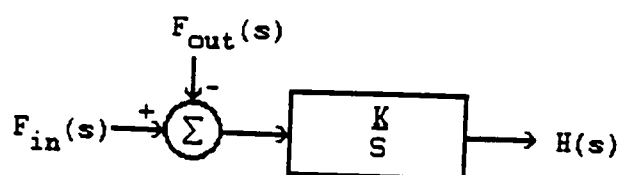


Figure 2-2. Block Diagram of the Level Process

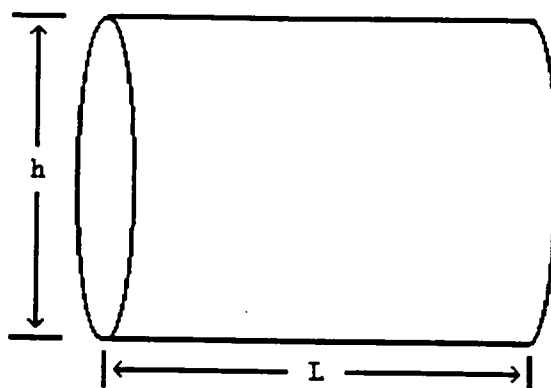


Figure 2-3. Horizontal Cylindrical Tank Geometry

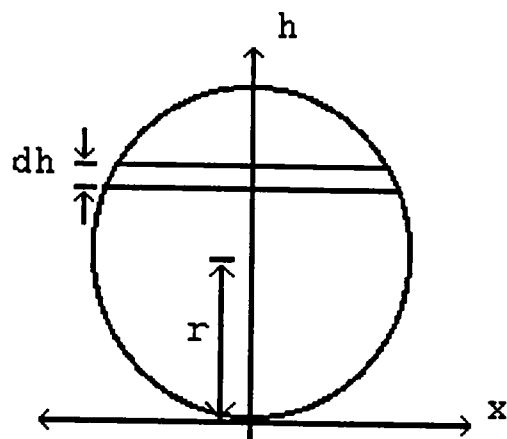


Figure 2-4. Horizontal Cylindrical Tank End View

Solving for x gives

$$x = (2rh - h^2)^{1/2} \quad (2-6).$$

Substituting equation (2-6) into (2-4) results in

$$dV = 2L(2rh - h^2)^{1/2}dh \quad (2-7),$$

and dividing both sides by dt gives

$$dV/dt = 2L(2rh - h^2)^{1/2}dh/dt \quad (2-8).$$

Finally, substituting (2-8) into (2-3) results in

$$2L(2rh - h^2)^{1/2}dh/dt = [F_{in} - F_{out}] \quad (2-9a),$$

or since L is constant

$$(2rh - h^2)^{1/2}dh/dt = K[F_{in} - F_{out}] \quad (2-9b)$$

where $K = 1/2L$.

Equation (2-9b) is a nonlinear differential equation which implies that linear transformations (e.g., Laplace Transform) are no longer applicable.

In order to avoid this nonlinear effect, there are some valid approximations that will aid in linearizing equation (2-9b). Assuming that the system will operate over a narrow range defined by $\Delta[h]$ (Figure 2-5), then the $\Delta[h]$ slice of the cylinder appears as an approximated rectangular volume. The volume of the rectangular box is $\Delta V = WL\Delta[h] = K\Delta[h]$, and by substitution into (2-3) the equation becomes

$$d(\Delta[h])/dt = K[F_{in} - F_{out}] \quad (2-10).$$

Applying Laplace Transformations gives

$$\Delta H(s) = (K/s)[F_{in}(s) - F_{out}(s)] \quad (2-11).$$

The block diagram model of Figure 2-2 holds for equation (2-11).

Once again, the model appears simple, however, for a steam drum level application the model is incomplete. The simple model of (2-11) is complicated by a thermodynamic phenomenon known as "Shrink and Swell." "Shrink and Swell" is a phenomenon that occurs as the steam load disturbance changes. Assume a steady-state condition of constant heat input, normal water level, and water flow equal to steam flow in mass. As the steam flow increases above the water flow, there is a smaller mass of water for the same heat input, which results in a decrease in water density. This density

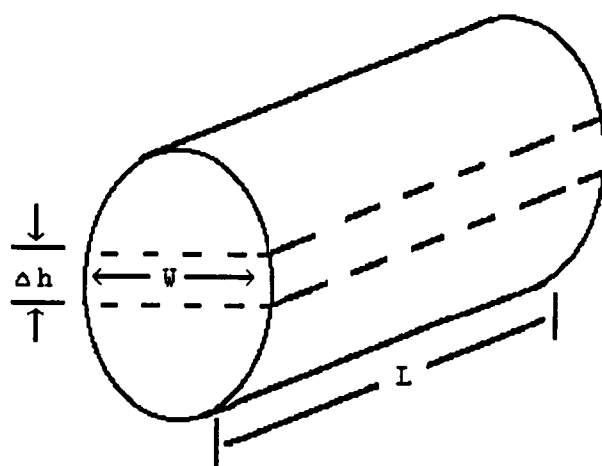


Figure 2-5. Cylinder Approximated as a Rectangular Volume

change is reflected by an increase or "Swell" in water level which lasts for a few seconds. The water level will then begin to decrease according to the integrating nature of the level process. For a decrease in steam flow, there is a similar "Shrink" effect in which the drum level decreases momentarily and then resumes the integrating nature. Figure 2-6 illustrates the "Shrink and Swell" phenomenon for a step change in steam flow.

In order to properly model the steam drum level process, the "Shrink and Swell" effect must be incorporated into the model. This can be done by adding a lag term to equation (2-11) as shown in Figure 2-7. The effect of this term is to provide a correction indicated by the dashed line in Figure 2-6. (Shinskey, 236-237; Stephanopoulos, 390-391) This model has been called the Inverse Response (I.R.) Model.

The parameters, K_1 , K_2 , and T_2 , in the model must be evaluated. Rather than using a structured mathematical formulation, this model can be generated using a combination of formulation and graphical techniques. Figure 2-8 illustrates several step responses of the steam drum level under study. As the steam flow changes stepwise by 10%, the drum level clearly "Swells" as shown in Figure 2-8a and 2-8b or "Shrinks" as shown in Figure 2-8c and 2-8d according to the step direction. The level then begins to linearly increase or decrease in an integrating fashion.

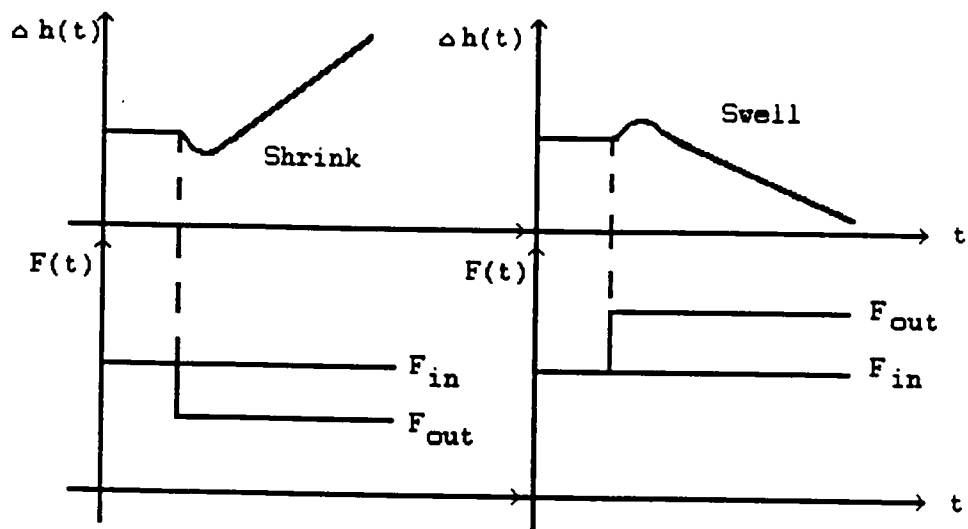


Figure 2-6. Shrink and Swell Effect for a Step Load Change

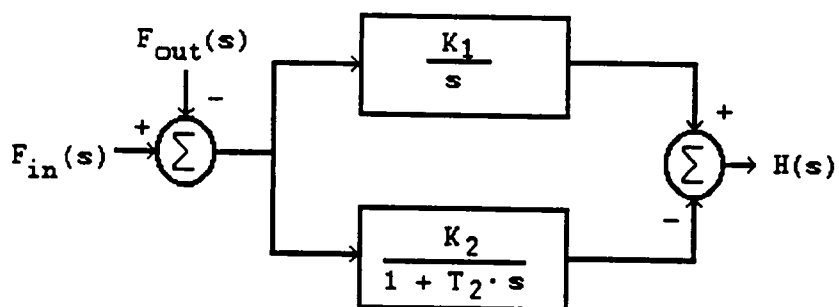
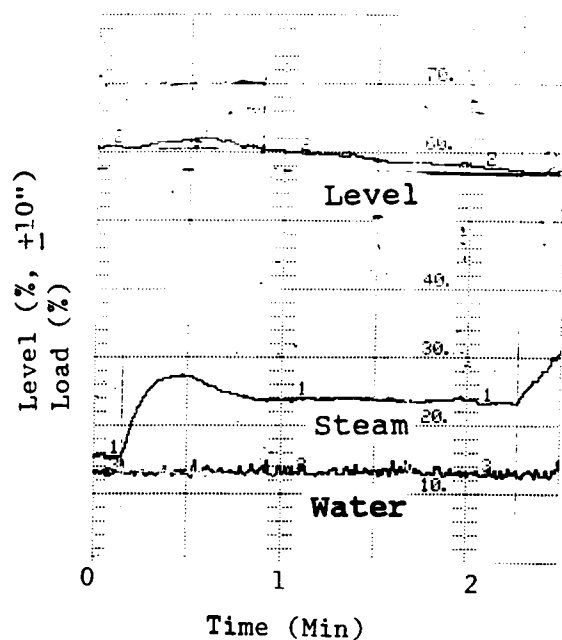
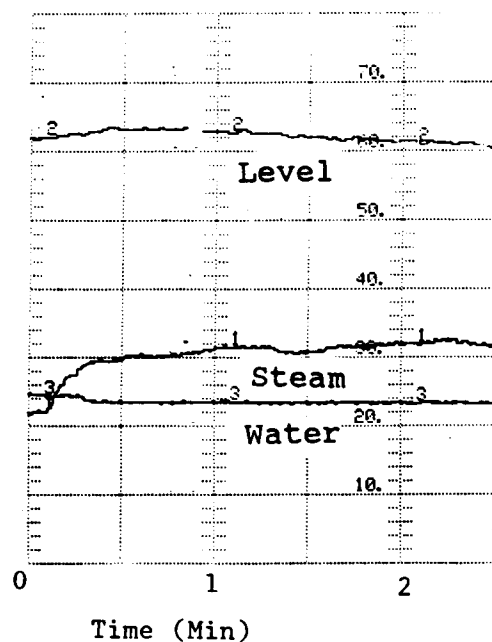


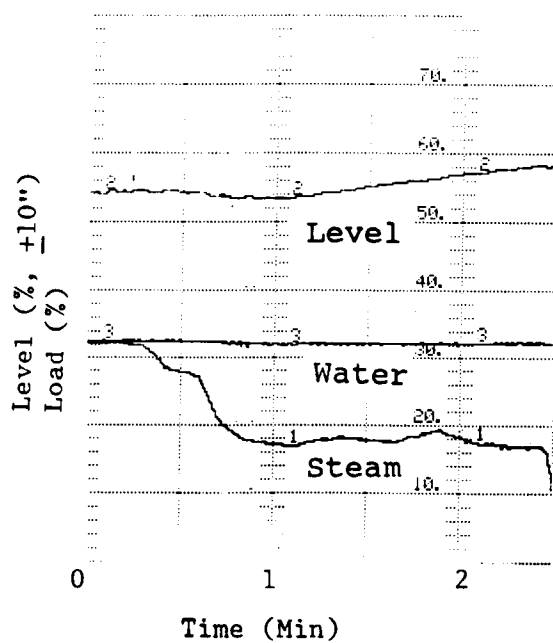
Figure 2-7. Inverse Response Model Block Diagram



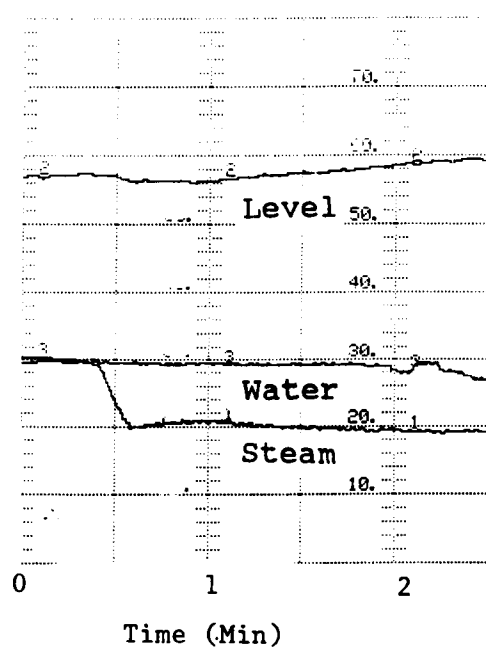
a



b



c



d

Figure 2-8. Swell (a,b) and Shrink (c,d) Effect

To extract a mathematical model from the step response, there are three graphical parameters that must be known: the total reversing time, T_R , the reversing peak time, T_p , and the integrating slope, m . These parameters are illustrated in Figure 2-9. The step response of the I.R. model is given by

$$\Delta H(s) = K_1[1 + (T_2 - K_2/K_1)s]\Delta F/[s^2(1 + T_2s)] \quad (2-13).$$

Transforming (2-13) into the time domain results in

$$\Delta h(t) = \{K_1 t - K_2[1 - T_2 e^{-(t/T_2)}]\}u(t) (\Delta F) \quad (2-14).$$

The reversing starting time is the zero reference for all other times. From the response data, there are two times at which the function $\Delta h(t) = 0$, $t = 0$ and $t = T_R$. First evaluating (2-14) at $t=T_R$ results in

$$\Delta h(T_R) = [K_1 T_R - K_2 + K_2 T_2 e^{-(T_R/T_2)}]\Delta F = 0 \quad (2-15a).$$

Dividing both sides by K_2 yields

$$(K_1/K_2)T_R - 1 + T_2 e^{-(T_R/T_2)} = 0 \quad (2-15b)$$

$$(K_1/K_2)T_R + T_2 e^{-(T_R/T_2)} = 1 \quad (2-15c).$$

Another helpful equation can be derived from noting the peak inverse time and solving the following

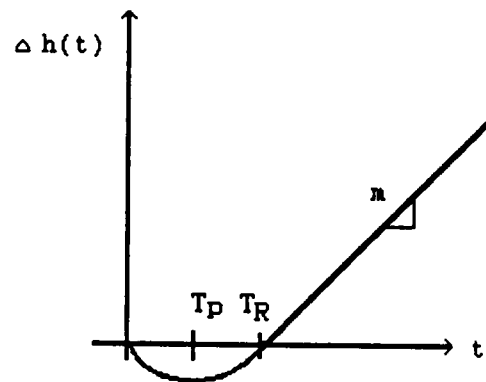


Figure 2-9 Inverse Response Diagram Illustrating the Graphical Parameters

$$d[\Delta h(t)]/dt = 0 \quad , \quad t = T_p \quad (2-16).$$

Substituting for $\Delta h(t)$ in (2-14) gives

$$d[(K_1 t - K_2 - K_2 T_2 e^{-(t/T_2)})\Delta F]/dt = 0 \quad , \quad t = T_p \quad (2-17a)$$

$$[K_1 - K_2 e^{-(T_p/T_2)}]\Delta F = 0 \quad (2-17b)$$

$$(K_1/K_2) = e^{-(T_p/T_2)} \quad (2-17c).$$

Now substitute (2-17c) into (2-15c):

$$[e^{-(T_p/T_2)}]T_R + T_2 e^{-(T_R/T_2)} = 1 \quad (2-18).$$

A final useful equation can be developed using the slope parameter. Since the slope of $\Delta h(t)$ is linear for $t > T_R$, the following equation results

$$d[\Delta h(t)]/dt = m \quad , \quad t > T_R \quad (2-19a),$$

$$[K_1 - K_2 \cdot e^{-(t/T_2)}]\Delta F = m \quad , \quad t > T_R \quad (2-19b).$$

As an example of the use of equations (2-17c), (2-18), and (2-19b), the values of T_p , T_R , and m from Figure 2-8a are 30 sec, 43 sec, and 1/125 inch/sec, respectively. Also, the step change value of steam flow, ΔF , is 10% of full scale. Substituting into the equations gives $K_1 = 1/1250$, $K_2 = 1/29$, and $T_2 = 8$ sec; or in a Laplace model format

$$\Delta H(s) = [1/(1250s)][(1-35s)/(1+8s)] \quad (2-20).$$

A step response for this model is shown in Figure 2-10a. Other step data is organized in Table 2-1, and the corresponding open-loop step response plots are shown in Figures 2-10b, c, and d.

Comparing the simulated model response with the measured data indicates that the model is reasonably accurate. The model will be useful for designing a controller to control the drum level process.

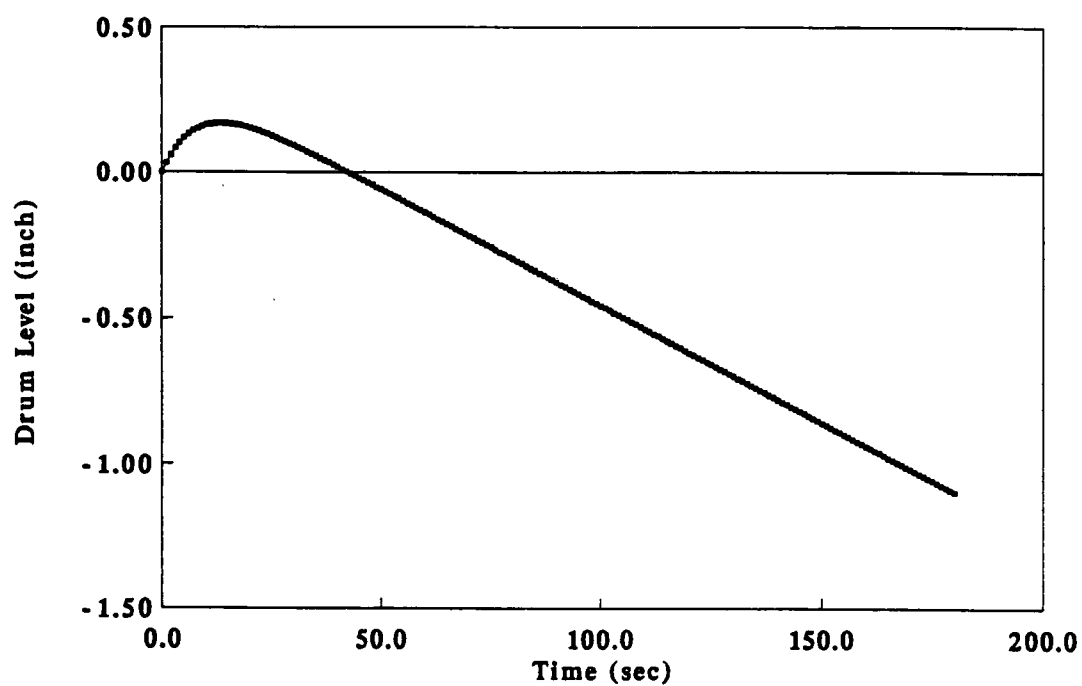
As the Inverse Response model describes the drum level system most accurately, there is another model type which approximates the process, namely, a Deadtime Model. Basically, the deadtime model is similar to the I.R. model with the exception of the reversing effect. The entire reversing time of the "Shrink and Swell" effect is approximated as a deadtime or delay. In fact, the deadtime model is an approximation to the Inverse Response model, which is a delayed integral model,

$$\Delta H(s)/\Delta F(s) = (K_1/s)e^{-(T_0s)} \quad (2-21)$$

where T_0 is the delay time. From Figure 2-9, one sees that $T_0 = T_r$ and $T_0 = K_2/K_1$.

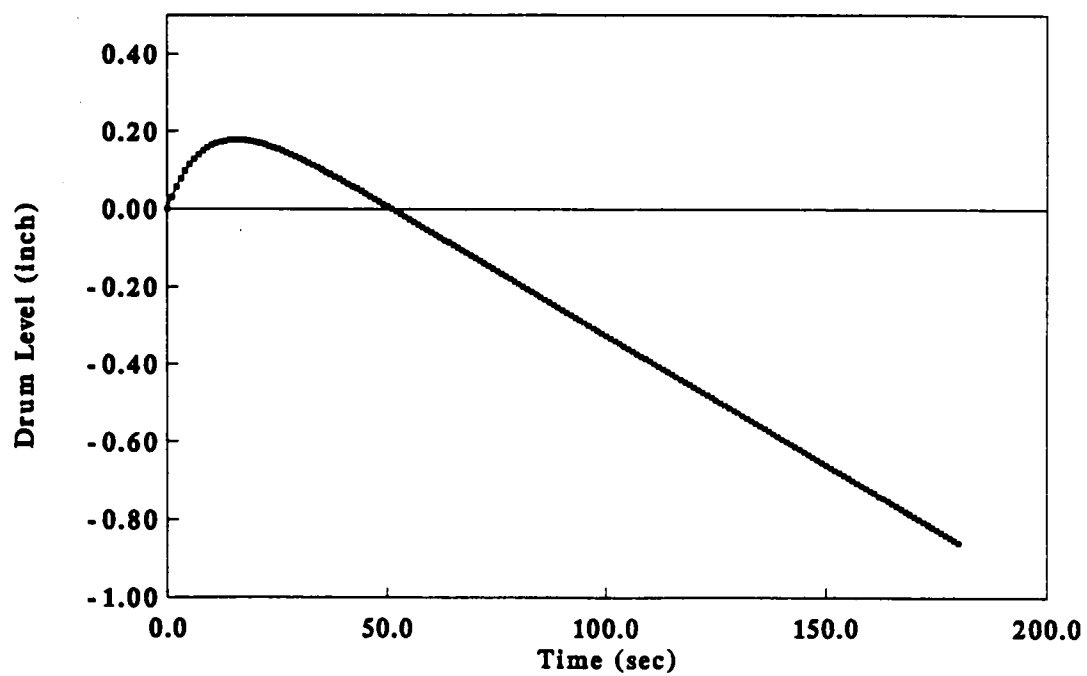
Table 2-1. Inverse Response Model Data for Figure 2-8

Figure	T_p	T_r	n	ΔF	K_2/K_1	T_2	K_1
2-10a	30 s	43 s	.0080	10.2%	43 s	8.0 s	1/1250
2-10b	35 s	50 s	.0067	8.64%	51 s	8.9 s	1/1295
2-10c	30 s	45 s	.0133	15.1%	45.7s	7.9 s	1/1136
2-10d	32 s	46 s	.0083	9.72%	47.2s	8.3 s	1/1167



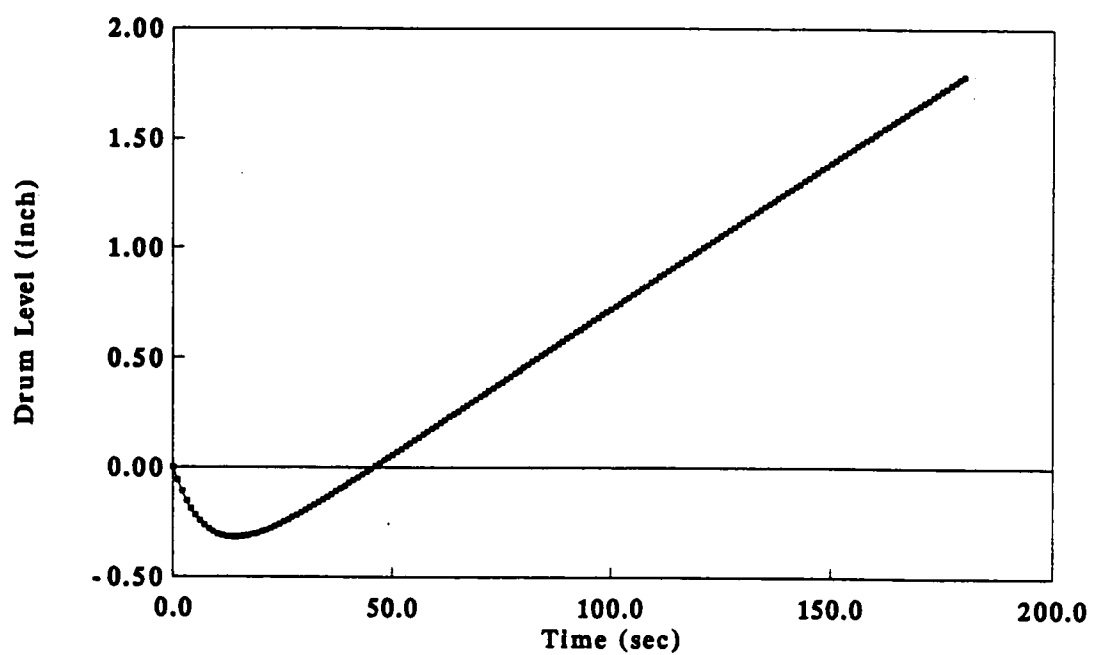
a

Figure 2-10a. Computer Simulation for Inverse Response Model of (a) Figure 2-8a, (b) Figure 2-8b, (c) Figure 2-8c, and (d) Figure 2-8d



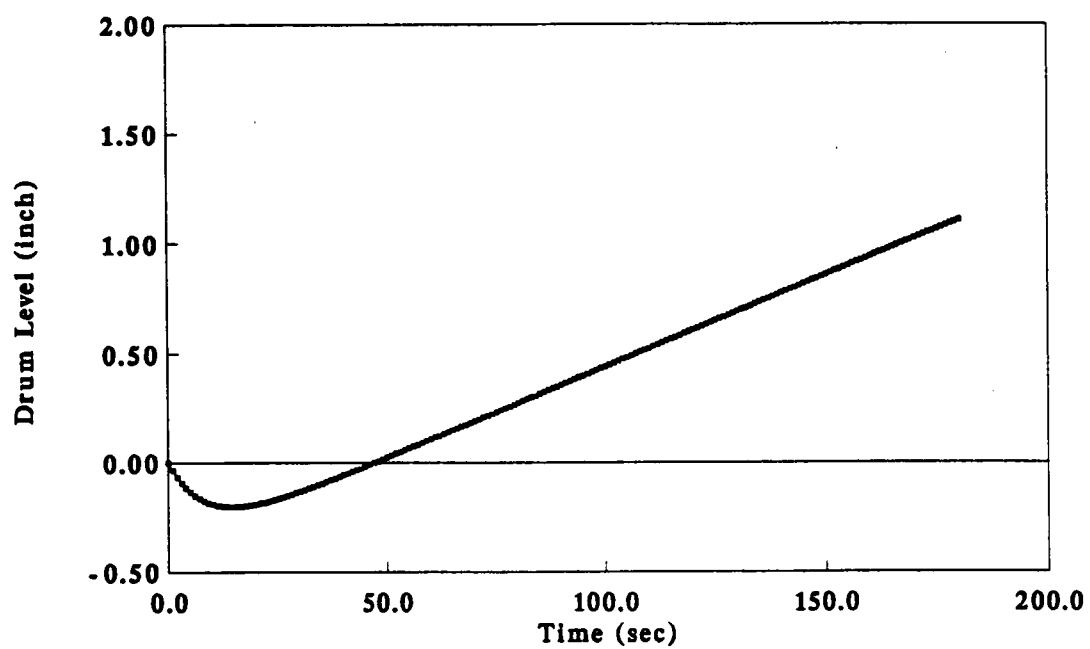
b

Figure 2-10. Continued



c

Figure 2-10. Continued



d

Figure 2-10. Continued

Transforming the step response of (2-21) into the time domain gives

$$\Delta h(t) = K_1[t - (K_2/K_1)]u[t - (K_2/K_1)]\Delta F \quad (2-22).$$

Figure 2-11 shows the deadtime effect for a step change in steam load.

To quantify the deadtime model parameters of the system under study, there are two graphical values that are helpful: the total reversing time, T_R , and the integrating slope, m . Evaluating equation (2-22) at $t = T_R$ gives

$$(K_2/K_1) = T_R = T_D \quad (2-23).$$

Also, solving (2-22) for $t > T_R$ results in

$$K_1\Delta F = m \quad (2-24).$$

An example best illustrates the use of (2-23) and (2-24). From the I.R. model example, T_R and m were previously defined as 43 sec and 1/125 inch/sec, respectively. Inserting these values into the two equations yields $K_1 = 1/1250$ and $K_2 = 1/29$.

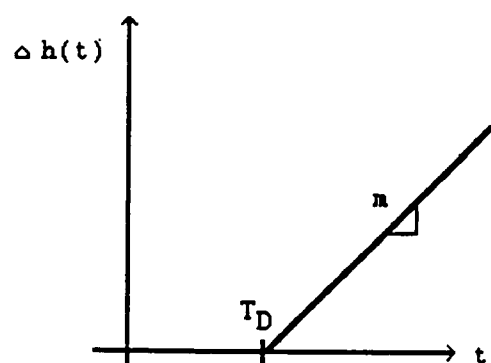


Figure 2-11. Step Response of a Drum Level Deadtime Model

Table 2-2 lists graphical parameters and deadtime model parameters for the step responses of Figure 2-8a, b, c, and d. Using data from Figure 2-8a, the deadtime model is represented as

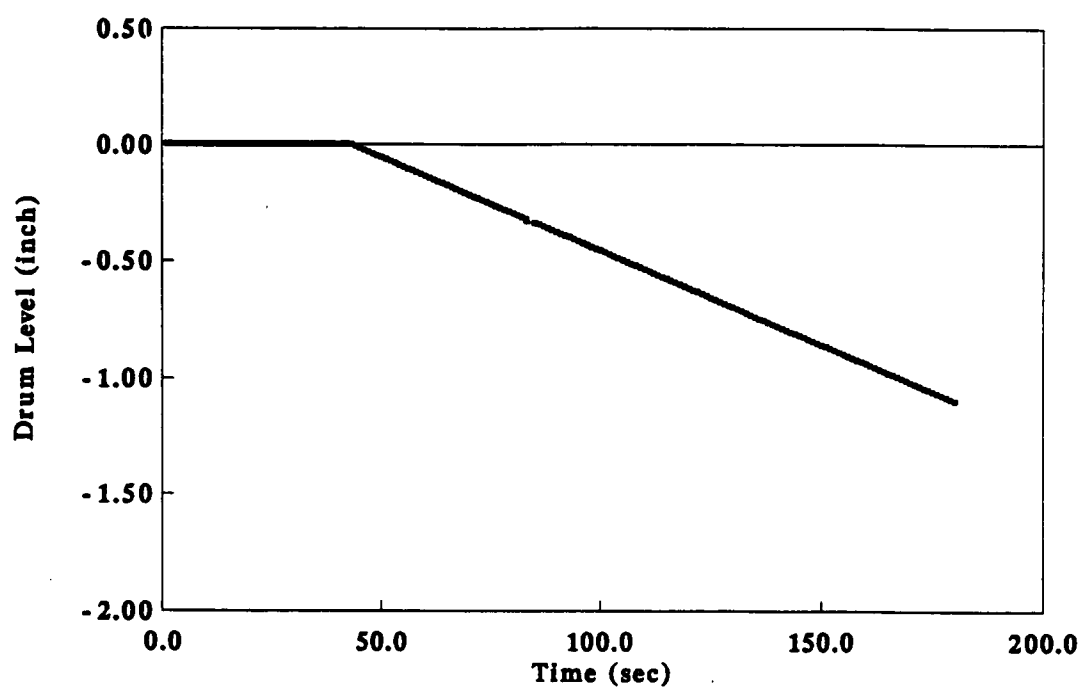
$$G(s) = [1/(1250s)]e^{(-43s)} \quad (2-26).$$

This equation can be expressed graphically as in Figure 2-12.

As stated previously, control system design can never be completed unless the process under study is fully understood. In this chapter, two models have been developed which describe a boiler steam drum level process: the Inverse Response model and the Deadtime model. Each model described well the integral nature of the process; however, the "Shrink and Swell" effect was best described by the Inverse Response model. Therefore, the Inverse Response model will be used throughout the control system design process.

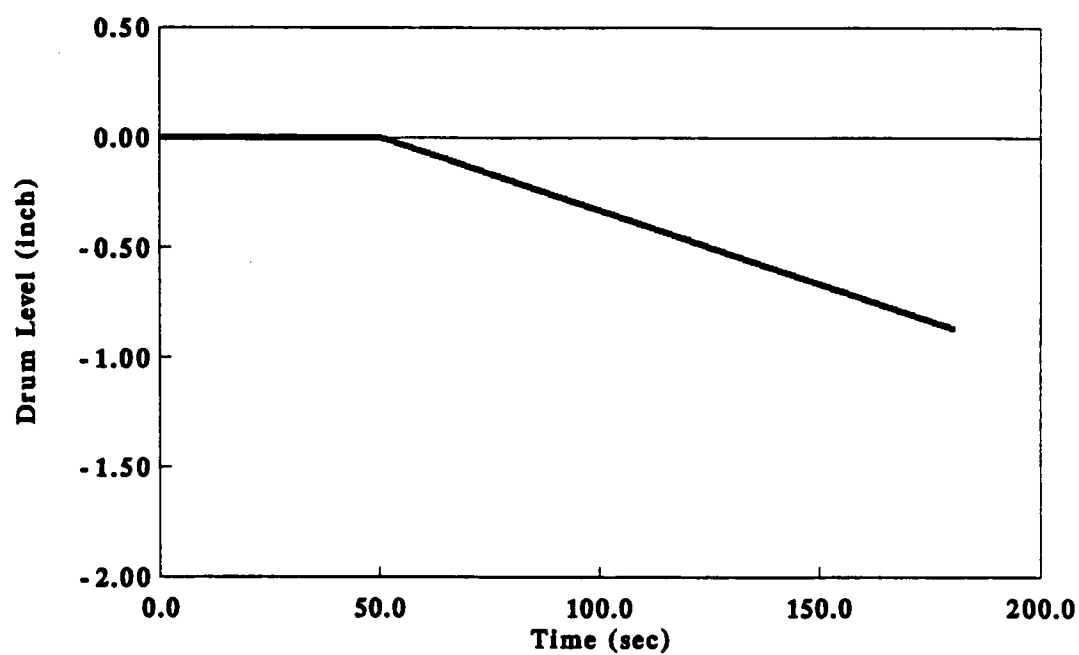
Table 2-2. Deadtime Model Data for Figure 2-8

Figure	T_r	m	ΔF	K_2	K_1	T_D
2-12a	43 s	.0080	10.2%	1/29.7	1/1250	43 s
2-12b	50 s	.0067	8.64%	1/25.9	1/1295	50 s
2-12c	45 s	.0133	15.1%	1/25.2	1/1136	45 s
2-12d	46 s	.0083	9.72%	1/25.4	1/1167	46 s



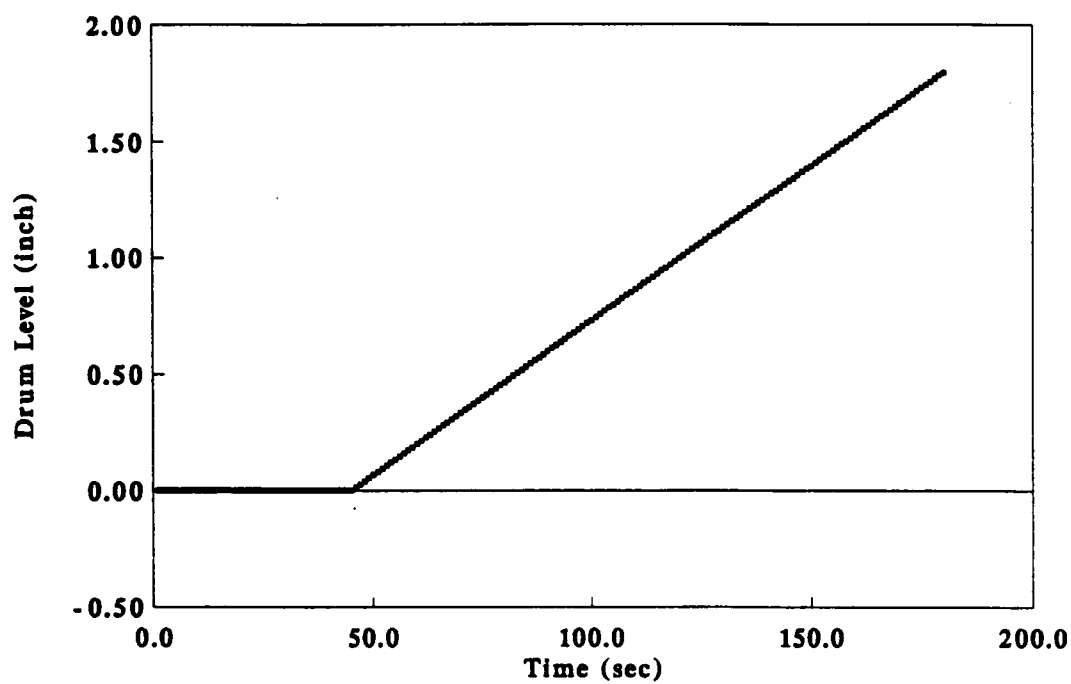
a

Figure 2-12. Computer Simulation of Deadtime Model for (a) Figure 2-8a, (b) Figure 2-8b, (c) Figure 2-8c, and (d) Figure 2-8d



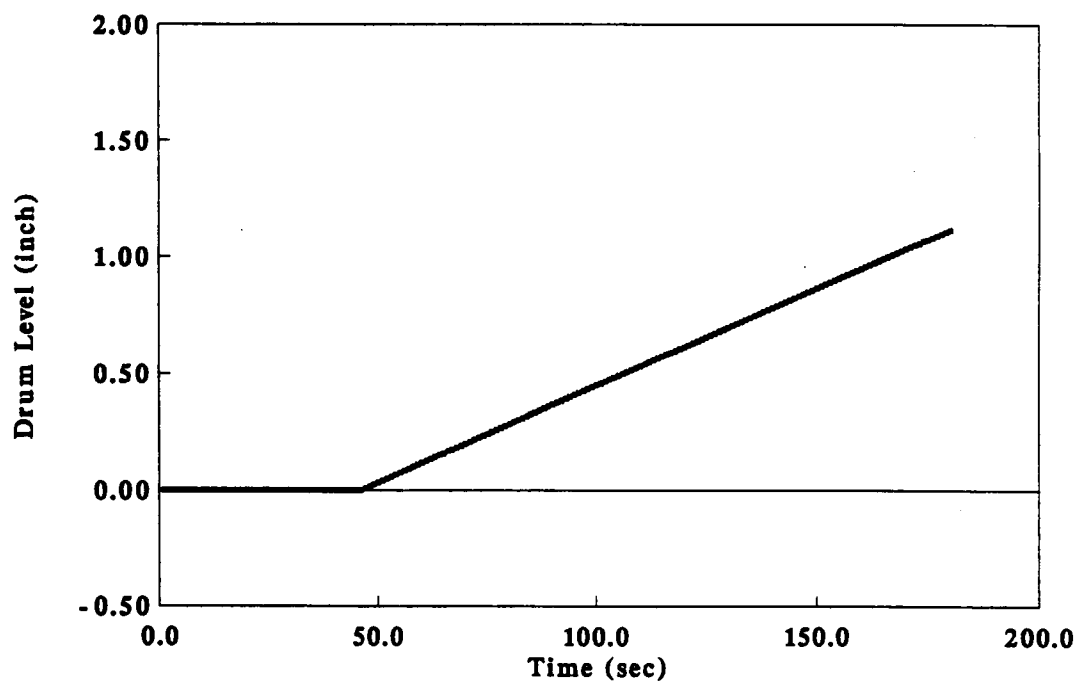
b

Figure 2-12. Continued



c

Figure 2-12. Continued



d

Figure 2-12. Continued

CHAPTER 3

CONTROLLER DESIGN AND SIMULATION

With a reasonably accurate model in place, a solution to the drum level control problem will be developed. To achieve the ultimate objectives of this paper, several control designs will be explored. Basically, a proportional controller will be used as a basis and must be designed initially. Feed-forward techniques and Smith-Predictor Model techniques will conclude the system design.

Before designing a proportional controller for the level process, the Root Locus of the Inverse Response Model will be observed. Using Equation (2-20), the s-plane plot portrays two poles and one zero. The poles are located at $-1/8$ and 0 , indicated by an X, and the zero is located in the RHP at $+1/35$, indicated by an O. When the loop is closed around the plant, the system becomes a second order system with a characteristic equation as in (3-1),

$$s^2 + (1/8 - K)s + K/35 \quad (3-1).$$

From the Routh's stability criterion, the system is closed loop stable for $K < 1/8$. The Root Locus Plot is shown in Figure 3-1 and 3-2.

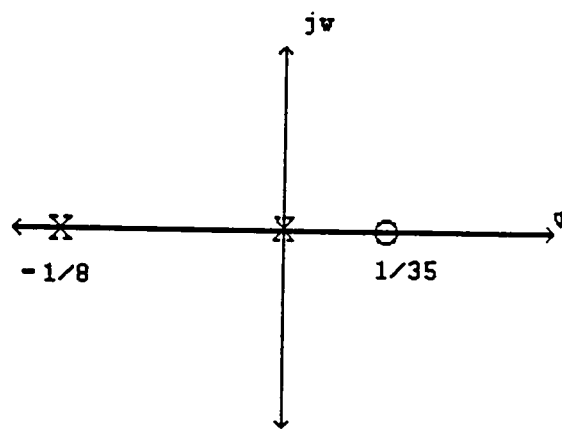


Figure 3-1. Pole and Zero Placement for Plant

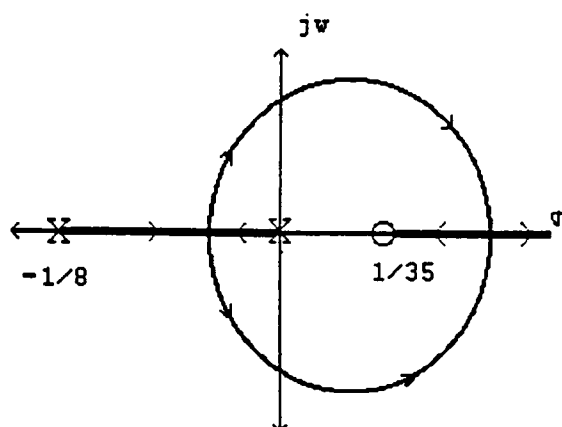


Figure 3-2. Root Locus Plot for Plant

PROPORTIONAL DESIGN

In designing a proportional controller, the control criterion chosen will be a damping ratio of 0.5. This value will allow for a reasonably fast responding stable loop. Using Root Locus design techniques, the control boundary is drawn as shown in Figure 3-3. The intersection of the damping line and the root locus curve gives the closed-loop poles, $s = -0.0248 \pm j0.042$. Using these poles, the value of gain, K , can be calculated from

$$K = 1/G(s) \quad (3-2),$$

which gives

$$K = 0.0307 \quad (3-3).$$

Since $K = K_c K_v K_p$, and $K_p = 0.0008$, then the gain, $K_c K_v$, is 38.4. A popular compensation design technique typically applied to PID compensated loops is the Ziegler Nichols tuning technique. This design approach results in a loop gain of 0.0143 based on an ultimate period of 1/35. This yields a $K_c K_v$ gain of 17.85. Figure 3-4 shows the block diagram for the proportional system. A simulation of this controller design is shown in Figure 3-5.

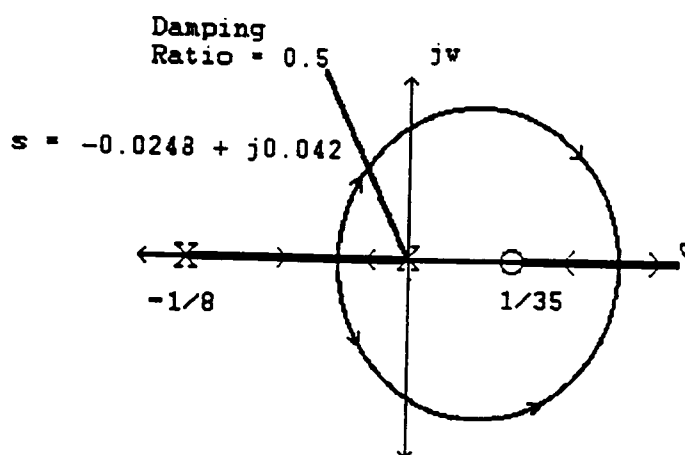


Figure 3-3. Root Locus Plot with Control Boundary

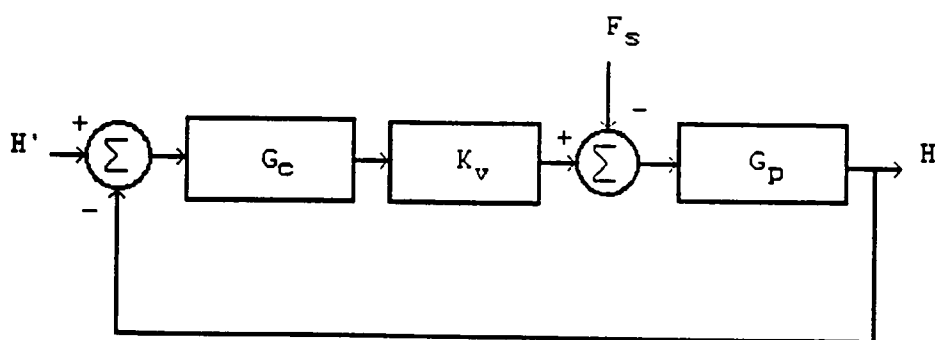
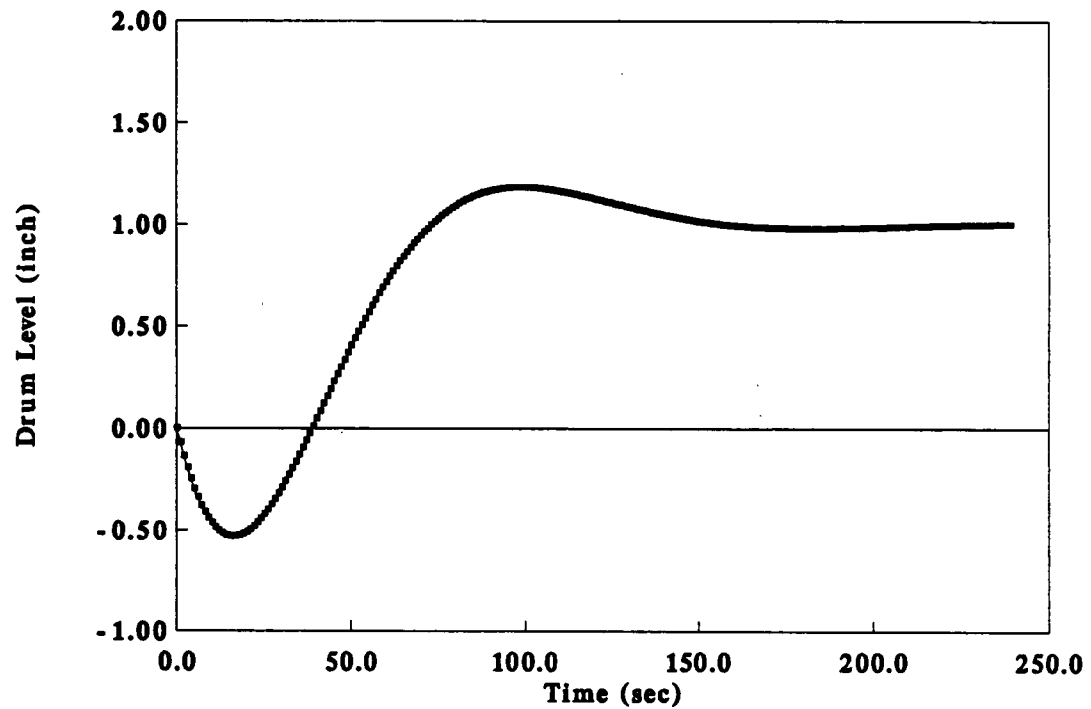
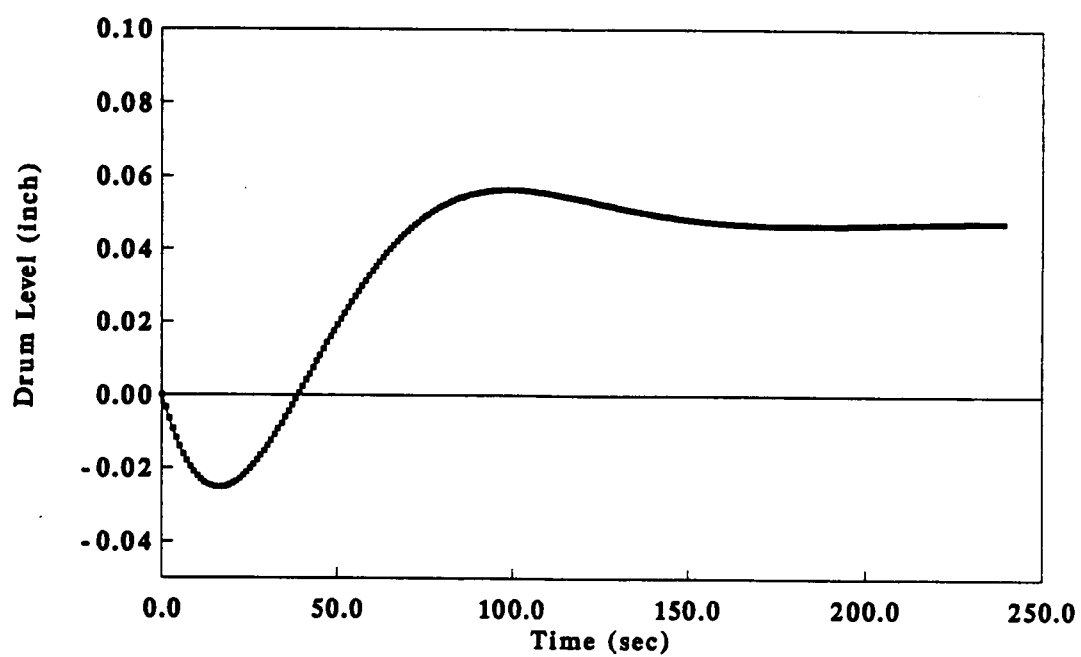


Figure 3-4. Block Diagram of a Proportional Control System



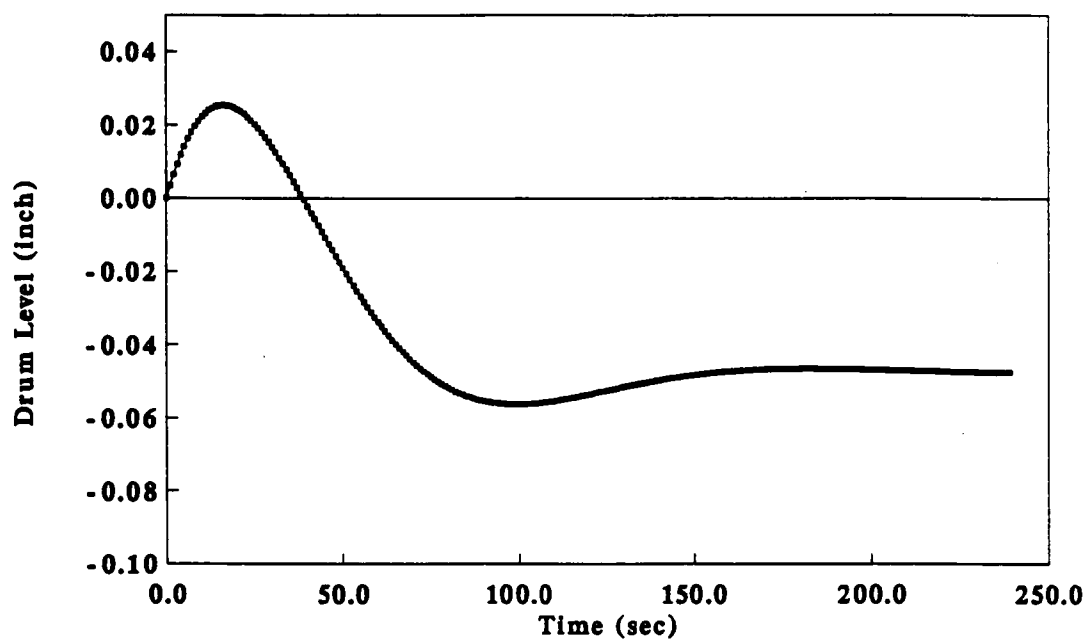
a

Figure 3-5. Closed Loop Step Response of the Proportional Level Control System due to (a) a Setpoint Change, (b) a Negative Load Disturbance, and (c) a Positive Load Disturbance



b

Figure 3-5. Continued



c

Figure 3-5. Continued

Figure 3-5 indicates that the proportional design is reasonably responsive for a simple controller; however, the steam process presented previously cannot tolerate the inverse action that is still present in the proportional design. Essentially, the only improvement is an increase in the speed of response. A more productive controller for the level process would include a method to counteract the inverse response action. A popular counteracting control feature is the use of feedforward techniques.

FEEDFORWARD DESIGN

Since steam flow is viewed as a disturbance to the drum level process, this variable is suitable to feedforward into the control loop. From a physical point of view, feeding the steam flow into the loop offers the proper control action against the "Shrink and Swell" phase of the level. As the steam flow increases, the level swells, which would tend to cause a decrease in the level controller output. This initial tendency is cancelled as the feedforward signal is added to the controller output, which is an increasing signal. For a decrease in steam flow, the effect is similar but opposite.

From a mathematical point of view, the feedforward requirements are clearly displayed. From the mass balance equation 2-1, the steam flow disturbance signal is subtracted from the water flow. If this signal is added to the control

loop and the signal's dynamic effects are emulated, the disturbance effect would clearly be cancelled.

The feedforward controller is designed to cancel all dynamic and gain parameters introduced by the disturbance. A block diagram of the feedforward concept is illustrated in Figure 3-6. This block diagram can be represented mathematically as

$$H = [G_c K_v G_p H' - (1 - K_v/G_{ff}) G_p F_s] / [1 + G_c K_v G_p] \quad (3-4).$$

From this analysis, the only means for the disturbance signal to be cancelled is if the $(1 - K_v/G_{ff})$ term is equal to zero. To satisfy this condition, the G_{ff} term must be $(1/K_v)$.

Strictly from a mathematical and computer simulated viewpoint, the disturbance can be ignored as the feedforward design will result in complete cancellation. However, since K_v is typically an approximated value, there will be a minimal dynamic impact due to physical modelling error. Assuming that the ratio of K_v to G_{ff} is in error of 5 %, then the disturbance term of (3-4) can be expressed as

$$-[0.05 G_p] / [1 + G_c K_v G_p] F_s \quad (3-5).$$

A step response illustrating the impact of the disturbance, assuming the error suggested above, is shown in Figure 3-7. Basically, the inverse response characteristic is present,

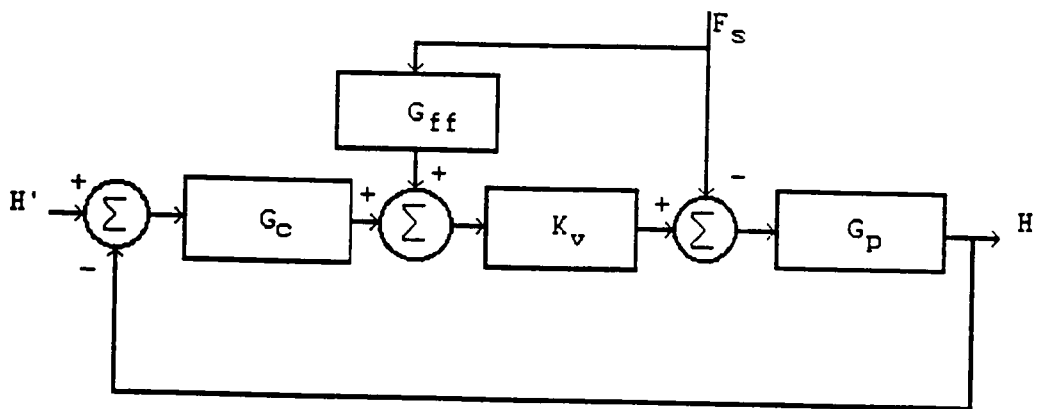
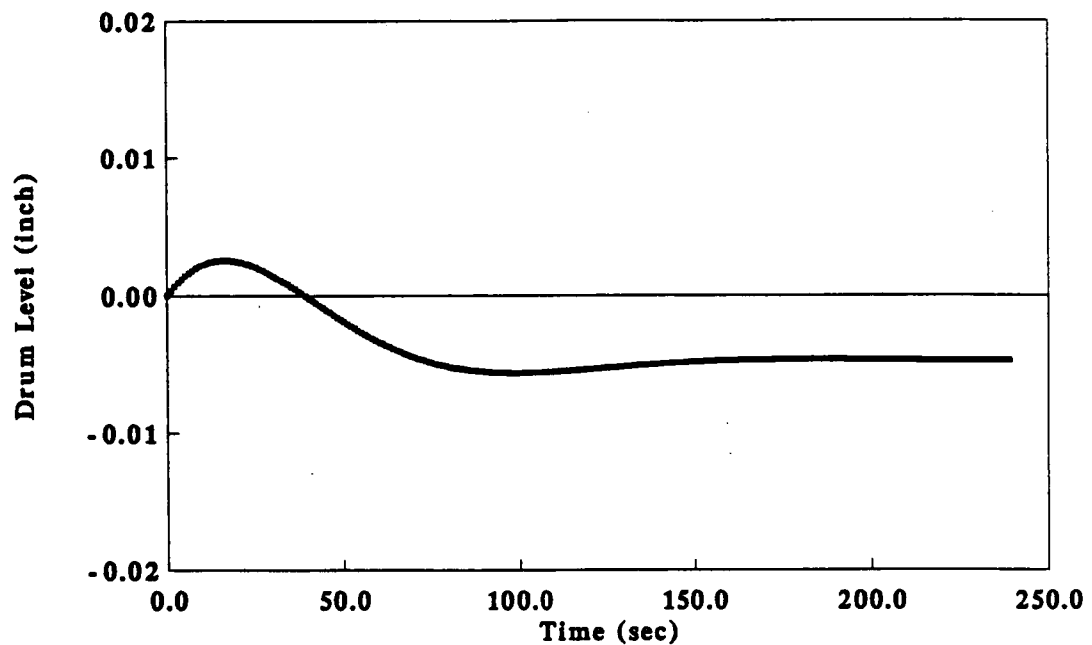
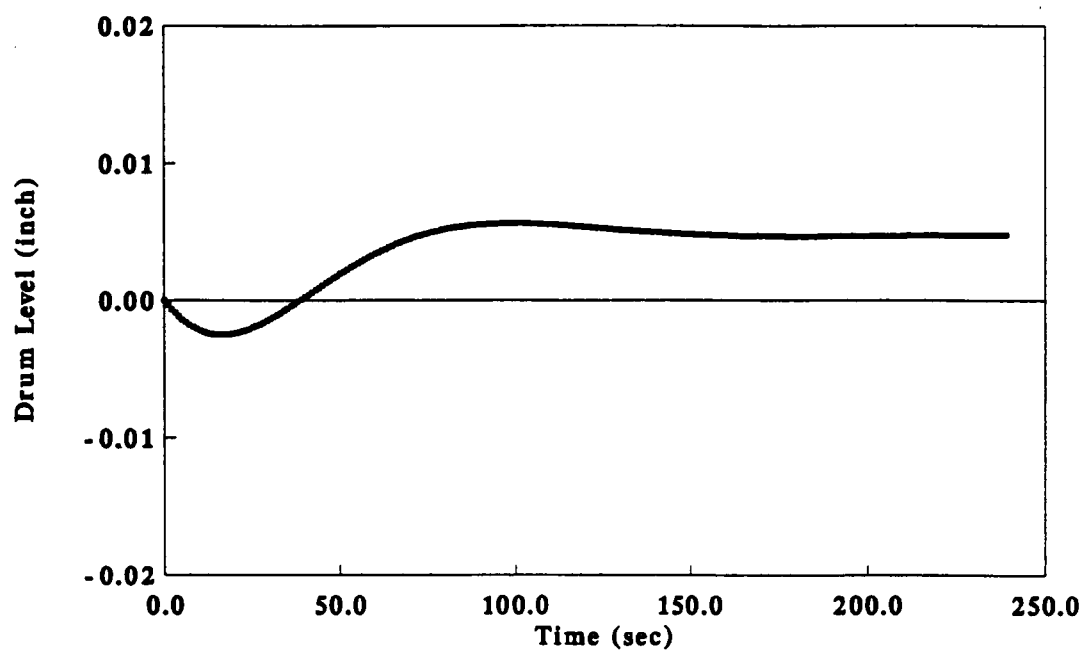


Figure 3-6. Block Diagram of a Proportional Level Control System with Feedforward



a

Figure 3-7. Closed Loop Step Response of the Feedforward Level Control System to (a) a Negative Load Disturbance Change with an Imperfect Compensator and (b) a Positive Load Disturbance Change with an Imperfect Compensator



b

Figure 3-7. Continued

however, it has been reduced by a factor of 95%. The feed-forward design would clearly be an excellent choice for eliminating the inverse response effect due to the steam flow disturbance.

SMITH - PREDICTOR DESIGN

As a final control design, the performance of another control design will be explored, namely, a Smith-Predictor Model. Basically, this design model was developed to minimize the effect of the dreadful deadtime process element. A generalized block diagram for the Smith-Predictor Model is illustrated in Figure 3-8.

The model adopted the title "Predictor," as the system predicts and compensates for the deadtime of the system. The deadtime compensation (DTC) block typically assumes the form of

$$G_{DTC} = (1 - G_{p2})K_v G_{p1} \quad (3-6),$$

where $G_{p2} = e^{-T_D s}$. Applying the proper block diagram reduction techniques, Figure 3-8 can be represented by the block diagram of Figure 3-9. Mathematically, the closed loop transfer function can be written as

$$H/H' = G_c K_v G_{p1} G_{p2} / [1 + G_c K_v G_{p1}] \quad (3-7).$$

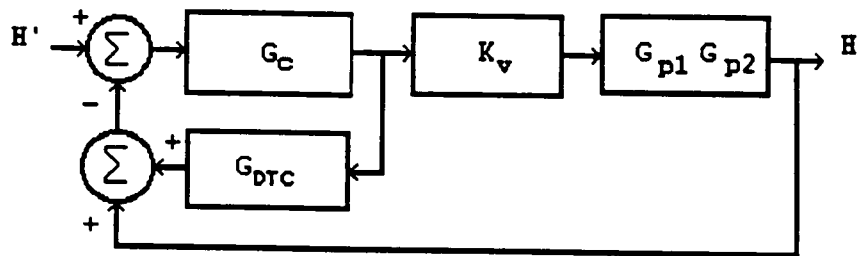


Figure 3-8. Smith-Predictor Model Block Diagram with Deadtime Compensation

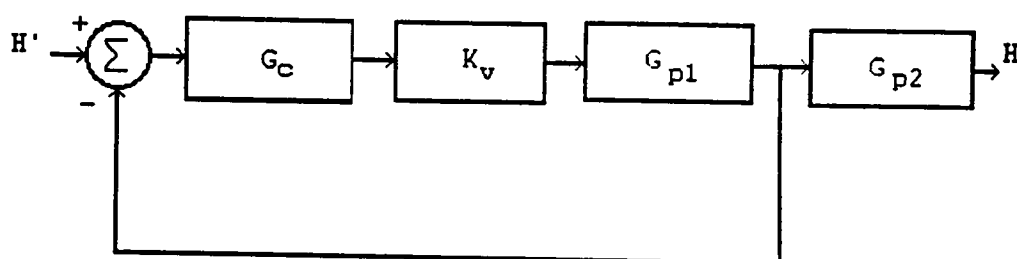


Figure 3-9. Reduced Smith-Predictor Model with Deadtime Compensation

Although the Smith-Predictor Compensation technique is commonly applied to a process with deadtime, the G_{DTC} block can also be designed to compensate for an inverse response (IRC) action. (Stephanopoulos, 390) The block diagram for this model is shown in Figure 3-10. The G_{IRC} block for this model assumes the form of

$$G_{IRC} = (1 - G_{p2})K_v G_{p1} \quad (3-8),$$

where $G_{p2} = [1 - T_2s]/[1 + T_1s]$. Applying block reduction techniques, Figure 3-10 can be simplified to that shown in Figure 3-11. Mathematically, the closed loop transfer function can be written as

$$H/H' = [G_c K_v G_{p1} G_{p2}]/[1 + G_c K_v G_{p1}] \quad (3-9).$$

The analysis results for both the DTC and IRC models, equations (3-7) and (3-9), respectively, indicate that the deadtime or positive zero is moved outside the feedback path. With the delay element removed from the control loop, a dynamic compensator, G_c , can be designed. This compensator is unconcerned with the delay element; The design is based on the remaining elements, $K_v G_{p1}$. Referring to the reduced IRC model of Figure 3-11, the valve gain, K_v , and plant, G_{p1} , can be written as

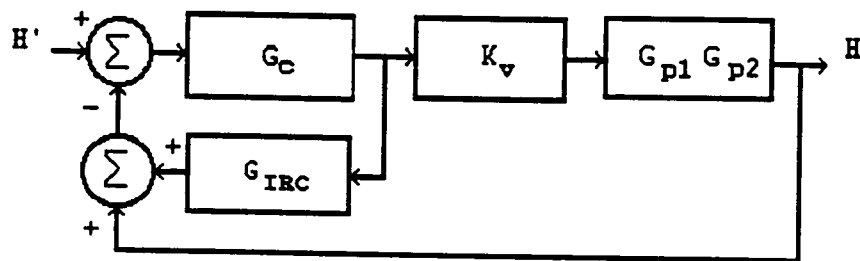


Figure 3-10. Smith-Predictor Model Block Diagram with Inverse Response Compensation

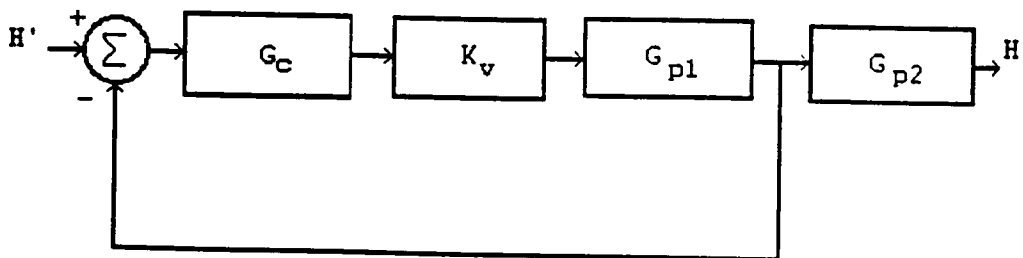


Figure 3-11. Reduced Smith-Predictor with Inverse Response Compensation

$$G_{p1} = K/s \quad (3-10),$$

where $K = K_v K_{p1}$. The root locus of this process lies on the $j\omega$ axis. Since this system is first order, the system is closed loop stable for all values of gain. Using a proportional controller, the closed loop system will yield a first order response, where the speed of response is determined by the value of controller gain.

To achieve a reasonably fast response, a high value of controller gain, K_c , would be chosen. However, in practice, a true first order system does not exist due to transmitter and control valve dynamics. Therefore, a more practical gain value of 10.0 will be used. A computer simulation plot for a step change in the input without the delay is shown in Figure 3-12. Figure 3-13 illustrates a plot of the same feedback loop with the IR delay, G_{p2} , cascaded.

Although the models referenced above compensate for setpoint changes, the impact on the disturbance source should be reviewed as the compensation will be different. Reinstating the original inverse response model from Figure 2-7, the DTC or IRC model is shown in Figure 3-14. An analysis of the block diagram in Figure 3-14 results in

$$H = \frac{\{G_c K_v G_{p1} G_{p2} H' - [G_{p1} G_{p2}] [G_c G_{IRC} + 1] F_s\}}{[1 + G_c K_v G_{p1}]} \quad (3-11).$$

This equation yields a block diagram as shown in Figure 3-15.

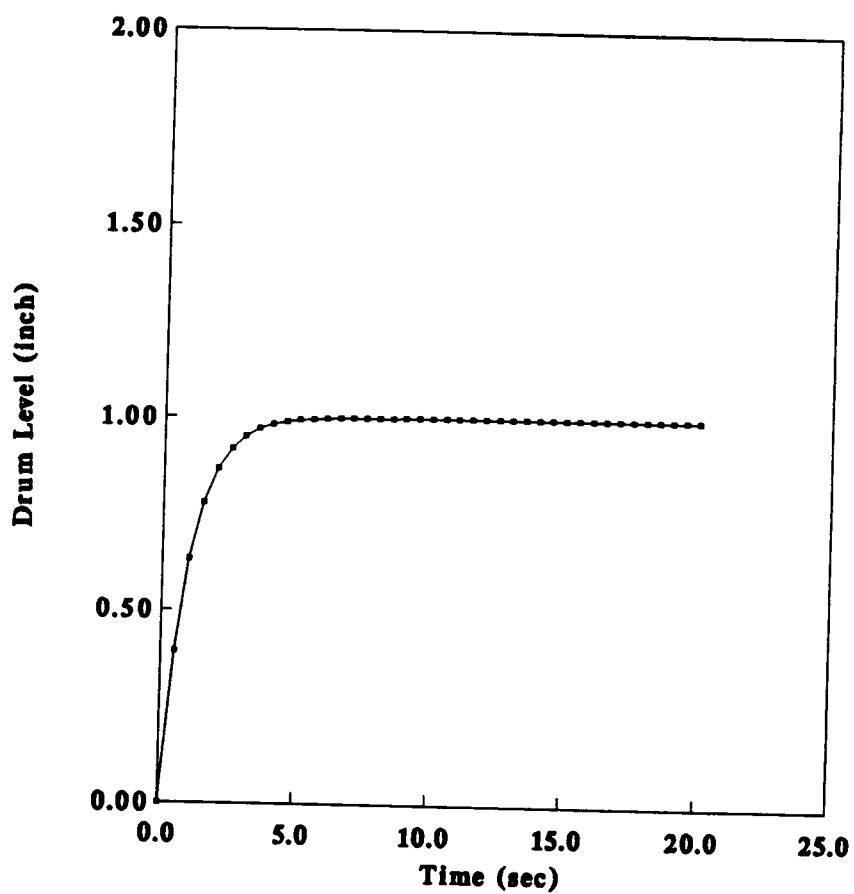


Figure 3-12. Closed Loop Step Response for the Compensated Integration Plant

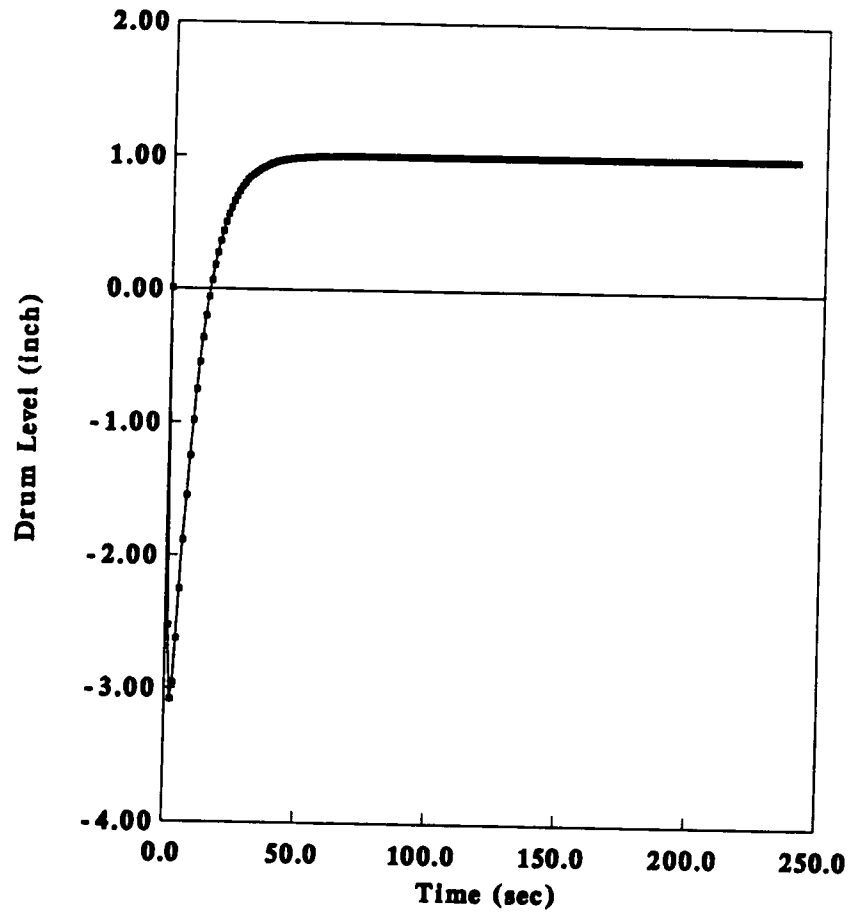


Figure 3-13. Closed Loop Step Response for the Compensated Integration Plant and Inverse Response Delay

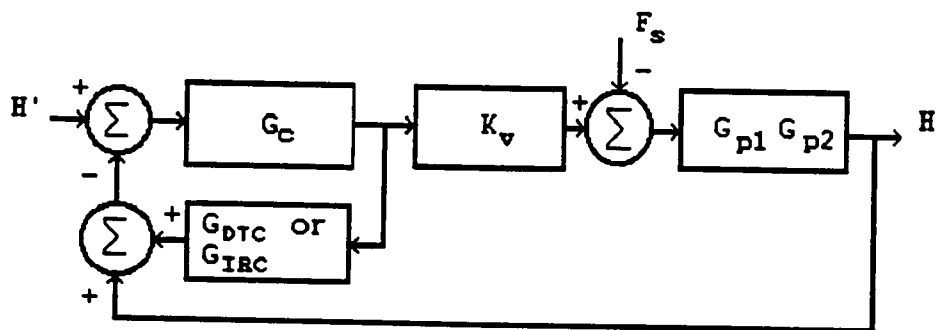


Figure 3-14. Smith-Predictor Model with Plant Disturbance Block Diagram

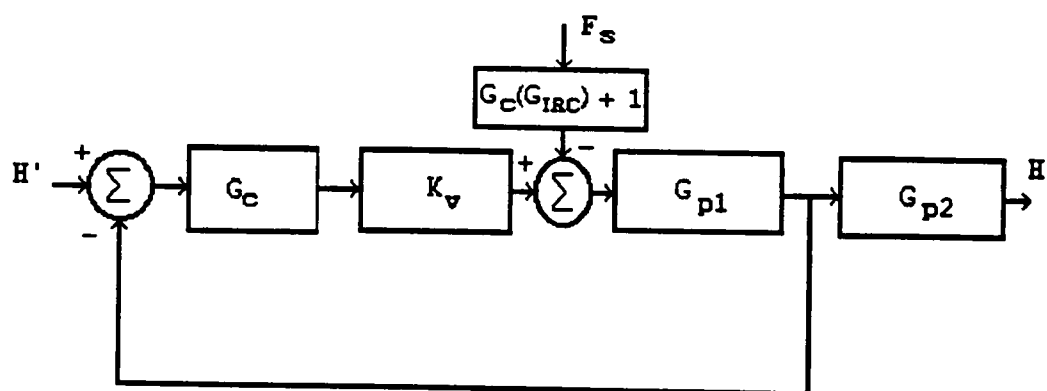


Figure 3-15. Reduced Smith-Predictor Model with Plant Disturbance

From equation (3-11) and Figure 3-15, the disturbance signal passes through a dynamic alteration block before it enters the control loop. Although the Smith-Predictor model moved the delay outside the control loop, the disturbance signal continues to disrupt the control loop performance. Once again, a solution to eliminate the disturbance impact is to apply the traditional feedforward technique.

In order to compensate for the disturbance signal, the compensation dynamics must be computed. Typically, the dynamics are computed with the disturbance signal summed with the controller output; however, since DTC or IRC is used, the dynamics will be calculated for two cases, the feedforward sum junction after the IRC take-off and the feedforward sum junction before the IRC take-off.

CASE I - Feedforward Sum After the IRC Take-off

Figure 3-16 illustrates the IRC-FF model for Case I. An analysis of this block diagram reduces to

$$H = \frac{\{G_c K_v G_{p1} G_{p2}\}}{[G_c G_{IRC} + 1][G_{ff} K_v - 1][1 + G_c K_v G_{p1}]} H' - \frac{\{G_{p1} G_{p2}\}}{[1 + G_c K_v G_{p1}]} F_s \quad (3-12).$$

Since feedforward techniques eliminate the disturbance, then

$$[G_{ff} K_v - 1] = 0 \quad (3-13)$$

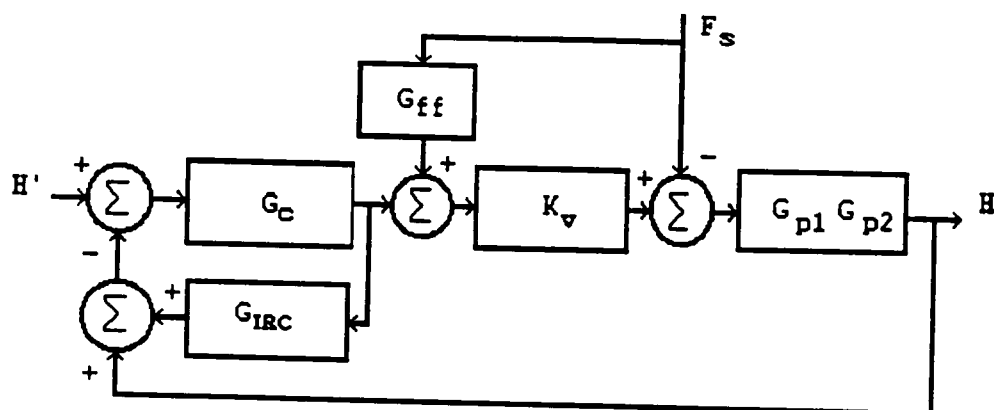


Figure 3-16. Inverse Response - Feedforward Compensation Model Block Diagram with Inverse Response Compensator Input before the Feedforward Sum

or

$$G_{ff} = 1/K_v \quad (3-14).$$

This result is identical to the feedforward compensator design developed previously. Now, the model is reduced to

$$H = \{[G_c K_v G_{p1} G_{p2}]/[1 + G_c K_v G_{p1}]\} H' \quad (3-15).$$

CASE II - Feedforward Sum Before the IRC Take-off

Figure 3-17 illustrates the IRC-FF model for Case II. An analysis of this block diagram reduces to

$$H = \{[G_c K_v G_{p1} G_{p2}]/[1 + G_c K_v G_{p1}]\} H' - \{[G_{p1} G_{p2}][G_{ff} K_v - 1 - G_c \cdot G_{IRC}]/[1 + G_c K_v G_{p1}]\} F_s \quad (3-16).$$

As in Case I, the following equation must be satisfied for total cancellation of the disturbance,

$$[G_{ff} K_v - 1 - G_c G_{IRC}] = 0 \quad (3-17a)$$

$$G_{ff} = [1 + G_c G_{IRC}]/K_v \quad (3-17b).$$

Using this value of G_{ff} reduces equation (3-16) to the equivalent of equation (3-15).

The three controller designs developed in this chapter have varying degrees of effect on the steam drum level

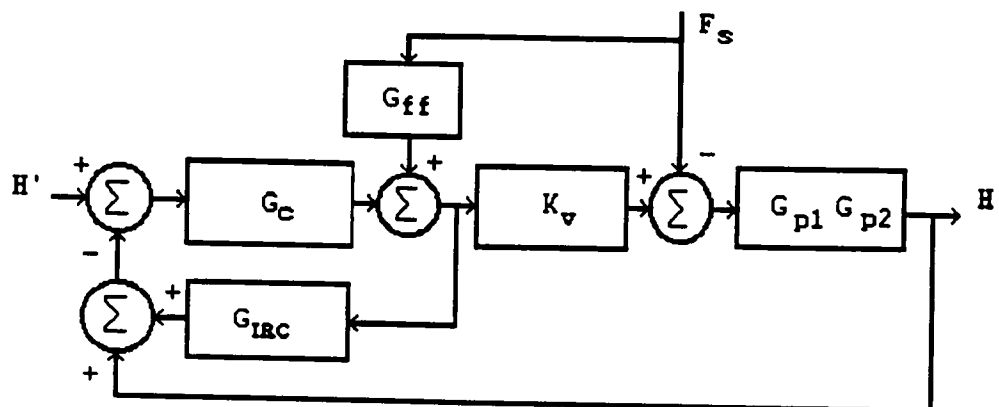


Figure 3-17. Inverse Response - Feedforward Compensation Model Block Diagram with Inverse Response Compensator Input after the Feedforward Sum

response. The proportional design is the most inexpensive, but it is the poorest performer. This design should be used only in a steady load application. The feedforward design offered improvement and is suitable for a non-steady load. This design is basically a load disturbance terminator. And finally, the DTC design offers an improvement similar to the feedforward technique, but it should be used only if an accurate model is available.

CHAPTER 4

SUMMARY, CONCLUSIONS, AND IMPLICATIONS

Throughout this paper, a problem has been addressed to offer enlightenment in choosing a control scheme to manage the water inventory of an 80,000 pph boiler when subjected to a cyclic load disturbance. Too often, the opportunity to resolve problems with available technology and ideas is overlooked due to a misunderstanding of a system's dynamic behavior. In this document, the basic problem was presented in chapter one. In an effort to recognize the behavior of the system, a mathematical model which accurately defined the system was developed in chapter two. Finally, three feasible solutions were studied in chapter three to offer a choice for field application, namely, a proportional feedback system, a proportional feedback-feedforward system, and a Smith-Predictor Model system.

Of the three controller designs that were presented, the proportional feedback-feedforward scheme became the solution of choice to manage the plant of Figure 1-2 and 1-3. This system was chosen based on performance and controller availability.

The proportional feedback-feedforward scheme reduced the impact of the load disturbance by anticipating the level deviation during a load upset. The proportional feedback system took a much longer time to respond and had a greater

inversion amplitude. The Smith-Predictor Model, although an excellent performer under ideal conditions, did not allow for a large plant model margin of error. As chapter two illustrated, there were four models developed to represent the drum level system. Each model had a slightly different gain and time constant. In order to achieve stability for all possible plant models, the proportional feedback-feedforward scheme proved to be more robust.

From an availability point of view, the proportional feedback and the proportional feedback-feedforward schemes are simple to implement with the controllers that are available today. The proportional feedback-feedforward controller is available in most single loop digital controllers. On the other hand, the Smith-Predictor Model scheme typically requires additional hardware for modelling the inverse effect. Therefore, the proportional feedback-feedforward controller is a more practical investment. To further reinforce the use of the proportional feedback-feedforward scheme, the installation requires a flowmeter to measure steam flow. This measurement is typically available at the panel for steam flow monitoring; therefore, the cost of using the feedback-feedforward scheme is no greater than a traditional feedback system.

A single loop digital controller was installed to accomodate a feedback-feedforward arrangement. Since the controller is digital, the chapter three s-plane design would not be accurate, but would offer ballpark starting values for

field tuning. Avoiding the details of the z-plane controller design, the controller performance can be recognized in Figure 4-1.

From Figure 4-1, the drum level is maintained within a small control band of ± 1 inch. This is a tremendous improvement from the ± 4 inch band suggested in chapter one using a thermostatic regulator. The water flow follows the steam flow during the entire dynamic load swing.

As a performance measure, Figure 4-2 illustrates the water and steam flow of a boiler with thermostatic regulator level control. For a steam load as that applied in Figure 4-1, the water flow lags behind the load disturbance. This shows the poor control response that is inherent with the thermostatic regulator when subjected to a cyclic load.

As demonstrated in this document, the replacement of the thermostatic regulator philosophy with feedback-feedforward technology has proven to be an excellent choice for the plant described in chapter one. The feedback-feedforward scheme has offered quicker response in maintaining a constant drum level constant. The reduction in level swings not only improves the life of the boiler but also improves the efficiency of the evaporation process. To understand the level system's dynamic behavior, a mathematical model was developed which also proved useful for controller design. In chapter three, several possible control schemes were presented to establish the best

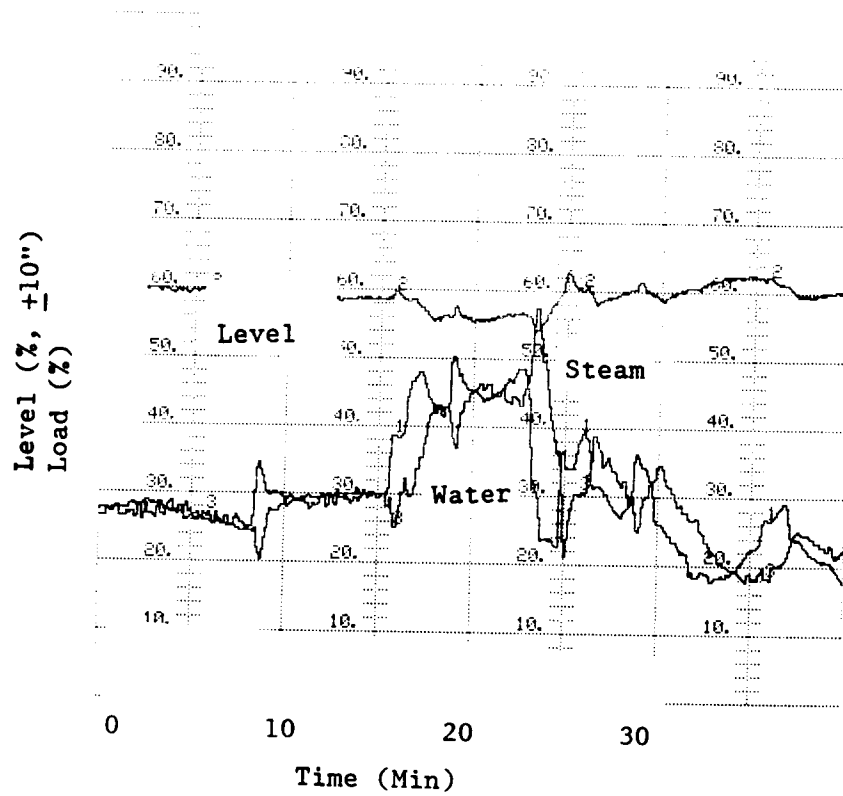


Figure 4-1. Feedback-Feedforward System Field Data

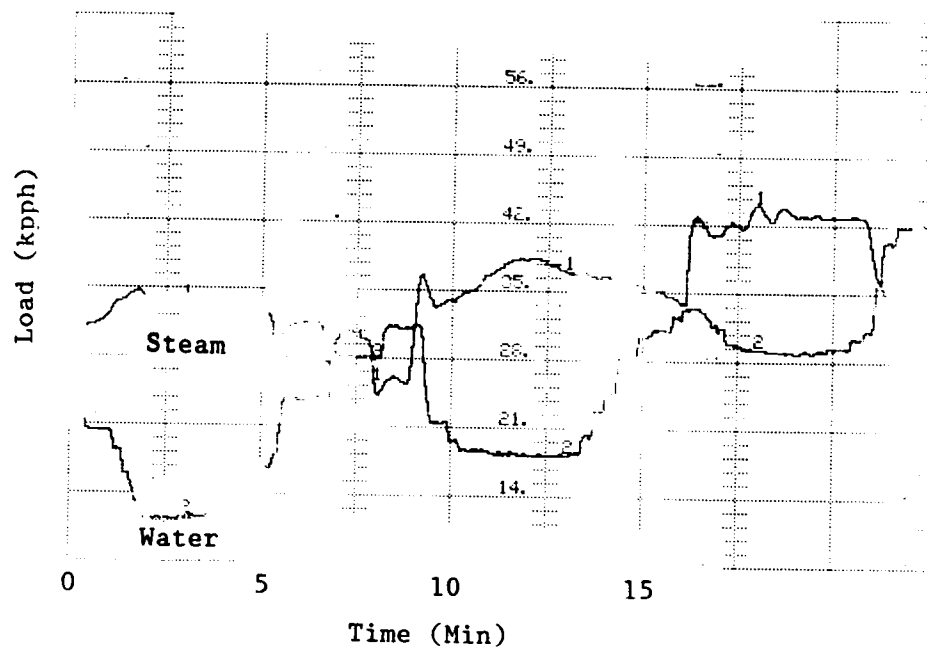


Figure 4-2. Thermostatic Regulator System Field Data

control approach for the type of load disturbance anticipated. The choice controller was applied in the field to demonstrate the performance of the new system. Although the thermostatic regulator has been an industrial boiler workhorse for years, the control designer must take care in selecting a system that is appropriate to the application.

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In the summer of 1988, Kenneth accepted a graduate teaching/research assistantship at the University of Tennessee, Knoxville and began study toward a Master's degree. In January 1989, he accepted a position as Project Engineer with J.M. Huber Corporation in Etowah, Tennessee. In September 1989, he re-enrolled in the University to complete his graduate work. In July 1992, he accepted a position as Instrument Engineer with American Cyanamid in New Orleans, Louisiana. The author received a Master of Science degree with a major in Electrical Engineering in August 1992.

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