To the Graduate Council:

I am submitting herewith a thesis written by Karl Philip Zimmerman entitled "Mathematical Modeling and Analysis of a Pressure Control System." 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.

James Hung, Major Professor

We have read this thesis
and recommend its acceptance:

Marshall O. Pace

J. Vaughn Blalock

Accepted for the Council:

Associate Vice Chancellor and
Dean of The Graduate School

Mathematical Modeling and Analysis of a Pressure Control System

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

Karl Philip Zimmerman
May 1997

Dedication

This thesis is dedicated to my mother

Dr. Elena Irish Zimmerman

whose love and guidance have been the cornerstones of my life.

Acknowledgments

I would like to thank the University of Tennessee Graduate School and the College of Electrical Engineering for allowing me to pursue the advanced degree for which this thesis has been written. In particular, I would like to thank Dr. James C. Hung for his support and guidance during the writing of this document. I would also like to thank Lockheed Martin Energy Research Corporation for their generous financial support of this masters program.

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Abstract

In this study a pressure control system with a history of performance problems is examined. A mathematical model for the system is developed using a combination of analytical and experimental data. Model responses to inputs are determined and compared to actual system responses to validate the model. Desired system responses are discussed and optimum proportional-integral-derivative controller parameters are determined using a graphical identification approach. Component span adjustment and other system improvements from a hardware standpoint are also proposed. It is found that, even though the pressure control system is not linear, an acceptable model can be developed using Laplace transform techniques that will adequately simulate the system. This model, when used with the optimal controller tuning parameters, produces some improvements in response to both step and disturbance inputs. Improvements in response are also produced as a result of certain component span changes. Hardware modifications can also improve system performance and changes are recommended for the controller, the valves, and the sense lines.

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List of Abbreviations

ac	alternating current
FOLPD	first-order-lag-plus-delay
ft	feet
gal	gallons
gpm	Gallons per minute
HFIR	High Flux Isotope Reactor
IAE	integral of absolute value of error
in	inches
ITAE	integral of time multiplied by the absolute value of error
lb	pounds
ma	milliamps
min	minutes
MW	megawatts
PB	percent proportional band
PID	proportional-integral-derivative
psi	pounds per square inch
psig	pounds per square inch gauge
sec	seconds

Chapter 1

Introduction

In the study of control system theory, many texts devote a chapter or so to the development of transfer functions. Several methods such as the simultaneous equation method and the block diagram method (Dransfield 1968) are usually presented to develop these input-output relationships. Once this information is discussed, the study of the analysis and design techniques that follow usually involves a transfer function that is given rather than derived. For this reason, not much time is spent learning the process of determining transfer functions, a skill that is important to the control system engineer since the transfer functions of most systems are unknown.

The derivation of system transfer functions can be a frustrating experience as many times the responses of the model developed do not match the actual system responses. Systems such as pressure control systems can cause additional problems because the input-output relationship of the process involves a flow-pressure relationship which is not linear. Normally Laplace transforms cannot be used to solve nonlinear differential equations. If transfer function analysis is desired, then linear approximations for the nonlinear elements must be determined. This makes these types of systems more difficult to model than a linear system whose component transfer functions can be expressed solely by constants .

This study seeks to explore the process of control system modeling. A pressure control system at a nuclear reactor facility will be modeled using linear approximations, where necessary, in order to utilize Laplace transfer function techniques. It has been chosen because of a past history of problems, the opportunity to examine the system, and the desire to develop recommendations that will serve as a basis for making system improvements.

Problem Statement

The Pressure Control System at the High Flux Isotope Reactor (HFIR) monitors the pressure of the demineralized water in the Primary Coolant System and adjusts the letdown valves to maintain the pressure at the desired set point. Although pressure control is fairly stable once equilibrium is reached, system pressurization and other operator actions that disturb this equilibrium often result in undesirable overshoot or undershoot and have caused unnecessary alarms and reactor scrams. Since the proportional-integral-derivative (PID) controller parameters were determined by the manufacturers tuning instructions for the controller rather than by analytical methods, a system analysis may produce PID parameters that can optimize performance. Controller tuning parameters cannot be evaluated, however, unless a system model can be developed.

Study Objectives

The objectives of this study will be:

(1) To determine the desired response characteristics for the control system.

(2) To create the model for the system. The block diagram will be constructed and the transfer functions for all system components will be determined. These will be combined to form the complete system model.

(3) To determine the response of the system by simulation based on the current controller settings.

(4) To determine optimal PID controller settings in an effort to improve system response characteristics.

(5) To simulate the output responses with the new controller settings and compare them with the original settings.

This study is intended to validate the existing controller settings or provide the basis necessary for proposing new controller settings that will enhance system performance.

Chapter 2

Control System Description

High Flux Isotope Reactor

The High Flux Isotope Reactor (HFIR) is a beryllium-reflected light water cooled and moderated aluminum-clad fuel plate flux-trap type reactor which utilizes highly enriched U^{235} fuel (ORNL 1968). The HFIR is located at Oak Ridge National Laboratory about 1 mile south of the main site. The power level is 85 MW and a fuel cycle is about 21 days.

The primary purpose of the HFIR is the production of transuranium isotopes. Gram quantities of elements such as Californium, Palladium, and Iridium are produced annually. Californium is used widely in industry to calibrate neutron instruments, examine fuel rods and aircraft luggage, and perform irradiation testing of sensitive electronics. It is also used in medicine to treat cervical cancer. Palladium is used in the treatment of prostate cancer. Iridium is used in the nondestructive examination of welds, mainly in the oil pipeline industry and in naval shipyards.

In addition to the production of isotopes, the HFIR also contains neutron-scattering and irradiation facilities which attract scientists from around the world. Neutron-scattering is utilized to explore the composition of a wide range of materials. It has been used to advance research in such areas as superconductivity and polymer composition. Irradiation

damage studies benefit from the exceptionally high neutron flux available. A hydraulic rabbit is provided for short term irradiation while irradiation facilities in the beryllium reflector surrounding the core are provided for longer term tests.

Primary Coolant System Overview

The Primary Coolant System circulates approximately 13,800 gallons of demineralized water at about 16,500 gallons per minute (gpm) to remove heat from the reactor core (Figure 2.1). The cooling water route takes it through the reactor core and then down a header where it branches and passes through the primary side of three vertical heat exchangers (a fourth is in standby). The cooled water leaving each heat exchanger enters an associated vertical shaft centrifugal circulation pump. The output flows from each pump are recombined and flow back to the reactor vessel through a strainer. Instrumentation in the system monitors pressure, temperature, and flow in order to maintain these variables within prescribed limits. At the exit of each heat exchanger, letdown headers divert approximately 125 gpm to the Primary Coolant Cleanup System. Here the coolant water is deaerated, filtered, demineralized, and pumped back into the primary coolant loop upstream of the strainer.

The Primary Coolant System is pressurized by multistage centrifugal pressurizer pumps which also function to transfer water from the Cleanup System back to the Primary Coolant System. The pressure of the primary coolant system is controlled by letdown valves located in the heat exchanger letdown headers. These pneumatic valves are

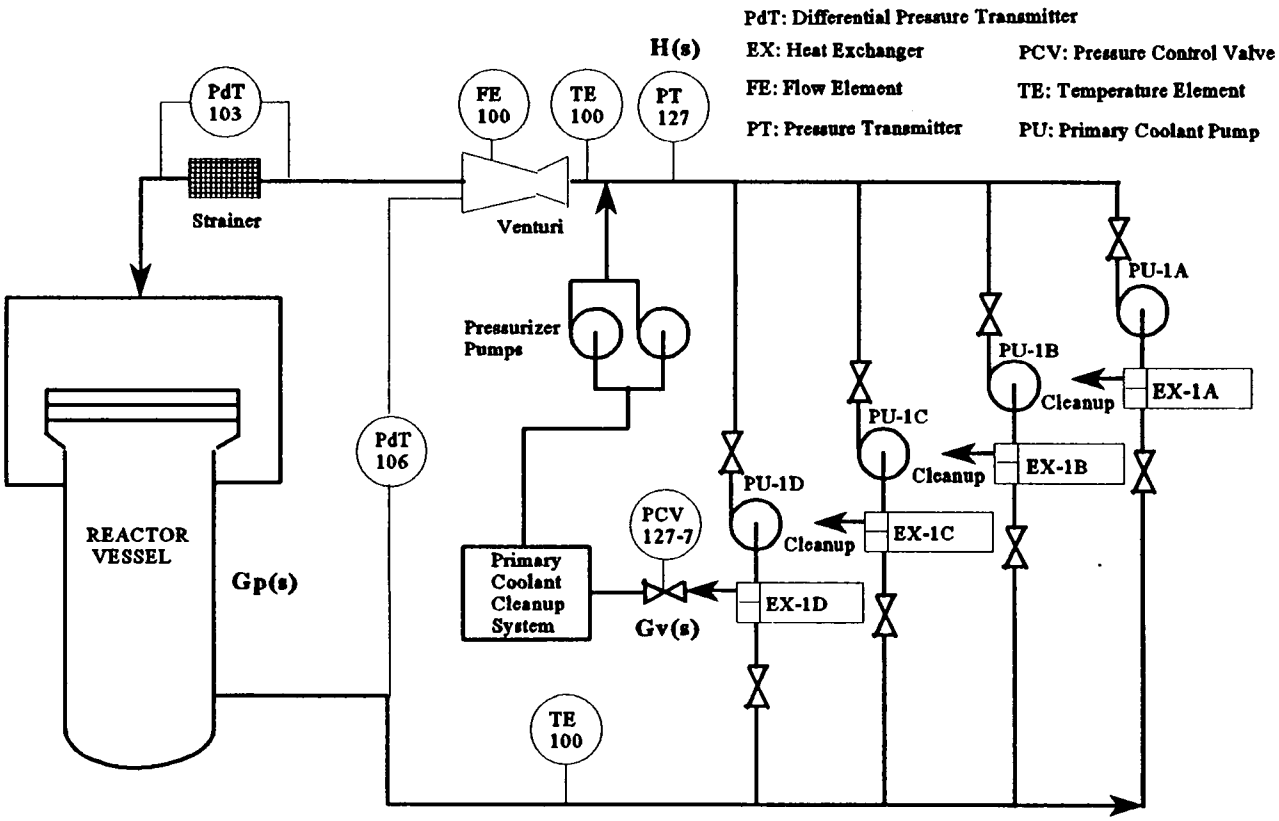


Figure 2.1. Primary Coolant System Diagram

controlled by a proportional-integral-derivative (PID) controller which receives a pressure sensor signal and adjusts the letdown valves to maintain the desired system pressure.

Reactor Vessel

The reactor vessel is an 8 ft. wide carbon steel pressure vessel which houses all the reactor core components including the fuel elements, target assembly, beryllium reflector, control plates, and various core support and assembly components. It is located in an 18 foot wide 36 foot deep pool of demineralized water which provides the biological shield for normal operation and refueling activities. The pressure drop through the vessel is approximately 125 psi under full flow conditions.

Heat Exchangers

The primary heat exchangers transfer heat to the secondary coolant system utilizing a shell and U-tube type design. Each heat exchanger is 34 feet long and mounted vertically in an individually shielded compartment with its associated circulating pump and isolation and letdown valves. Each is designed to remove 34.3 MW of heat at 5000 gpm from the primary to the secondary system, therefore, for 85 MW operation, only three are required to be in service. The primary (tube) side is designed for 1000 psi at 200°F. The secondary (shell) side is designed for 150 psi at 200°F.

Primary Coolant Pumps

The primary coolant pumps provide the motive force for the coolant water circulating

through the reactor core and vessel components. Each vertical shaft centrifugal pump takes its suction from its associated heat exchanger and is located adjacently in the heat exchanger cell. These pumps are driven by 2300 volt ac, 600 HP squirrel cage induction motors coupled to the pump shaft. Directly coupled to each ac motor shaft is an auxiliary 3 HP series-wound dc motor. These motors are powered from a battery system and provide 1300 gpm each under loss of power conditions.

Pressurizer Pumps

The two pressurizer pumps provide makeup from the cleanup system to the primary coolant system. The speed of the pressurizer pumps, therefore, determines the letdown flow rate. They are nine stage horizontal shaft centrifugal pumps that can deliver 300 gpm at 1000 psi. Each pump is driven by a 2300 volt 300 HP squirrel-cage induction motor that is coupled to the pump by a magnetic clutch. A potentiometer in the control room adjusts the slip of the clutch which determines pump speed and therefore makeup flow. Only one pump is in service during reactor operation.

Strainer

An inlet strainer located on the reactor vessel inlet header removes foreign material from the primary coolant before entry into the reactor vessel. A stainless steel mesh with a maximum opening size of 0.036 inch filters out particles large enough to block the 0.050 inch coolant channels in the reactor fuel. Differential pressure across the strainer is monitored in the control room.

Pressure Control System

The Pressure Control System maintains primary coolant system pressure at a level which provides sufficient margin to prevent incipient boiling in the fuel region. Maintaining sufficient pressure is critical if fuel damage and the release of fission products are to be prevented. Normally, pressure is maintained at 468 psi. On decreasing pressure, valves automatically close at 393 psi to isolate the primary coolant system from other systems such as the cleanup system in order to maintain pressure. At 368 psi the reactor automatically scrams.

The pressure control system is shown in Figure 2.2. It consists of a proportional-integral-derivative (PID) controller which receives a transmitted signal proportional to system pressure from a pressure transmitter whose sense line taps the Primary Coolant System upstream of the strainer. The controller compares system pressure to the adjusted set point and adjusts the three in-service letdown valves to maintain the desired pressure. Each letdown valve is also connected to a solenoid valve which closes the letdown valve when its corresponding primary coolant pump is deenergized. Pressure switches also monitor transmitter output and initiate alarms on high and low pressure. The pressurizer pump pressurizes the system and controls the letdown flow rate, however, letdown valve position controls system pressure. The pressure control system is a pneumatic system. All components are supplied with 20 pounds of regulated air and transmit or respond based on a 3-15 psi signal. The signal is transmitted through ⅜ inch copper tubing.

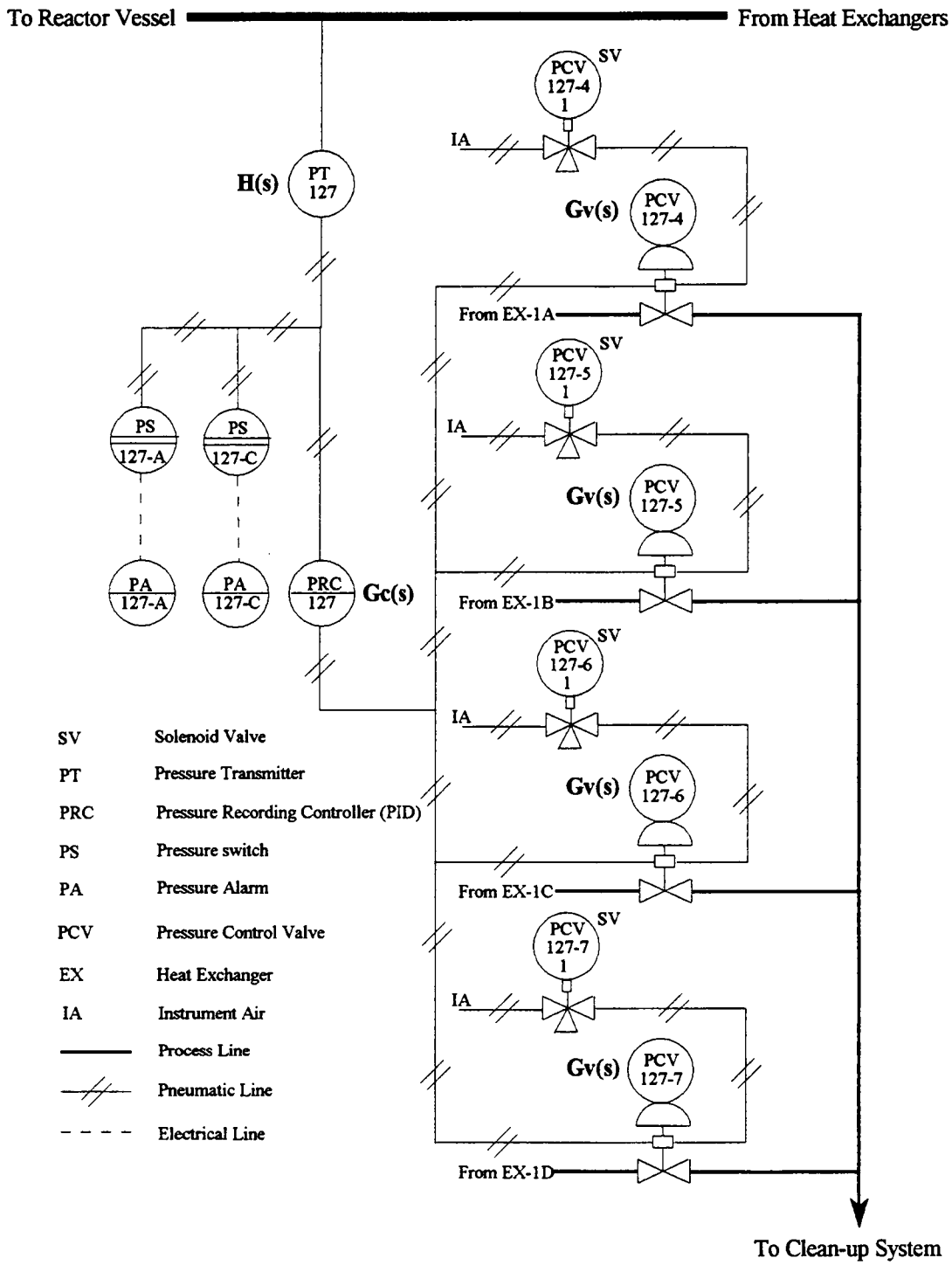


Figure 2.2. Primary Pressure Control System

Pressure Transmitter

The pressure transmitter is a type 11GM made by the Foxboro Company (Foxboro, 1964). A schematic diagram of the transmitter is shown in Figure 2.3. It is a special seal welded, force balance instrument that measures pressure and transmits it as a proportional 3-15 psi pneumatic signal. The pressure is applied to a bellows capsule. The force on the bellows capsule is transmitted through a flexure to the lower end of the force bar. The metal diaphragm is a fulcrum for the force bar. The force is transmitted through the flexure connector to the range rod which pivots on the range adjustment wheel. Any movement of the range rod causes a minute change in the clearance between the flapper and nozzle.

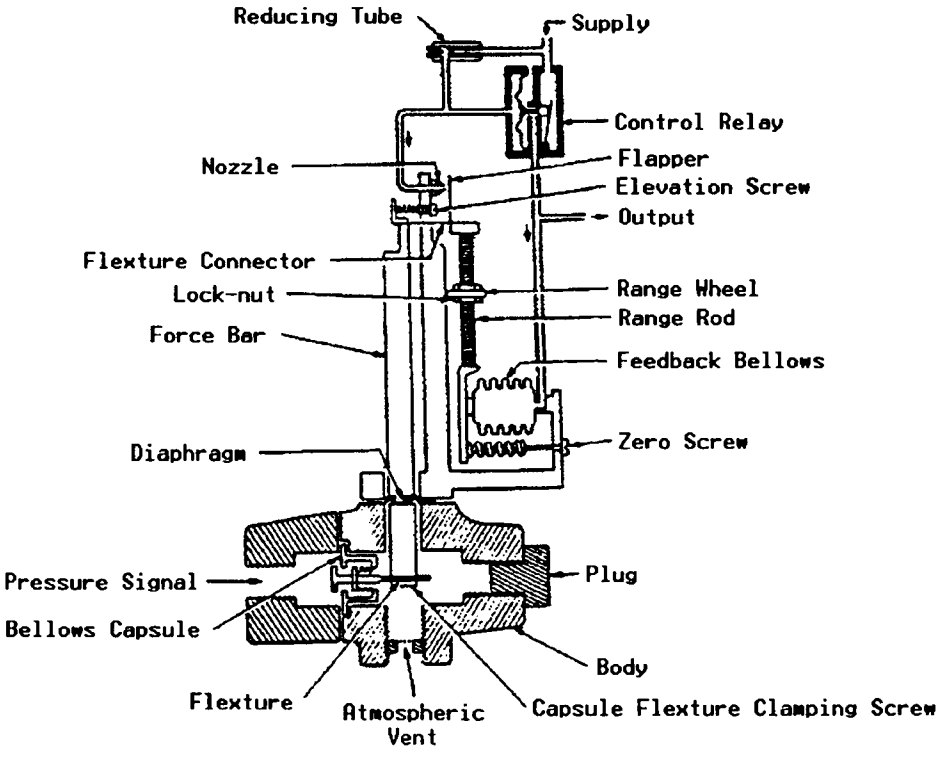


Figure 2.3. Pressure Transmitter, H(s), Internal Construction

This produces a change in the output pressure from the relay to the feedback bellows until the force in the feedback bellows balances the force on the bellows capsule. The output pressure which is established by this force balance is the transmitted signal and is proportional to the pressure applied to the bellows capsule.

Controller

The PID controller is a model 58 also manufactured by The Foxboro Company. It too operates on the force-balance theory. A schematic diagram of the controller is depicted in Figure 2.4. It receives the 3-15 psi pneumatic signal representing system pressure from the pressure transmitter and outputs a 3-15 psi pneumatic signal to the letdown valves for pressure control. An increase in the measurement pressure increases the output which further opens the letdown valves to reduce system pressure until it corresponds to the set point pressure. The floating disc acts as the flapper of a conventional flapper-nozzle system. The resultant of the forces due to the upward pressure of the bellows unit determines the position of the floating disc in relation to the nozzle. Hence, the relay output pressure varies with changes in pressure in any of these bellows (Foxboro 1964).

A change in pressure in the measurement or set bellows moves that side of the floating disc up or down and causes a change in nozzle pressure, which results in an increased or decreased output pressure from the control relay. This variance in output pressure acts to reposition the control valve, thus bringing about a change in the measurement bellows. It is also fed back to the proportional bellows. This continues until a balance of forces is

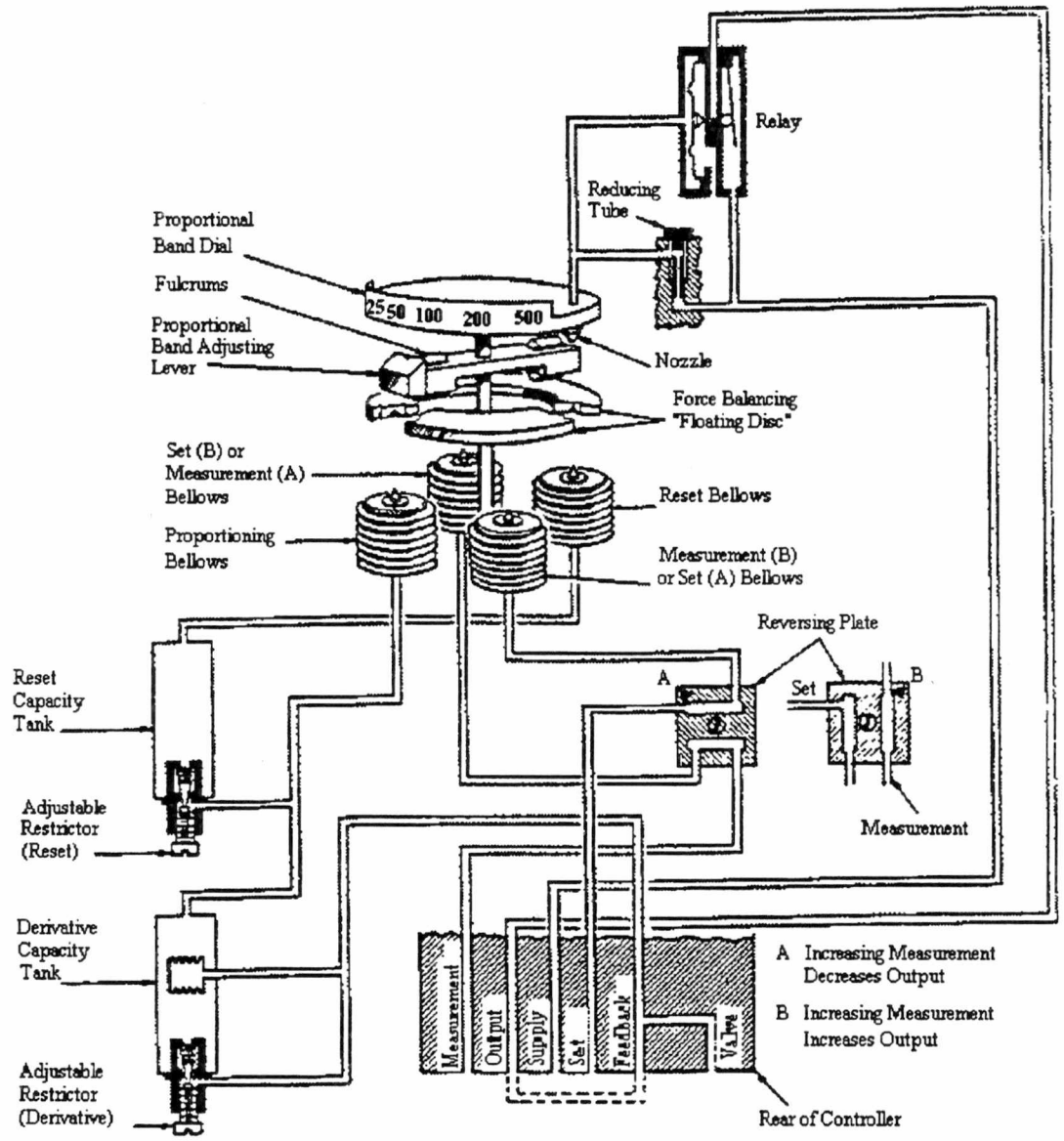


Figure 2.4. Proportional-Integral-Derivative Controller, $G_c(s)$, Internal Construction

restored against the floating disc. Thus, changes in output pressure are proportional to changes in measurement or set pressure.

Changes in the output pressure are fed to the reset bellows as well as to the proportional bellows, but at a rate depending on the setting of the reset restriction or resistance. This resetting action continues until the pressures in the proportioning and reset bellows are equal. However, this action occurs at an ever decreasing rate as the final balance point is reached, at which time the measurement and set pressures are equal. Thus, reset action is dependent on the deviation of the measurement from the control set point and the setting of the reset restriction.

A derivative effect is obtained by interposing a derivative resistance in the feedback line between the output and the proportional bellows. Thus, air flowing to or from the proportional bellows due to changes in the measurement or set pressure causes a pressure drop to occur across the derivative restriction. Pressure in the output line is greater or less than that in the proportioning bellows by the amount of this drop. Hence, the valve position is determined by the combined proportional and reset effect plus or minus the derivative effect. Any valve position will thus be reached more quickly when derivative action exists.

Letdown Valves

At the heat exchanger exits, a nominal 120 gpm bypass flow is routed to the primary

cleanup system through the letdown valves. Each letdown line has an associated letdown valve for regulation and two block valves for isolation. The letdown valves are ITT Hammel-Dahl valves with ITT Conoflow positioners. They are 1 inch unbalanced globe valves made of 316 stainless steel with a 600 pound body pressure rating. They use a special "flash flow" trim to combat cavitation. They have a stroke time of about 10 seconds. The pressure drop across these valves is about 344 psi.

Block Diagram

Figure 2.5 shows an initial block diagram of the pressure control system.

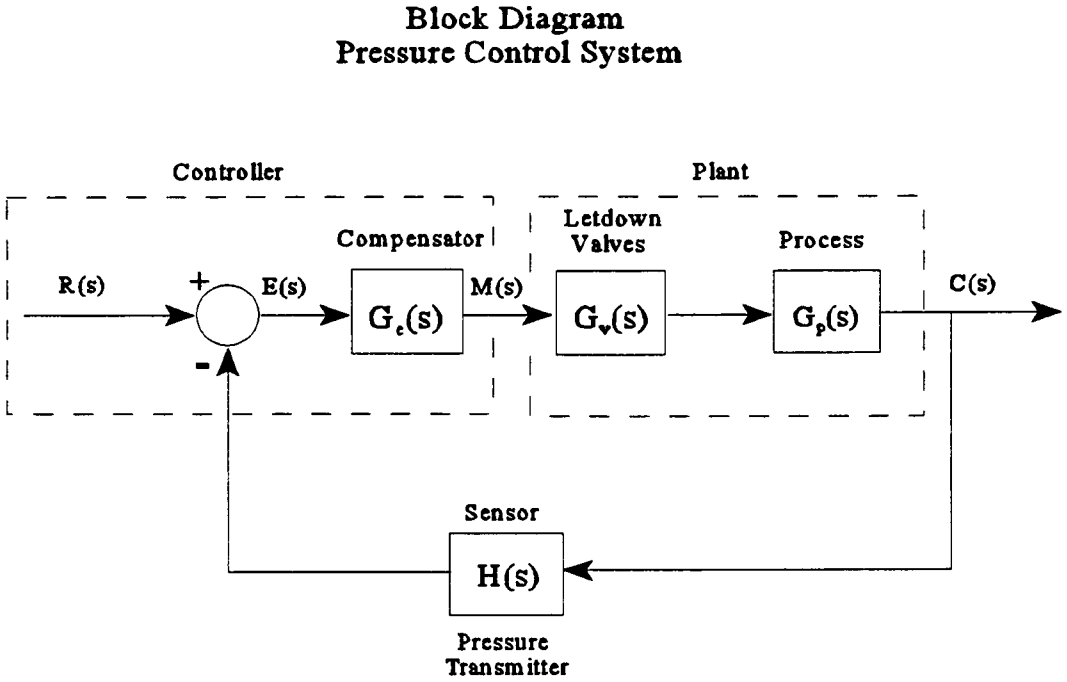


Figure 2.5. Block Diagram of Pressure Control System

Chapter 3

The Mathematical Model

All physical systems are inherently nonlinear and no mathematical model of a physical system is exact (Phillips, 1991), therefore, the goal of mathematical modeling is to develop a model that most closely matches the system while minimizing model complexity. Many systems and system components are linear over a certain range of operation and in this range can be represented as linear. If the system is linear and time-invariant, a transfer function representation may be used. If the range of interest is larger than the linear range or if no linear range exists then transfer functions may still be used, however, a linearization technique of some kind must be employed. The mathematical model for the pressure control system will be developed using transfer functions. The use of transfer functions implies that all the components of this system can be described by linear time-invariant differential equations. Since this is probably not the case for the pressure control system, linear approximations will be determined for the nonlinear components. This will allow these components to be represented by transfer functions and will allow Laplace transform techniques to be used.

In order to develop the mathematical model for the pressure control system, the transfer function for each component must be determined. This is accomplished analytically by developing simultaneous equations utilizing basic laws associated with mechanical,

electrical and fluid pressure dynamics such as the laws of Faraday, Ohm and Newton. These equations are combined in such a way as to eliminate all variables but the input and output variables in question and to equate the output variable divided by the input variable to a function of time and the equation constants. The resultant transfer functions are then combined to generate the overall system model. Because an existing system is being analyzed, these transfer functions can be determined either by calculation or measurement. Since calculated models often do not match the actual process, test data is taken to determine actual system component gains and time constants. This data is then used to validate the analytically determined transfer functions and adjust them as necessary to more closely match system behavior.

Plant Data

The plant is comprised of the letdown valves which control system pressure and the process which includes the piping, reactor vessel, pumps, heat exchangers, and strainer. In order to determine the transfer functions for the plant components, certain operating parameters must be known. Table 1 shows plant data taken on June 27, 1996. Column 1 lists actual system pressure in psig and the corresponding 3-15 psig signal, column 2 lists the makeup flow into the system in gpm, column 3 lists the letdown flow leaving the system in gpm, and column 4 lists the valve position showing the percent closed and the corresponding 3-15 psig signal. The reactor was in a shutdown condition with the Primary Coolant System operating at normal flow and pressure. This data was taken by adjusting the input signal to the letdown valves and observing the resulting output flow, valve

position, and system pressure changes. The controller was placed in manual to eliminate any feedback effects, providing open loop responses.

Table 1. Plant pressure, flow, and valve position data taken on June 27, 1996.

Pressure psig/psig	Makeup Flow gpm	Letdown Flow gpm	Valve Position % open/psig
480/6.84	90	89	27.5 / 6.3
470/6.744	108	107	32.5 / 6.9
450/6.6	140	138	38.33 / 7.6
430/6.44	161	158	43.33 / 8.2
400/6.2	190	181	51.67 / 9.2

Letdown Valves

Although three letdown valves are normally in service during system operation, they will be modeled as one valve. The letdown valves are of the linear flow/displacement type, therefore, flow and displacement are related by the equality

$$q_o = xK_v \qquad (1)$$

where q_o = Flow out of valve

x = Displacement of valve stem

K_v = Flow characteristic of valve

Interpolating from the data in Table 3.1, during normal operation at 468 psig, q_o =

approximately 110 gpm and x = about 32.5% closed. 32.5% equates to 0.325 inch for one valve or 0.975 inch for the three letdown valves combined. Therefore, from the above equation:

$$K_v = \frac{110}{0.975}$$

$$= 112.82 \ gpm/inch$$

The letdown valves are air operated by a reverse diaphragm, spring opposed actuator which causes the valve stem to rise on an increase in pressure. The relationship between the pressure drop in the sense line to the actuator and the air flow in that sense line is the Ohm's law analogy

$$q_{air} = \frac{p_i' - p_a'}{R} \qquad (2)$$

where q_{air} = Air flow in sense line

p_i' = change in input pressure signal

p_a' = pressure change in actuator chamber

R = Pneumatic resistance of sense line

When air flows into the actuator chamber, the pressure will rise in relation to the capacitance of the chamber. This relationship is expressed as

$$p_a' = \frac{1}{C}\int q_{air}\,dt = \frac{q_{air}}{CD} \qquad (3)$$

where C = capacitance of actuator chamber

q_{air} = Air flow into chamber

D = d/dt

Pressure is also related to valve stem displacement by the equation

$$x' = \frac{p_a' A}{k} \tag{4}$$

where x' = valve stem displacement from initial position

p_a' = pressure change in actuator chamber as before

A = diaphragm area, 75 in^2 for one valve or 225 in^2 for three valves

k = spring constant

For a p_a' of 12 lb/in^2 the valve stem displacement is 3 inches and k can be calculated as follows:

$$k = \frac{(12)(225)}{3}$$

$$= 900 lb/inch$$

The transfer function for the three letdown valves modeled as one valve can now be determined. Combining equations (2) and (3) and then substituting in equations (1) and (4):

$$\frac{p_i' - p_a'}{R} = p_a' CD$$

$$p_i'=p_a'RCD+p_a'=p_a'(RCD+1)$$

$$p_i'=\frac{x'k}{A}(RCD+1)=\frac{q_o'k}{K_vA}(RCD+1)$$

$$\frac{p_i'}{q_o'}=\frac{k}{K_vA}(RCD+1)$$

$$\frac{q_o'}{p_i'}=\frac{K_vA/k}{1+RCD}$$

Substituting in the values determined above, namely:

K_v = 112.82 gpm/in

A = 225 in^2

k = 900 lb/in

and taking the Laplace transform

$$\frac{Q_o'}{P_i'}=\frac{(112.82)(225)/900}{1+RCs}$$

$$\frac{Q_o'}{P_i'}=\frac{28.21}{1+RCs}$$

Since the letdown valves have a stroke time of about 10 seconds, this time will be used as the settling time or 4τ value and the time constant will initially be chosen as ¼ that time or 2.5 seconds. This time may require adjustment later. $G_v(s)$, the transfer function of the valve now appears as:

$$\frac{Q_o'}{P_i'}=\frac{28.21}{1+2.5s}$$

From the data contained in Table 1, flow versus valve position is plotted and displayed as Figure 3.1. From this plot it can be seen that the gain of the letdown valves is, in application, not linear in the range of interest. It is close to linear, however, and will be approximated as linear by a straight line drawn tangent to the curve at the operating point. An examination of this line indicates the actual gain to be approximately 40. A gain of 40, therefore, will be used instead of the 28.21 calculated earlier making the valve transfer function

$$G_v(s) = \frac{40}{1+2.5s}$$

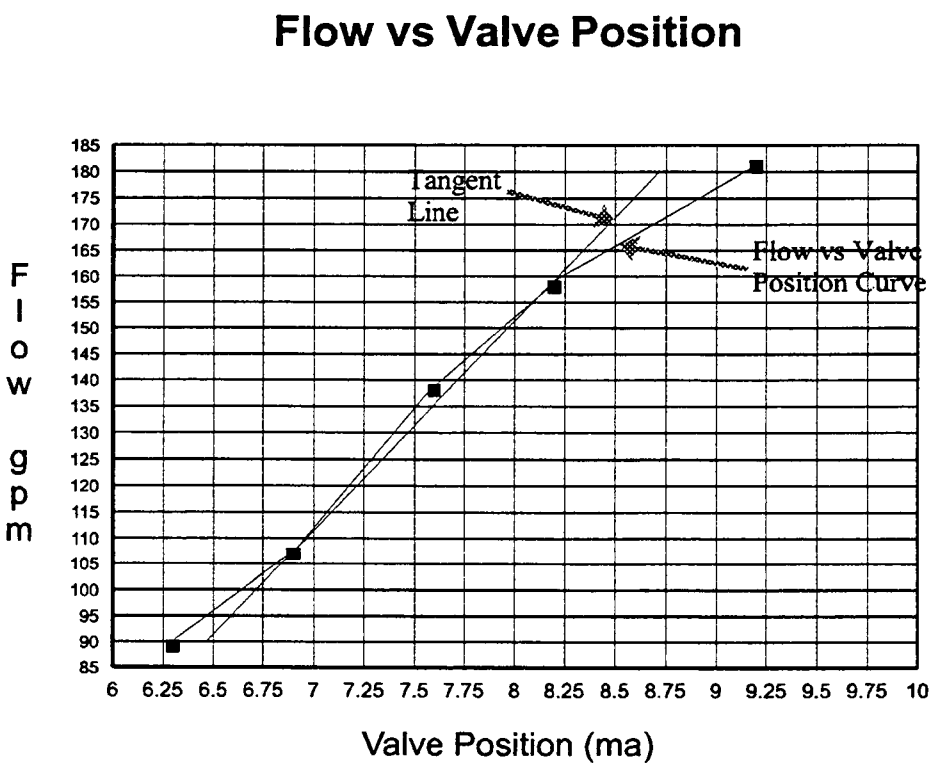


Figure 3.1. Curve of Flow Versus Valve Position

Process

The Primary Coolant System is modeled as a fixed volume chamber with an inflow and outflow corresponding to the makeup and letdown flows respectively. This system can normally be expressed by the equality

$$q=q_{in}-q_{out} \tag{5}$$

where q_{in} is the inflow rate, q_{out} is the outflow rate, and q is the net flow rate into the chamber. Since pressure in the system is being controlled by the letdown or outlet valves, we are concerned with the net flow rate out of the chamber rather than the net flow rate in. Therefore the net flow rate out of the system can be expressed by the equation

$$q=q_{out}-q_{in} \tag{6}$$

For a system in equilibrium, $q_{in} = q_{out}$ and $q = 0$. If a change in input flow (q_{in}') or output flow (q_{out}') occurs, then the net flow rate out of the system can be expressed as

$$q=q'_{out}-q'_{in} \tag{7}$$

As in the valve actuator previously discussed, a certain capacitance must be accounted for in the chamber. When dealing with fluids, compressibility rather than capacitance represents the stored energy in a chamber. Also used is the reciprocal of compressibility, the bulk modulus. Liquids at high pressures are governed by the bulk modulus relationship

$$q=\frac{V}{\beta}p'D \tag{8}$$

where V = the volume of the chamber, 1559.7 ft^3, or 11697.75 gal

β = the bulk modulus of the fluid, approximately 330,000 lbf/in^2 at 120 °F

p' = the pressure change in the chamber due to q

D = d/dt

The change in inflow rate to the chamber q_{in}', can be determined from the Ohm's law analogy

$$q'_{in} = \frac{p'}{R} \tag{9}$$

R in this equation is the ratio of pressure to flow and represents the gain of the process.

With q and q_{in}' defined, the transfer function for the process can now be determined. Substituting equations (8) and (9) into equation (7) and rearranging terms:

$$\frac{V}{\beta} p'D = q'_{out} - \frac{p'}{R}$$

$$q'_{out} = \frac{V}{\beta} p'D + \frac{p'}{R}$$

$$q'_{out} = p'\left(\frac{V}{\beta} D + \frac{1}{R}\right)$$

$$\frac{p'}{q'_{out}} = \frac{1}{\frac{1}{R} + \frac{V}{\beta} D}$$

Taking the Laplace transform:

$$\frac{P'}{Q'_{out}} = \frac{R}{1 + \frac{RV}{\beta} s}$$

With values for V and β already known, R must be found before the process transfer function can be determined. Since R is the gain of the process, we must find the change in system pressure for a given change in outlet flow. Figure 3.2 is a plot of pressure versus outlet flow for the process taken from the data in Table 1. It can be seen from this plot that this pressure/flow relationship is not linear or close to linear as is the letdown valve curve. Therefore, the gain of the process is not a constant. To determine the gain at the normal operating pressure of 468 psig a tangent line is drawn at that point. The slope of this line is approximately 0.625 psig/gpm and represents the process gain at the operating point and its vicinity. Below about 450 psig the slope of the curve increases and the

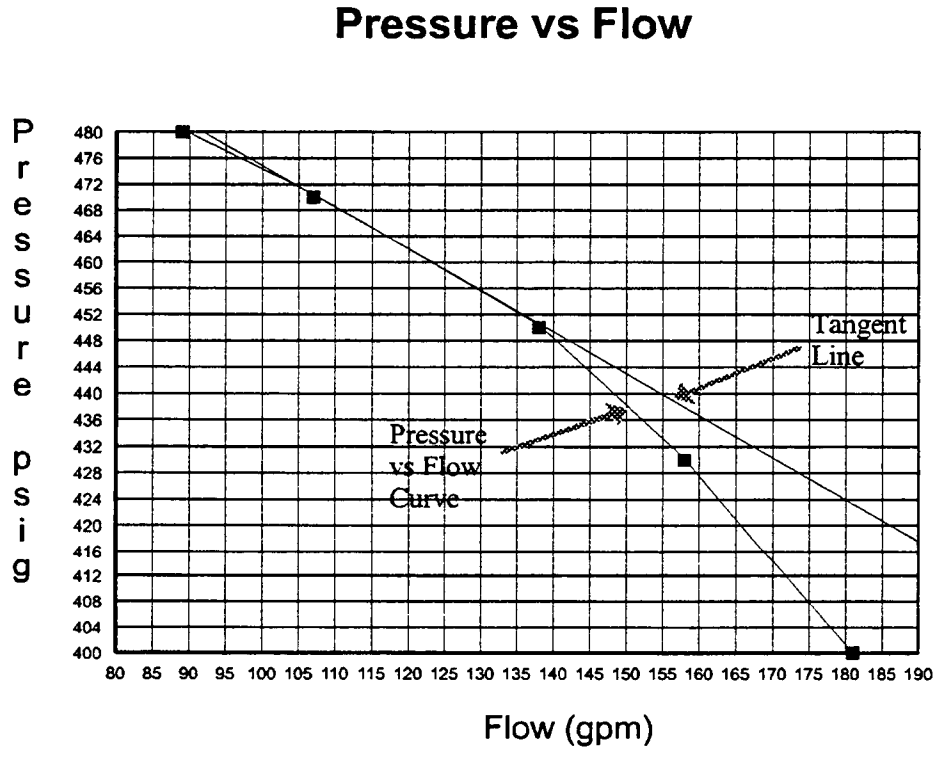


Figure 3.2. Curve of Pressure Versus Flow

tangent line drawn at 468 psig ceases to be a good approximation for the pressure/flow relationship. In fact, any single line drawn is a poor approximation for the range of pressure shown in Figure 3.2. Therefore, the gain determined from this approximation is only valid over a small range (about 450 to 480 psig) and operating points outside this range will require that a new tangent line be drawn and a new gain calculated. It is important to realize from the pressure versus flow plot and the transfer function above that since R, the gain of the process, is not a constant that the time constant τ, is also not a constant since τ=RV/β.

With R now determined the process transfer function is calculated as follows:

$$\frac{P'}{Q'_{out}} = \frac{(0.625 psi/gpm)}{1 + \frac{(0.625 psi/gpm)(11697.75 gal)(60 sec/min)}{(330000 psi)} s}$$

$$\frac{P'}{Q'_{out}} = \frac{0.625}{1 + 1.33 s}$$

As a check, this transfer function can be determined another way. The change in pressure per unit time (dp/dt) to flow (q) can be determined by evaluating equation (8). Evaluating this equation for an input of 1 gal/min:

$$\frac{dP}{dt} = \frac{(1 gal/min)(330000 psi)}{(11697.75 gal)(60 sec/min)}$$

$$= 0.47 psi/sec$$

To end up with an output in psi, an integrator must be used. Using Laplace transform and block diagram notation, the transfer function shown in Figure 3.3 is obtained. If this block alone represented the process then the output would be a ramp function. The inlet flow, however, acts as a feedback loop allowing the process to be self-regulating. When the letdown valves close, outlet flow is reduced and pressure increases. As pressure in the process rises, the resistance to inlet flow increases, decreasing the inlet flow and thereby slowing the increase in pressure until the inlet and outlet flows are again equal. With inlet and outlet flows again equal, the pressure will again be constant. The feedback loop gain can be determined from the data in Table 1 and Figure 3.2. The tangent line drawn earlier to determine the gain at the operating point determined the ratio of pressure to outlet flow to be 0.625. The feedback loop is simply the ratio of the inlet flow to the pressure. Since inlet and outlet flows are about the same, this ratio is approximately the reciprocal of 0.625 or 1.6. The block diagram of the process is now revised to look as in Figure 3.4. This block diagram can now be reduced to yield the process transfer function using the standard closed loop equation:

$$G_p(s) = \frac{G(s)}{1+G(s)H(s)}$$

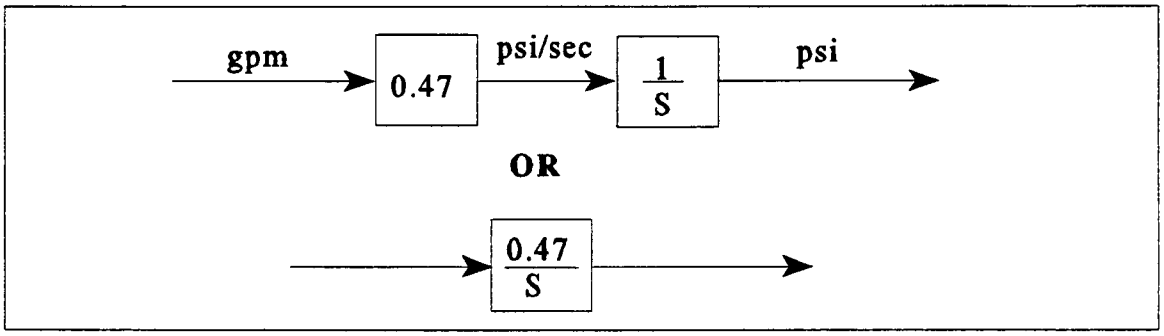


Figure 3.3. Block Diagram Derivation of Process

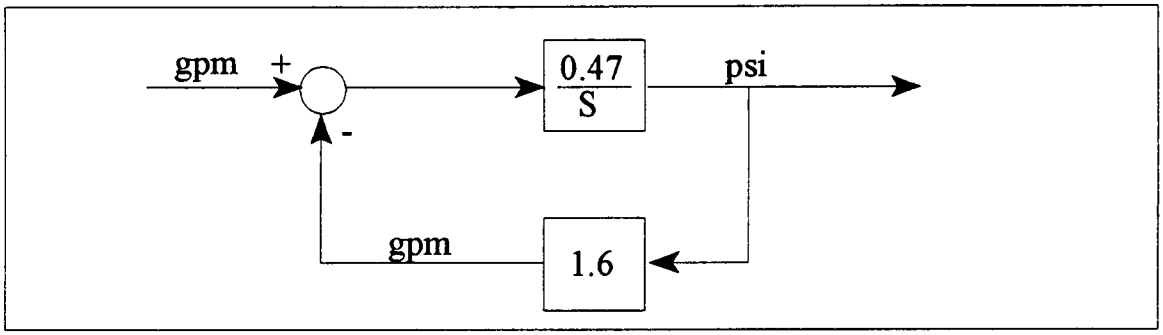


Figure 3.4. Block Diagram of Process Showing Feedback Loop

where G(s) = 0.47/s

H(s) = 1.6

Inserting the values of 0.47/s and 1.6 for G(s) and H(s) respectively and solving for $G_p(s)$:

$$G_p(s) = \frac{0.47/s}{1+(0.47/s)(1.6)}$$

$$= \frac{0.47}{s+(0.47)(1.6)}$$

$$= \frac{0.47}{s+.75}$$

$$= \frac{0.625}{1+1.33s}$$

which is the same transfer function as that determined for the process earlier. This transfer function, therefore, will be used as the transfer function of the process, $G_p(s)$, at the normal operating point..

The transfer function of the plant can now be determined by combining the valve and process expressions into one transfer function.

$$G_{Pl}(s) = G_v(s)\, G_p(s)$$

$$= \left(\frac{40}{1+2.5s}\right)\left(\frac{0.625}{1+1.33s}\right)$$

$$= \left(\frac{16}{s+0.4}\right)\left(\frac{.47}{s+0.75}\right)$$

$$= \frac{7.52}{s^2+1.15s+0.3}$$

Now that we have calculated the plant transfer function, and before continuing on with the analysis of the remaining system components, it is prudent to compare what we have to the actual plant and be sure that the model we have created is an accurate one. At the same time the data in Table 1 was obtained, a step response of the plant near the normal operating pressure was recorded. To do this a recorder was connected to the system pressure signal and the valve input was increased suddenly, simulating a step input. Figure 3.5 is a graph of the recorded step response. The curve is not a smooth one but this is not unexpected. There are many variables that influence this systems transient response as well as its steady state response. Not only are the inlet and outlet flows varying. In addition to being pressurized, the water in the system is flowing at over 16,000 gpm. The water is heated in the reactor vessel and then cooled in the heat exchangers causing expansion and contraction. There is some loss in the system as indicated by the fact that the inlet and outlet flows in Table 1 are not equal, although they are assumed so in this analysis. In addition, the pressurizer pump which provides the makeup flow causes some pulsation in the system. Therefore, the goal is to find a transfer function which exhibits a good curve fit to the actual plant and, eventually, an overall transfer function which accurately models the whole system.

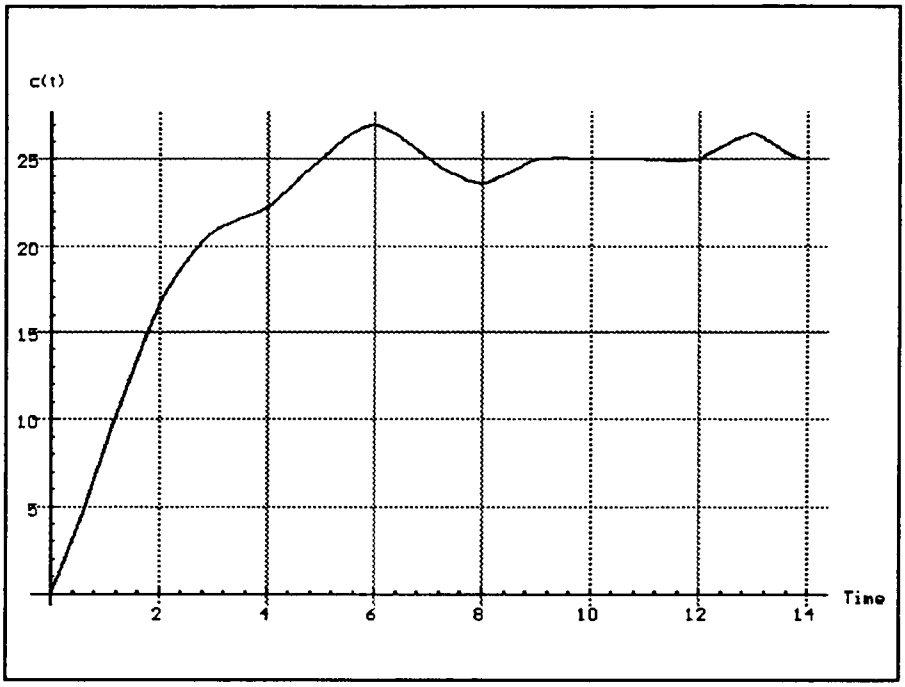


Figure 3.5. Open Loop Plant Step Response

Using the mathematical model for the plant derived above, the step response of the plant is now plotted. This step response is shown on Figure 3.6 superimposed on the actual recorded plant step response. Although it may appear that the plant is a little underdamped, an examination of the step response data beyond the 15 seconds presented in the plot shows this to be a normal fluctuation in system pressure and it is not therefore believed to be an indication of overshoot. The actual plant does, however, exhibit a faster rise time than the model so an adjustment in the model is needed. Since the time constant of the valve was somewhat arbitrarily assigned, it will be changed to 1 second and the plant replotted. Figure 3.7 is the revised plant again superimposed on the actual recorded plant step response. The correlation between the two curves is now much better so the valve transfer function will be revised to

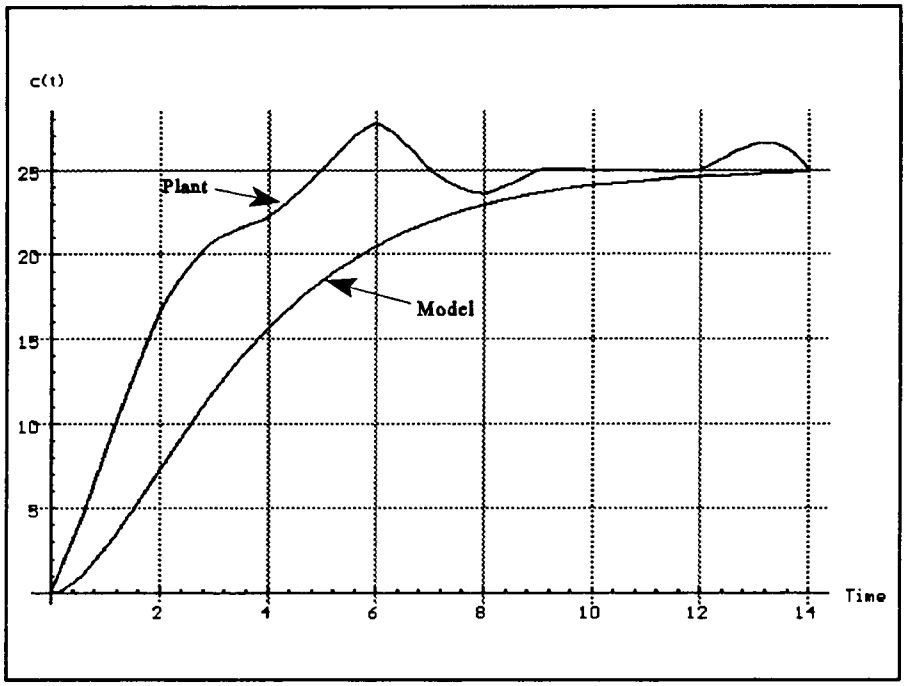


Figure 3.6. Open Loop Model Step Response Superimposed on Open Loop Plant Step Response

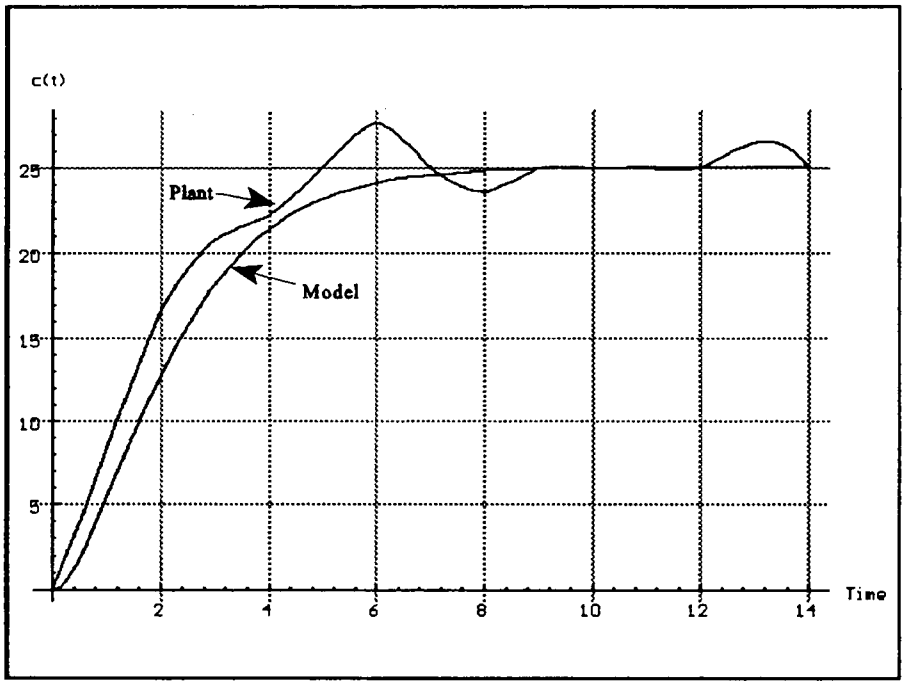


Figure 3.7. Revised Open Loop Model Superimposed on Plant

$$G_v(s) = \frac{40}{1+1s}$$

and the new plant transfer function is determined as before.

$$G_{Pl}(s) = \left(\frac{40}{1+1s}\right)\left(\frac{0.625}{1+1.33s}\right)$$

$$= \left(\frac{40}{s+1}\right)\left(\frac{.47}{s+0.75}\right)$$

$$= \frac{18.8}{s^2+1.75s+0.75}$$

Sensor

The sensor is a pressure transmitter located in the feedback path of the system. This pressure transmitter is relatively fast acting and, for now, will be assumed instantaneous. It monitors an input range of 0 to 1500 psig and outputs a corresponding 3 to 15 psig signal back to the controller for comparison with the set point. This input/output relationship can be expressed as the ratio

$$\frac{12psi}{1500psi} = 0.008$$

Therefore the gain of the feedback sensor, H(s), is 0.008 and is dimensionless.

Input Element

The input element is a part of the controller. It takes the set point and converts it into a 3 to 15 psig output signal for comparison with the sensor output signal. The set point can be

adjusted from 0 to 1500 psig. The input element is also assumed presently to be fast acting and its gain can be determined by dividing the output range by the input range thusly.

$$\frac{12psi}{1500psi}=0.008$$

The gain of the input element, A(s), is therefore 0.008, the same gain as that of the sensor.

Compensator

The compensator, as described previously, is a PID controller and therefore is expressed by the following equation.

$$M(t)=K_c[e(t)+\frac{1}{T_i}\int e(t)\,dt+T_d\frac{d}{dt}e(t)]$$

Where K_C = Proportional Gain (100/Percent Proportional Band)

T_i = integral time

T_d = derivative time

M(t) = controller output

e(t) = error signal

Dividing output by input and converting to the Laplace transform, the following equation is obtained.

$$\frac{M(s)}{E(s)}=K_c(1+\frac{1}{T_i s}+T_d s)$$

The controller is currently operating with a percent proportional band (PB) of 65, an integral time constant of 12 seconds, and a derivative time constant of 6 seconds. A

percent proportional band of 65 is converted to proportional gain (K) using the equation

$$K=\frac{100}{PB}$$

Substituting in the value for proportional band:

$$K=\frac{100}{65}$$

$$=1.538$$

The integral and derivative time constants are dial settings, not the actual integral and derivative times. They are, however proportional to the integral and derivative times (Foxboro, 1964). Due to the design and construction of most controllers there is interaction between the various modes. Interaction causes the effective values to differ from their ideal values. This interaction depends on the type of controller and whether the controller is constructed as a series or parallel controller. A series controller has its integral and derivative time constants connected in series, a parallel controller has them connected in parallel. Foxboro controllers are of the series connected type, therefore, the gain, integral, and derivative values must be determined from the following equations (Shinskey 1979).

$$K_c=K(1+\frac{D}{I})$$

$$T_i=I+D$$

$$T_d=\frac{1}{1/I+1/D}$$

Where K, I, and D are the respective gain, integral and derivative values determined above and K_c, T_i, and T_d are ideal values to be used in the compensator transfer function. Using the values determined above

$$K_c = 1.538\left(1+\frac{6}{12}\right) = 2.307$$

$$T_i = 12+6 = 18$$

$$T_d = \frac{1}{1/12+1/6} = 4$$

Substituting these values into the Laplace transform equation for the compensator:

$$\frac{M(s)}{E(s)} = 2.307\left(1+\frac{1}{18s}+4s\right)$$

which can be expressed as

$$G_C(s) = \frac{9.23s^2+2.307s+.128}{s}$$

Although the input and output signals going to and from the controller are both 3 to 15 psig signals, with the proportional band set at 100 (gain of 1), a 1 psig change of the input signal will not result in a one psig output signal change. This is due to additional inherent gain present in the system. System pressure is a function of letdown valve position and system flow. The valve stroke which is controlled by the output signal can only adjust system pressure about 300 psig by itself. The system flow rate must be changed if larger pressure changes are desired. Since a 300 psig input change (2.4 psig) results in a full scale output change (12 psig), a gain factor of 12÷2.4 or 5 exists between the controller input

and output that must be accounted for. This gain can also be seen by looking in Table 1 at the difference between the pressure signal and valve position signal changes for a given pressure change. Since the gain in question appears to exist between the controller input and output, it will be added to the controller transfer function and the transfer function for the controller is now expressed as

$$G_c(s) = \frac{(5)(9.23s^2+2.307s+.128)}{s}$$

This will be used as the transfer function for the compensator, $G_c(s)$.

System Transfer Function

The closed loop transfer function for the system can now be determined by combining the transfer functions of each system element according to the following equation:

$$\frac{C(s)}{R(s)} = \frac{A(s)G_c(s)G_v(s)G_p(s)}{1+G_c(s)G_v(s)G_p(s)H(s)}$$

Substituting in each element's transfer function yields

$$\frac{C(s)}{R(s)} = \frac{(0.008)\dfrac{(5)(9.23s^2+2.307s+.128)}{s}\dfrac{(40)}{(1+s)}\dfrac{(0.625)}{(1+1.33s)}}{1+\dfrac{(5)(9.23s^2+2.307s+.128)}{s}\dfrac{(40)}{(1+s)}\dfrac{(0.625)}{(1+1.33s)}(0.008)}$$

$$= \frac{(9.23s^2+2.307s+.128)}{s(1+s)(1+1.33s)+(9.23s^2+2.307s+.128)}$$

$$= \frac{6.94s^2+1.735s+0.096}{s^3+8.692s^2+2.486s+0.096}$$

The inverse Laplace transform of the above equation is determined to be:

$$c(t) = 1 - \frac{0.831068}{e^{8.39698t}} + \frac{0.229071}{e^{0.248656t}} - \frac{0.398003}{e^{0.0460931t}}$$

The step response of the system model is plotted in Figure 3.8.

To determine if the mathematical model is an accurate one, a step response of the system is needed for comparison purposes. On March 5, 1996, with the system pressurized, the

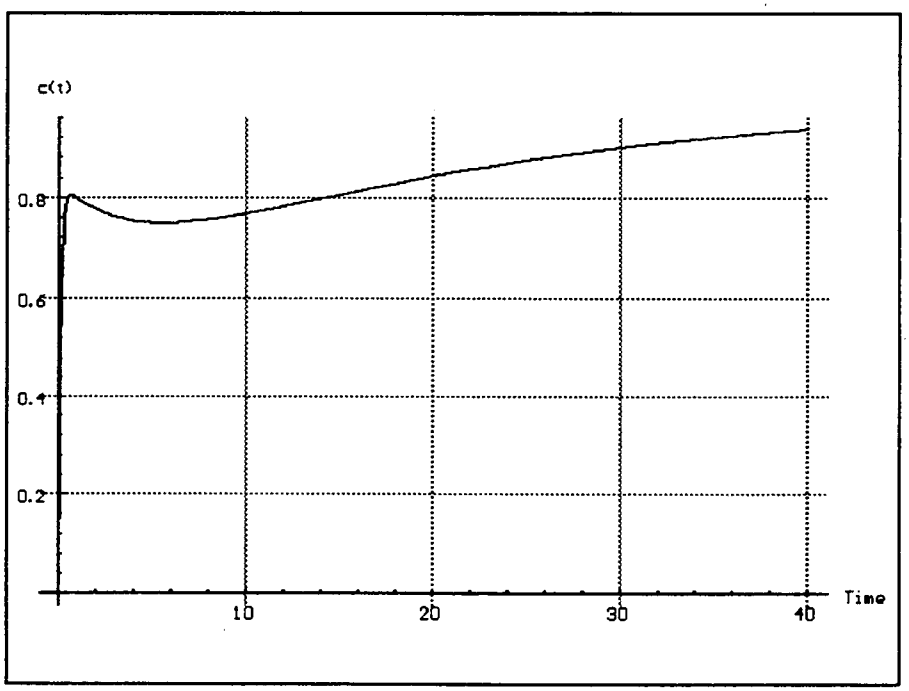


Figure 3.8. Step Response of System Model

controller in automatic, and a recorder connected to indicate system pressure, the set point of the controller was suddenly increased to simulate a step increase at different values of pressure. The step response of the system at the normal operating point of 468 psig was recorded. Figure 3.9 is the step response of the mathematical model superimposed on the actual system step response recorded. As can be seen, the model exhibits a much faster rise time than the plant. Apparently there are lags and/or delays in the plant that have not been accounted for in the model. Since the open loop model looks pretty good, the transfer functions for the valve, $G_v(s)$, and the plant, $G_p(s)$, will be left alone. Lags will be added to the input element, A(s), the controller, $G_c(s)$, and the sensor, H(s), until a combination is found that gives good results. This was experimented with at some length

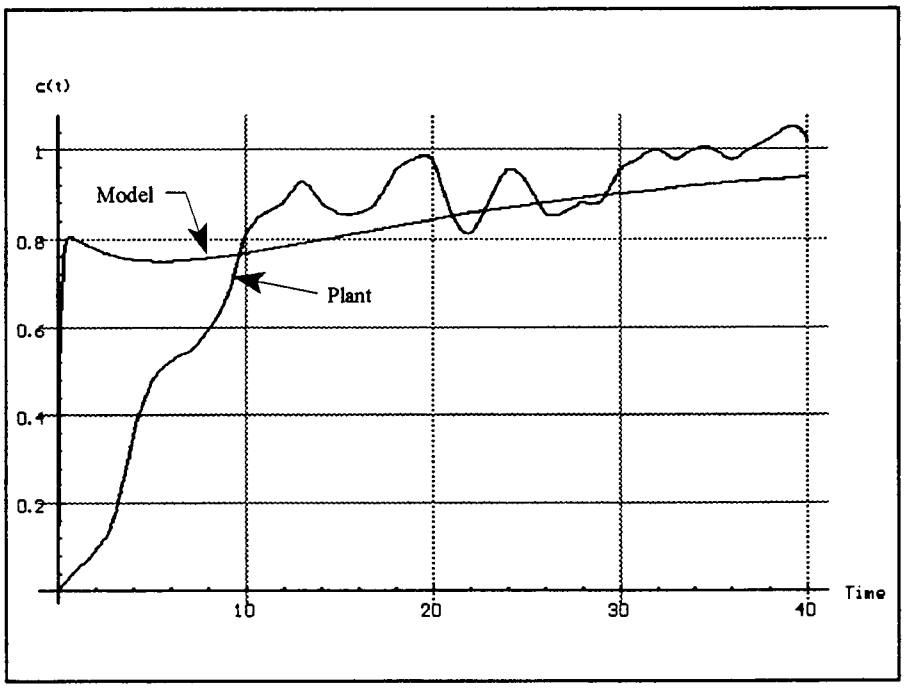


Figure 3.9. Closed Loop Step Response of Model Superimposed on Closed Loop Step Response of Plant

and the following values were settled upon.

$$A(s) = \frac{0.008}{1+3s}$$

$$G_c(s) = \frac{5(9.23s^2+2.307s+.128)}{s(1+9s)}$$

$$H(s) = \frac{0.008}{1+3s}$$

The plant transfer function now becomes

$$\frac{C(s)}{R(s)} = \frac{\frac{(0.008)}{(1+3s)}\frac{(5)(9.23s^2+2.307s+.128)}{s(1+9s)}\frac{(40)}{(1+s)}\frac{(0.625)}{(1+1.33s)}}{1+\frac{(5)(9.23s^2+2.307s+.128)}{s(1+9s)}\frac{(40)}{(1+s)}\frac{(0.625)}{(1+1.33s)}\frac{(0.008)}{(1+3s)}}$$

$$= \frac{9.23s^2+2.307s+.128}{s(1+9s)(1+s)(1+1.33s)(1+3s)+9.23s^2+2.307s+.128}$$

$$= \frac{0.257s^2+0.064s+0.004}{s^5+2.194s^4+1.565s^3+0.655s^2+0.092s+0.004}$$

and the new model step response is shown in Figure 3.10. The new model step response and the recorded plant step response are shown in Figure 3.11. It appears that the model is

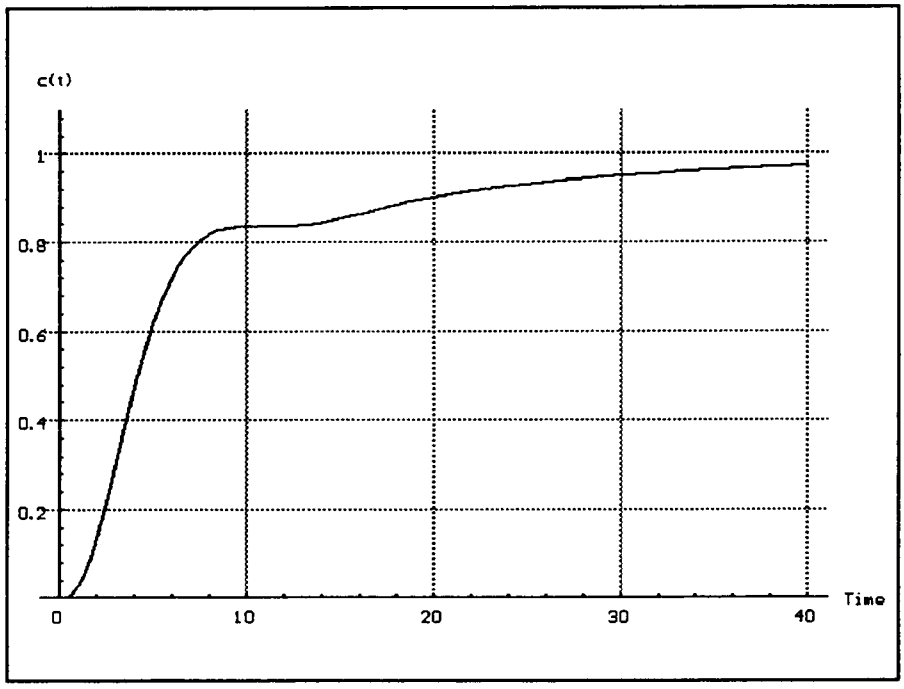


Figure 3.10. Closed Loop Step Response of Model After Adding System Delays

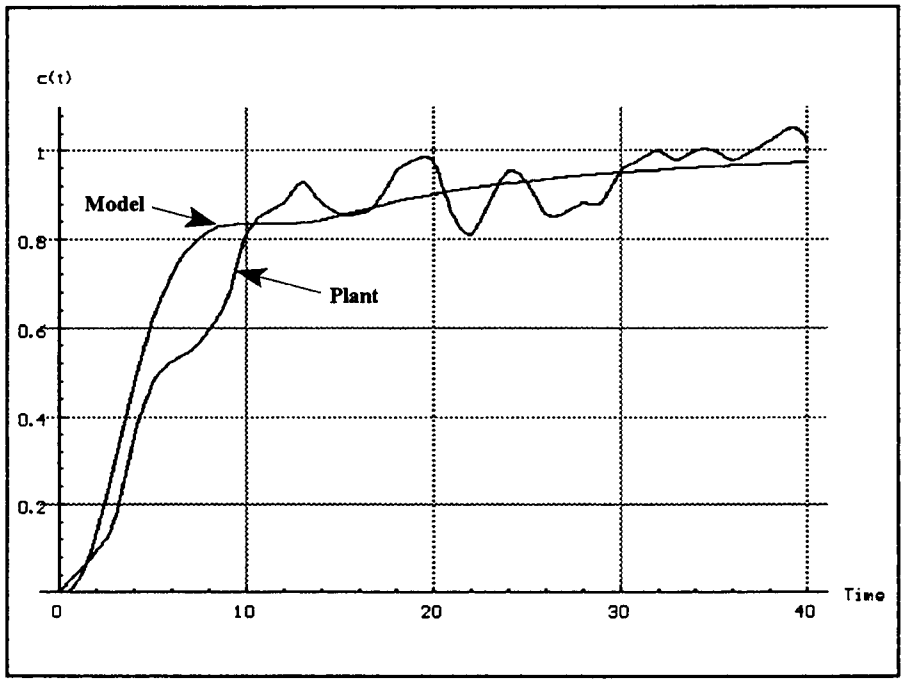


Figure 3.11. Closed Loop Step Response of Model with System Delays Superimposed on Closed Loop Step Response of Plant

now a good approximation of system performance. Figure 3.12 shows the final block diagram with the transfer functions included for each element of the system.

To ensure that the model determined is appropriate, the step response of the system at another operating point will be examined. We will look at 405 psig since this pressure is the farthest value from 468 psig for which a step response was obtained. This step response is shown in Figure 3.13. As can be seen , this response is underdamped and exhibits a faster rise time that analyzed at 468 psig. The mathematical models for the input element and sensor are constants and remain the same. The valve model gain and time constant have been assumed as constants and, therefore, also do not change. The controller model also remains as before since controller settings are not changed. Only the process model will change since the pressure/flow relationship is not linear. The gain of the process can be determined by drawing a line tangent to the pressure/flow curve of Figure 3.2 at the 405 psig operating point. This is shown in Figure 3.14. By looking at the slope of the line, this gain is determined to be approximately 1.3. According to earlier discussions, since the time constant of the process is dependent on the gain, the new time constant of the process is calculated thusly:

$$\tau = \frac{(R)\,(V)}{\beta}$$

$$= \frac{(1.3 psi/gpm)\,(11697.75 psi/gpm)\,(60 sec/min)}{330{,}000 psi}$$

$$= 2.77 sec$$

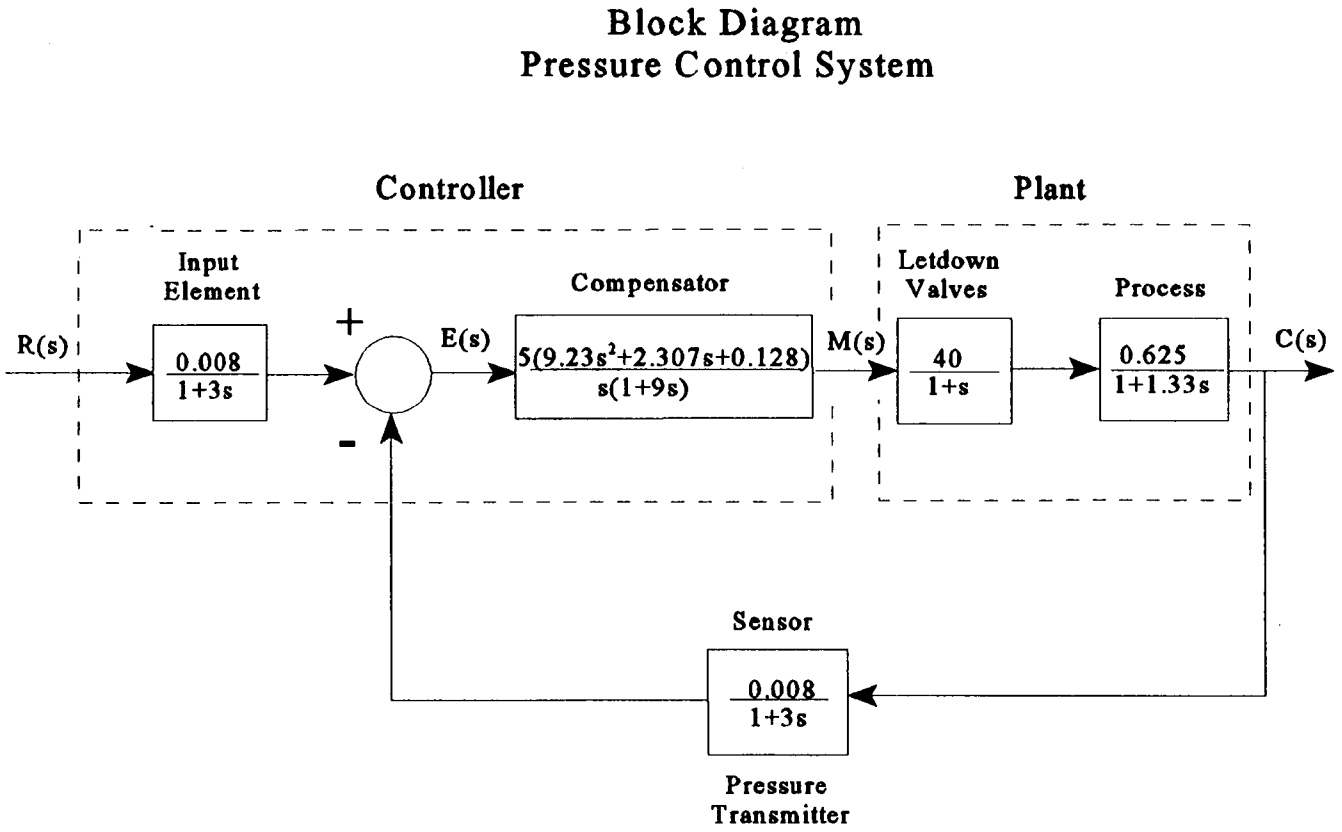


Figure 3.12. Block Diagram of Pressure Control System with Transfer Functions Indicated

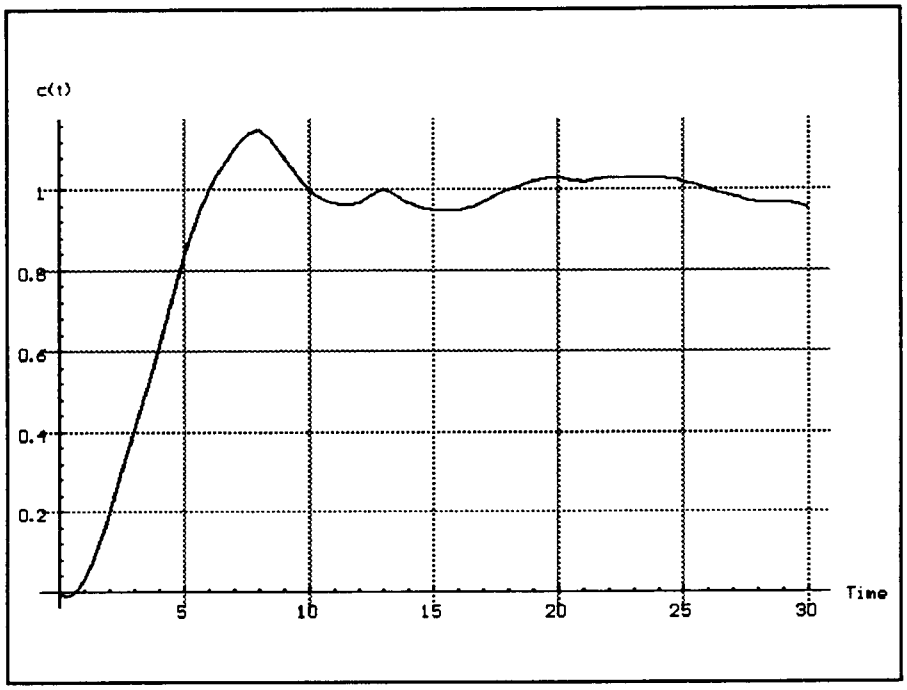


Figure 3.13. Closed Loop Step Response of Plant at 405 psig

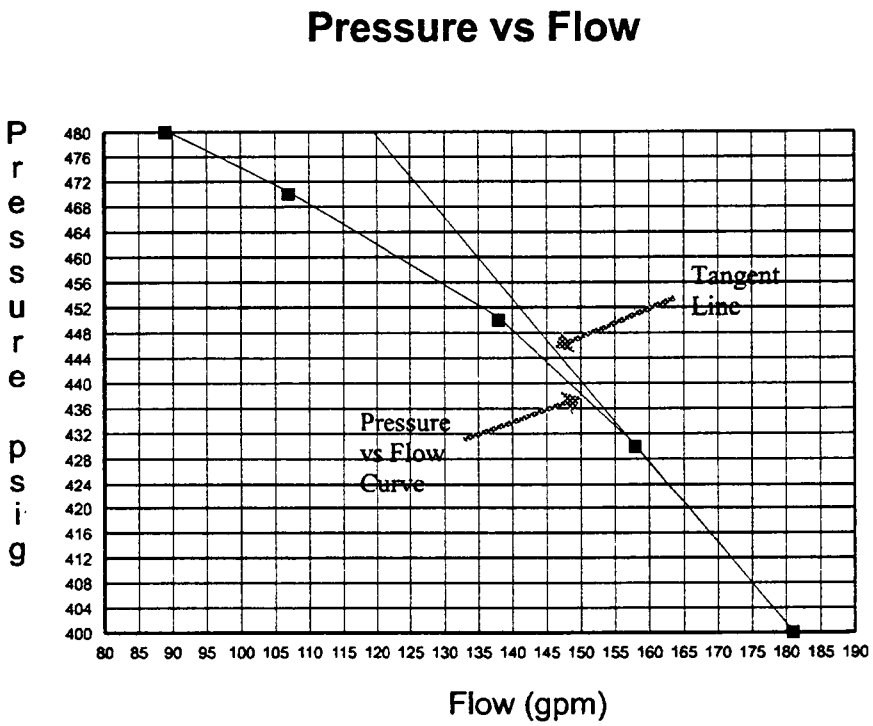


Figure 3.14. Curve of Pressure Versus Flow

This results in the process transfer function

$$G_{pr} = \frac{1.3}{1+2.77s}$$

The new system transfer function becomes

$$\frac{C(s)}{R(s)} = \frac{\frac{(0.008)}{(1+3s)}\frac{(5)(9.23s^2+2.307s+.128)}{s(1+9s)}\frac{(40)}{(1+s)}\frac{(1.3)}{(1+2.77s)}}{1+\frac{(5)(9.23s^2+2.307s+.128)}{s(1+9s)}\frac{(40)}{(1+s)}\frac{(1.3)}{(1+2.77s)}\frac{(0.008)}{(1+3s)}}$$

$$= \frac{(2.08)(9.23s^2+2.307s+.128)}{s(1+9s)(1+s)(1+2.77s)(1+3s)+(9.23s^2+2.307s+.128)(2.08)}$$

$$= \frac{0.257s^2+0.064s+0.004}{s^5+1.804s^4+1.001s^3+0.467s^2+0.078s+0.004}$$

After taking the inverse Laplace transform, the output response to a unit step is now plotted in Figure 3.15 superimposed upon the system response obtained at 405 psig. The model is quite close to the actual system response obtained experimentally. One can conclude from these two cases that the mathematical model is a reasonable approximation for the pressure control system being studied.

Load Changes

Now that some confidence has been developed in the system model, the effect of a load

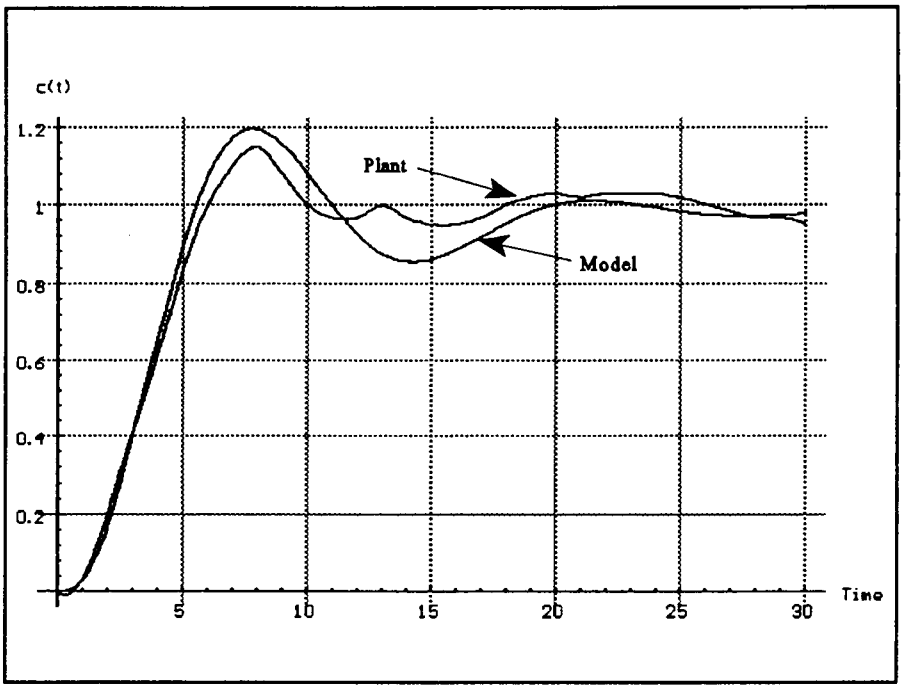


Figure 3.15. Closed Loop Step Response of Model at 405 psig Superimposed on Closed Loop Step Response of Plant at 405 psig

change, otherwise known as a disturbance, will be investigated. Although the set point of the system is adjusted every month, the controller is constantly dealing with disturbances, primarily due to inlet and outlet flow fluctuations. Therefore, it is a goal not only to optimize controller settings from an input standpoint but from a load standpoint as well. The most likely disturbance would be a flow disturbance that, if looking at Figure 3.12, would be introduced into the system between the letdown valves and the process. The closed loop transfer function for a flow disturbance at this point is

$$\frac{C(s)}{D(s)} = \frac{G_p(s)}{1 + G_c(s)\,G_v(s)\,G_p(s)\,H(s)}$$

$$= \frac{\dfrac{0.625}{(1+1.33s)}}{1+\dfrac{5(9.23s^2+2.307s+.128)}{s(1+9s)}\dfrac{40}{(1+s)}\dfrac{0.625}{(1+1.33s)}\dfrac{0.008}{(1+3s)}}$$

$$= \frac{(0.625)(s)(1+9s)(1+s)(1+3s)}{s(1+9s)(1+s)(1+1.33s)(1+3s)+(9.23s^2+2.307s+.128)}$$

$$= \frac{0.47s^4+0.679s^3+0.226s^2+0.017s}{s^5+2.194s^4+1.565s^3+0.655s^2+0.092s+0.004}$$

After taking the inverse Laplace transform, the output response of this simulation to a unit step is plotted in Figure 3.16. The goal in the controller setting optimization process is to determine if there are controller settings that will improve input response as well as disturbance rejection.

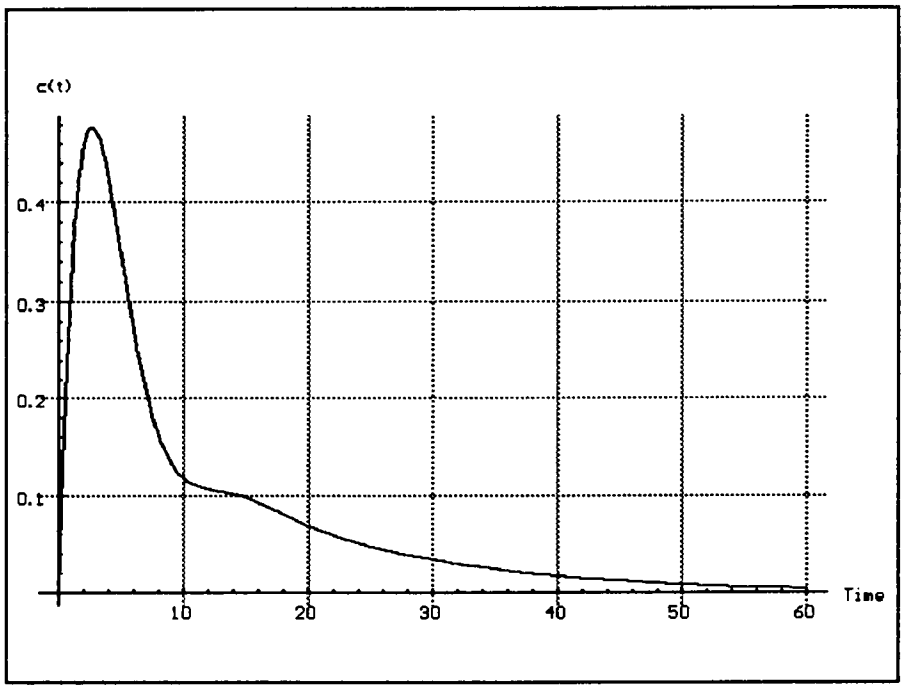


Figure 3.16. Simulation of Closed Loop System Response to a Flow Disturbance

Chapter 4

System Improvements

Before system improvements can be made, it is important to understand the objective. In other words, how do we want the system to respond to input and load disturbances. To arrive at desired performance criteria, another look at the system is required.

The pressure control system must control within certain limits. If the pressure rises to 493 psig an alarm will sound. At 518 psig the pressurizer pump will trip and the reactor will shutdown. Likewise, if pressure drops to 418 psig an alarm will sound and at 393 psig, certain safety actuations begin to take place. It is therefore important to minimize overshoot and undershoot during set point and load changes although a little can probably be tolerated. In order to ensure that safety and operating limits are not violated it is also important for the system to respond quickly to changes, especially load disturbances. During system pressurization, set point adjustment is much easier and less likely to produce unwanted fluctuations and alarms if system response is fast with minimal overshoot. Therefore, when examining possible system modifications, improvements will be those changes that can minimize overshoot and decrease system response time.

Controller Settings Optimization

Of all the possible system changes to implement, changing controller parameters is probably the easiest and should be attempted before more drastic measures are tried. For open loop plant step responses that do not overshoot, optimal PID controller parameters can be determined by using the first-order-lag-plus-delay (FOLPD) PID controller tuning formulas (Cheng, Hung 1985). In this method, values for plant gain K, delay time τ, and time constant T are determined either graphically or mathematically and inserted into the appropriate tuning formulas depending on whether the system is a tracking system or regulating system. The two things to be determined before this technique can be applied are the method for determining K, τ, and T and the type of system response you want, be it regulating or tracking.

The graphical methods considered for determining K, τ, and T are the Fit 1, Fit 2, and Fit 3 model methods. Analytically, the least-squares parameter algorithm is considered. The least-squares algorithm gives the best results but is an iterative process requiring a computer program or many trial and error calculations by hand. The Fit 3 model method is straight forward in its application, is the most accurate of the three graphical methods and gives results that rival the least-squares approach. Therefore, the Fit 3 model method will be used to determine the K, τ, and T parameters. The Fit 3 model method uses the following equations:

$$K = c(\infty) - c(0)$$

$$t_{0.284} = \tau + \frac{T}{3}$$

$$t_{0.632} = \tau + T$$

Where $c(\infty)$ = steady state value of the unit step response

$c(0)$ = initial value of the unit step response

$t_{0.284}$ = the time when the step response attains 28.4% of its final value

$t_{0.632}$ = the time when the step response attains 63.2% of its final value

Although the gain, K, can be determined from an examination of the step response, since we have the plant transfer function it is just as easy to get it there.

Now we look at regulating versus tracking. A regulator control system maintains a physical variable such as pressure at some constant value. A control system in which a physical variable is required to follow some desired time function is a tracking control system. Basically, a regulating system has no input, a tracking system tracks an input. Regulator control exhibits a faster rise time than tracking control when subjected to a step input but also exhibits more overshoot. Although the pressure control system being considered here is primarily a regulator, it operates as a tracking system when set point changes are made. Therefore, the pressure control system will be analyzed as both a regulator and a tracking control system and the two compared. The PID controller tuning formulas for regulating systems are

$$K_c = \frac{A_p (\tau / T)^{Bp}}{K}$$

$$T_i = \frac{T}{[A_i (\tau / T)^{Bi}]}$$

$$T_d = T[A_d (\tau / T)^{Bd}]$$

where Ap, Bp, Ai, Bi, Ad, and Bd are constants, the value of which depend on the minimum error-integrals criteria selected. The tuning formulas for tracking systems are the same except for T_i which has the form

$$T_i = \frac{T}{[A_i + B_i (\tau / T)]}$$

As recommended by Cheng and Hung, the modified IAE criterion and the modified ITAE criterion will be used for regulating and tracking systems respectively. The values for these criteria are listed in Table 2.

Table 2. Recommended Criterion for Regulating and Tracking Systems

Criterion	A_p	B_p	A_i	B_i	A_d	B_d
Modified IAE	3.0	-0.921	0.878	-0.749	0.482	1.137
Modified ITAE	1.2	-0.855	0.796	-0.147	0.308	0.929

Since the input element, A(s), and the feedback element, H(s), are the same, the system block diagram of Figure 3.12 can be reduced to a unity feedback system with just three

elements; the controller, the plant and a 0.008/(1+3s) component (Figure 4.1). This component, the plant, and the 9 second controller lag are combined to produce the open loop transfer function from which the step response in Figure 4.2 is obtained. The Fit 3 parameters can now be graphically determined using this figure. The times when the step response attains 28.4 percent and 63.2 percent of the final value are determined and plugged into the Fit 3 equations.

$$8.1=\tau+\frac{T}{3}$$

$$14.9=\tau+T$$

Reduced Block Diagram
Pressure Control System

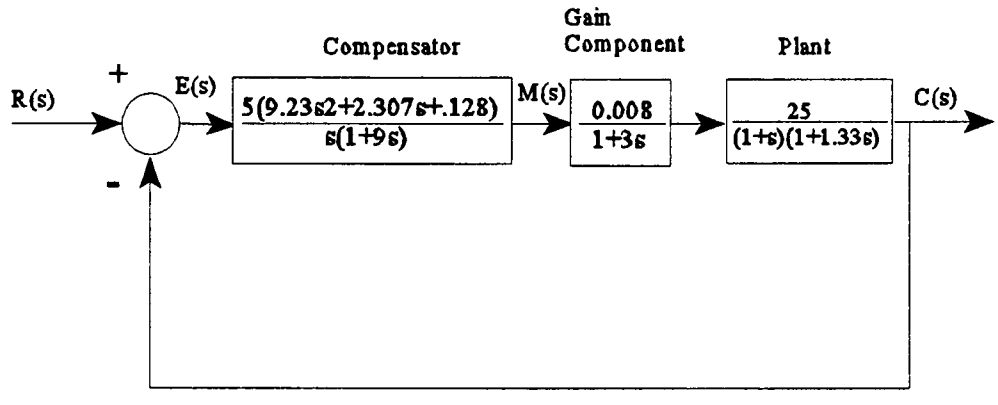


Figure 4.1. Pressure Control System Reduced Block Diagram

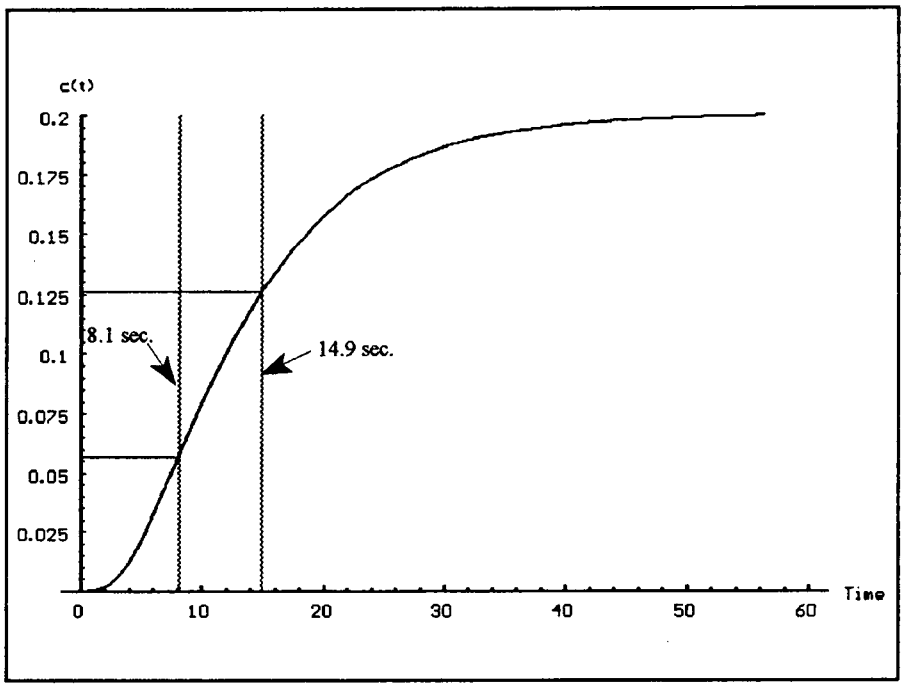


Figure 4.2. Graphical Determination of τ and T

The two simultaneous equations can now be solved for τ and T.

$$14.9=\tau+T$$

$$8.1=\tau+\frac{T}{3}$$

$$6.8=\frac{2T}{3}$$

$$\therefore T=10.2,\ \tau=4.7$$

K is the gain of the plant multiplied by the component gain or 25×0.008 which equals 0.2. It can also be determined by a visual examination of Figure 4.2.

Now that K, τ, and T are known, these values along with the modified IAE and ITAE parameters are plugged into the regulating and tracking PID controller tuning formulas to determine new values for PID settings.

For regulating systems:

$$K_c = \frac{3.0(4.7/10.2)^{-0.921}}{0.2} = 30.62$$

$$T_i = \frac{10.2}{[0.878(4.7/10.2)^{-0.749}]} = 6.50$$

$$T_d = 10.2[0.482(4.7/10.2)^{1.137}] = 2.04$$

For tracking systems:

$$K_c = \frac{1.2(4.7/10.2)^{-0.855}}{0.2} = 11.64$$

$$T_i = \frac{10.2}{[0.796 + (-0.147)(4.7/10.2)]} = 14.01$$

$$T_d = 10.2[0.308(4.7/10.2)^{0.929}] = 1.53$$

The values of K, Ti, and Td obtained above are the optimal regulating and tracking controller PID parameters and can be substituted into the controller transfer function as follows:

Regulator $$\frac{M(s)}{E(s)} = 30.62\left(1 + \frac{1}{6.50s} + 2.04s\right)$$

Tracker $$\frac{M(s)}{E(s)} = 11.64\left(1 + \frac{1}{14.01s} + 1.53s\right)$$

These revised controller transfer functions are now combined with the other component transfer functions to produce the optimal system and disturbance transfer functions for

regulating and tracking control.

System Transfer Functions

Regulator

$$\frac{C(s)}{R(s)} = \frac{0.348s^2+0.171s+0.026}{s^5+2.194s^4+1.565s^3+0.746s^2+0.198s+0.026}$$

Tracker

$$\frac{C(s)}{R(s)} = \frac{0.099s^2+0.065s+0.005}{s^5+2.194s^4+1.565s^3+0.497s^2+0.093s+0.005}$$

Disturbance Transfer Functions

Regulator

$$\frac{C(s)}{R(s)} = \frac{0.47s^4+.679s^3+0.226s^2+0.017s}{s^5+2.194s^4+1.565s^3+0.746s^2+0.198s+0.026}$$

Tracker

$$\frac{C(s)}{R(s)} = \frac{0.47s^4+0.679s^3+0.226s^2+0.017s}{s^5+2.194s^4+1.565s^3+0.497s^2+0.093s+0.005}$$

The responses of the system with regulating and tracking controller parameters to set point and disturbance inputs are compared in Figures 4.3 and 4.4 respectively to the response of the system with the current PID controller parameters. The comparisons show definite improvements in response time but less dramatic improvement in disturbance rejection. The tracking system has the best response to set point changes and settles out more rapidly when subjected to a disturbance. Its disturbance response does, however, show a little more overshoot than the model. The regulating system has the best

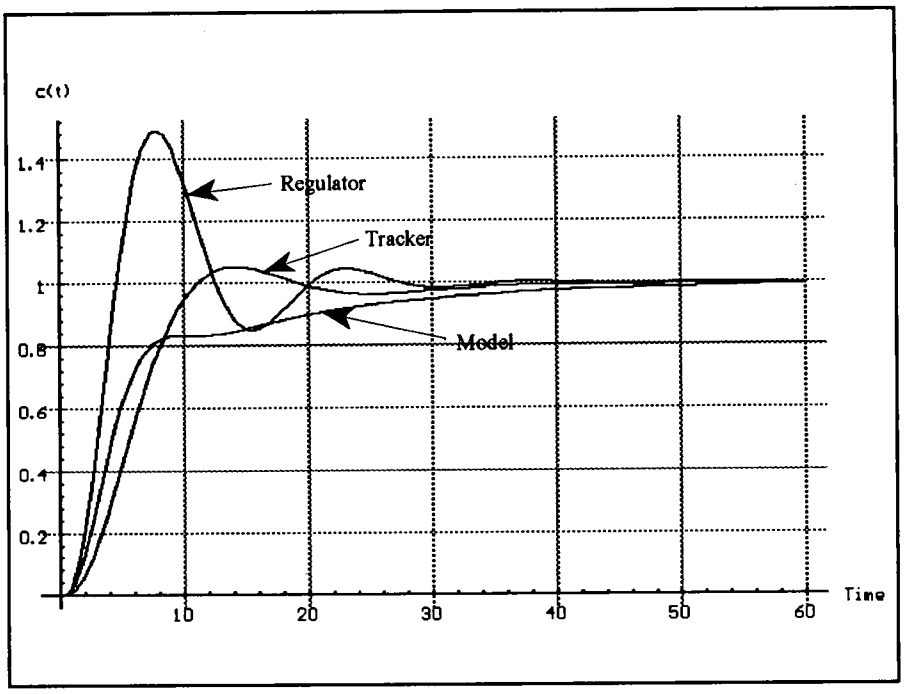


Figure 4.3. Comparison of Regulating, Tracking, and Existing System Step Responses

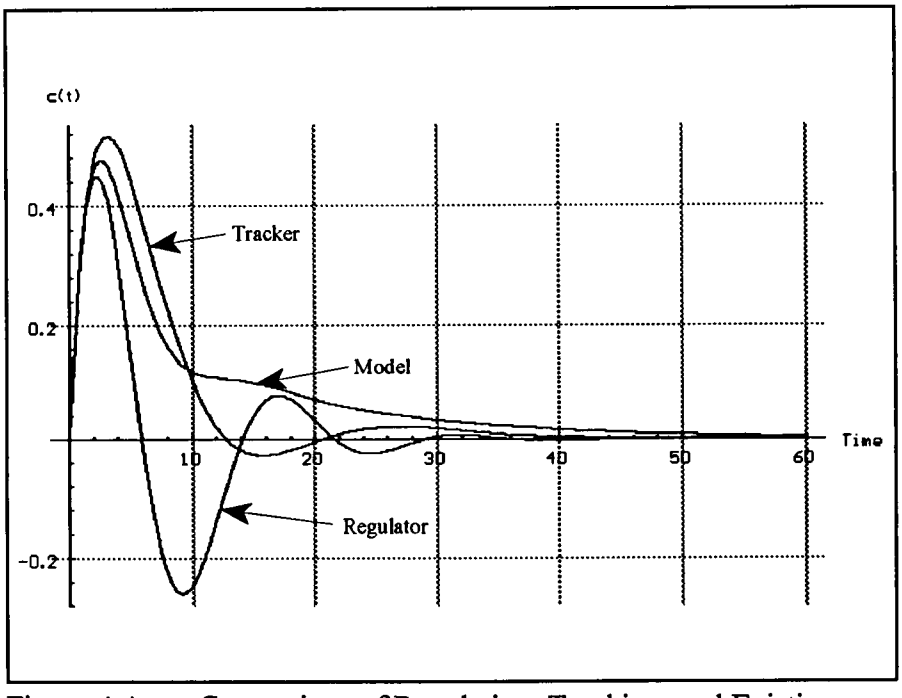


Figure 4.4. Comparison of Regulating, Tracking, and Existing System Disturbance Responses

disturbance rejection but the worst response to set point changes, having a large overshoot. Since the goal for this system is to minimize overshoot and reduce response time while improving disturbance rejection, the tracking system controller parameters, K=11.64, Ti=14.01 seconds, and Td=1.53 seconds, are chosen as the best parameters for this system. Overshoot is only about 6 percent, response is improved, and, even though disturbance overshoot is increased, the increase is minimal and is more than offset by the improved settling time. In addition, the roots of the tracker characteristic equation

$$s^5+2.194s^4+1.565s^3+0.497s^2+0.093s+0.005=0$$

are -0.081, -0.646, -1.152, -0.158+j0.242 and -0.158-j0.242 which indicate adequate stability margins. Dividing K by the controller gain constant, 5, the actual value of K is 2.33 or 43 percent proportional band.

These new controller parameters can be converted to dial settings by reversing the process described earlier. To compensate for interaction of parameters:

$$11.64=K\left(1+\frac{D}{I}\right)$$

$$14.01=I+D$$

$$1.53=\frac{1}{1/I+1/D}$$

Solving the last two equations simultaneously and then substituting into the first, the values for K, I and D are determined to be 10.21, 12.26, and 1.75 respectively. Dividing K

by the controller gain constant, 5, the new dial settings for the controller become

Gain (K) = 2.04 or 49 percent proportional band

Integral Time Constant (I) = 12.26 seconds or 0.2 minutes

Derivative Time Constant (D) = 1.75 seconds or .03 minutes

Span Adjustments

The input element and the sensor take 0-1500 psig signals and convert them to 3-15 psig signals. The gain of these components is therefore 12÷1500 or 0.008. The pressure in the primary coolant system is normally 468 psig and is never allowed, even for hydrotesting, to exceed 900 psig. The span of both the input element and the sensor could be reduced to 0-1000 psig. The gain of these components would then change to 12÷1000 or 0.012. This would increase the gain of the system and should improve system response. The results of this gain change were simulated and the step response is shown in Figure 4.5 along with the modeled step response of the system with the original span settings. Figure 4.6 is a comparison of the corresponding disturbance effects. Improvements can be seen for both step and disturbance responses with this span change. In fact, this change produces results in both step response and disturbance rejection which are arguably better than the results obtained using the tracker controller parameters

Hardware Modifications

In addition to changing the controller settings, this system analysis has uncovered other system improvements which, if implemented, could improve system performance.

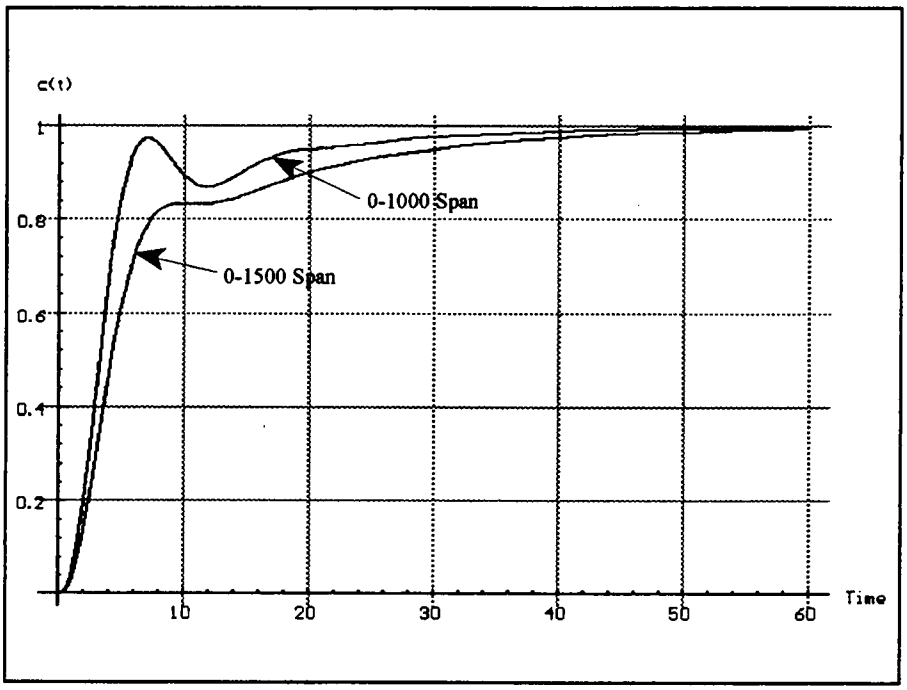


Figure 4.5. Closed Loop Step Response of Model with 0-1000 psig Span Superimposed on Closed Loop Step Response of Model with 0-1500 psig Span

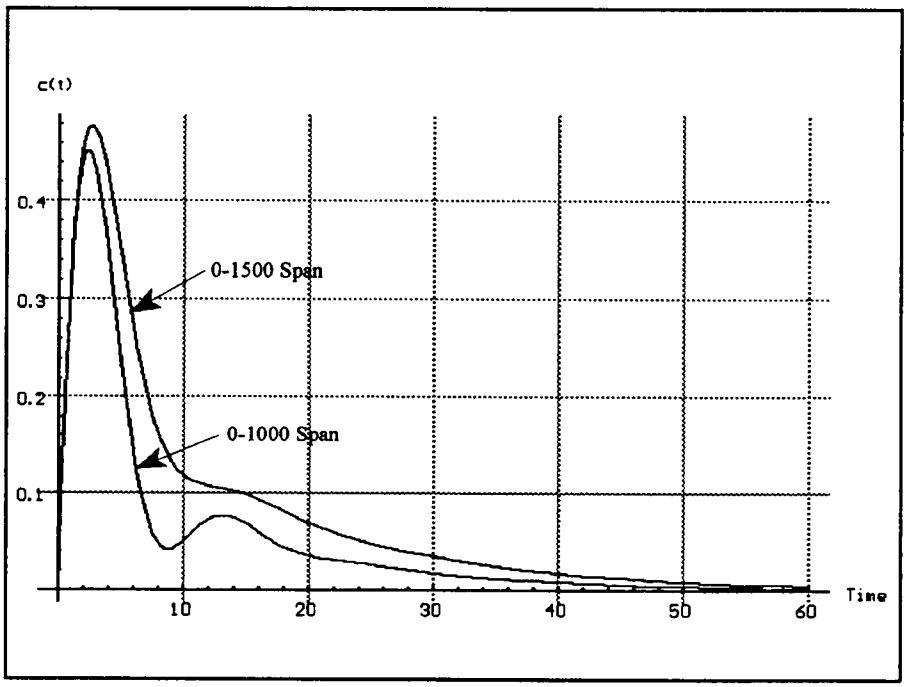


Figure 4.6. Disturbance Response Comparison for 0-1000 psig Span and 0-1500 psig Span

The Controller

First, the controller should be replaced. The controller being used is over 30 years old and utilizes a combination of mechanical linkages and bellows to perform its function. It is also constructed so that derivative action is applied regardless of whether a set point or load change has occurred. Controllers have, for many years now, been designed so that derivative action is not applied for set point changes. This prevents the unwanted anticipatory controller action that can otherwise occur and that does occur in the present system configuration. The new fuzzy logic controllers on the market today are highly recommended. They can increase pressure as fast as the system will respond and can be set to produce no overshoot.

The Valves

The valves used for pressure control are linear control valves which theoretically have a linear relationship between signal input and flow through the valve. In actuality, although not truly linear as shown in this analysis, they are somewhat linear and more linear than other valves used in similar applications (Liptak, 1985). Since the process gain is not a linear function, different loads produce different process gains and, therefore, different overall system gains. If a control valve can be selected to compensate for this nonlinearity, then the system will respond more uniformly at different loads. Although this pressure control system is set to the same pressure each operating cycle, it is desirable during pressurization to have the system give the same response to each step increase as the pressure is increased to 468 psig. Therefore, a constant overall gain is desirable. The

process gain decreases as system pressure increases creating an inverse load-to-gain relationship. Therefore, a control valve with a direct load-to-gain relationship should normally be used. Since pressure is controlled by the letdown valves, a decrease in flow, not an increase in flow causes system pressure to rise and gain to decrease. Therefore, a control valve with an inverse load-to-gain relationship is really needed. Figure 4.7 shows the load-to-gain relationships for the most widely used valve types. A quick opening valve has the required characteristics and should be considered for better overall system response.

Sense Lines

The sense lines running from the controller to the letdown valves and from the transmitter

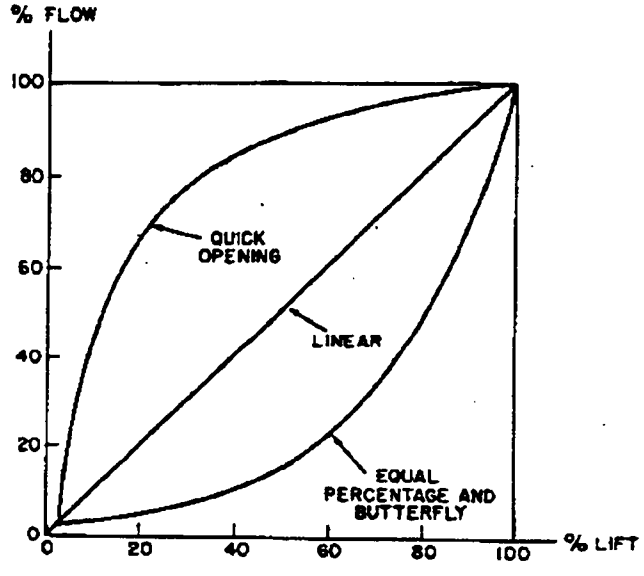


Figure 4.7 Inherent Flow Characteristics of Various Valves

to the controller are pneumatic and have inherent delays associated with them. Also, each letdown valve is a different distance from the controller so even though the valves are supposed to act in unison, they don't receive commands from the controller at exactly the same time. Pneumatic to electronic converters should be installed to transmit these signals electronically. This allows signal transmission at speed of light versus speed of sound and permits the valves to respond simultaneously. Some improvement in response time should result.

Chapter 5

Summary

This investigation has focused on the development of a mathematical model for the Pressure Control System at the High Flux Isotope Reactor and the determination of optimal controller parameters as well as other system improvements. A model has been developed that is flexible enough to accommodate the various nonlinearities of the system and produce satisfactory results at different system pressures with emphasis on the normal operating pressure of 468 psig where most disturbances will occur. This model is then used to theoretically determine optimal controller parameters after examining both regulating and tracking control systems and selecting parameters that best meet the performance criteria determined for the system. Other system improvements have been recommended to improve system performance, linearity and response time.

The Model

The model consists of a PID controller, a valve, the process, and a transmitter which provides the system feedback. The controller contains an input element which converts the 0 to 1500 psig set point to 3-15 psig. The transfer functions for each system component have been determined to be

Input Element $$A(s) = \frac{0.008}{(1+3s)}$$

Controller $$G_c(s) = \frac{5(.923s^2+2.307s+1.28)}{s(1+9s)}$$

Valve $$G_v(s) = \frac{40}{(1+s)}$$

Process $$G_p(s) = \frac{0.625}{(1+1.33s)}$$

Transmitter $$H(s) = \frac{0.008}{(1+3s)}$$

Optimal Controller Parameters

Although the Pressure Control System primarily performs a pressure regulating function, optimal PID controller parameters for a tracking control system were selected because they best met the criteria for reducing disturbance effects while minimizing overshoot and improving response time. The optimal effective values are

Percent Proportional Band = 43

Integral Time (Ti) = 14.01 seconds

Derivative Time (Td) = 1.53 seconds

These can be converted to give the corresponding controller dial settings:

Percent Proportional Band = 49

Integral Time Constant (I) = 0.2 minutes

Derivative Time Constant (D) = 0.03 minutes

Span adjustments to the input element and sensor also resulted in improvements in system response to step and disturbance inputs. These adjustments produced responses that are superior to the model and in most ways superior to the tracking control system responses.

Other System Improvements

The system was examined from a hardware standpoint and several improvements are recommended. The controller should be replaced with one that doesn't utilize derivative influence during set point changes, preferably a fuzzy logic controller that can provide optimum performance without overshoot. When it is time to replace the letdown valves, consideration should be given to using a different type of valve such as a quick opening valve which has the characteristics to offset some of the nonlinearity of the process. In addition, signals between components in the system could be sent electronically instead of pneumatically to improve response time and valve coordination.

Conclusions

The focus of this study has been to examine the performance of a pressure control system by development of a system model and then to conduct an analysis of system performance based on that model. This study has shown that model development is not an easy task. An understanding of the functionality of each component in the system is necessary in order to determine process constants and to develop the equations that are combined to form the individual transfer functions. Experimental data is also necessary in order to determine the gain values for the system components. Many data points may be necessary

in order to determine the gain curve for a nonlinear component such as the process in this study. PID controller modeling has also proven to be challenging. Dial settings are not the same as effective values. there is a difference between integral and derivative time constants and integral and derivative times on many controllers. Mode interaction can effect actual integral and derivative times. A thorough understanding of the type of controller being used is essential to successful modeling. Even after all this, the model will probably require some adjustment before satisfactory results are obtained.

The Pressure Control System in this study has been shown to be non-linear, its input response varying depending upon the pressure of the system. It is overdamped at some pressures, however, it is underdamped and slower to respond at the normal operating pressure. This is due primarily to the process. PID controllers such as the one used in this system, are primarily tuned to provide either regulating or tracking control. Some controllers, however perform both functions. Using the FOLPD methodology, both regulator and tracker PID parameters can be determined that will improve some aspects of the system response. A decision must be made in this case as to what response is most important. If disturbance rejection is more important than input response, regulating parameters should be chosen. Likewise, if minimal overshoot during input changes is more important than disturbance rejection, tracking parameters should be used. Due to the potential problems associated with overshoot on this system, tracking parameters for the PID controller are preferable and were, therefore, chosen.

This study has demonstrated that a non-linear system can be modeled successfully if you are interested in a narrow range of output or if you know the non-linear characteristics well enough to adjust the model accordingly. It has also shown that the input and disturbance responses of the HFIR Pressure Control System can be improved by adjusting the PID controller parameters and that most system response characteristics can be optimized to meet performance objectives by configuring the controller as a tracker rather than a regulator. System response improvements can also be made by adjusting the span of certain components. In addition, although controller tuning should provide notable improvements, the hardware changes proposed within will improve system linearity and response and should enhance overall system performance.

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Vita

Karl Philip Zimmerman was born in Chattanooga, Tennessee on February 10, 1956. He attended elementary school in Chattanooga, junior high school in Knoxville, Tennessee, and high school in Forest Park, Georgia where he graduated from Forest Park High School in May 1974. He entered The University of Tennessee in September 1974, and, in December 1978, received a Bachelor of Science Degree in Electrical Engineering. In January 1979, he went to work as an electrical engineer for the Tennessee Valley Authority (TVA) where he worked on the electrical and instrumentation designs of several of TVA's nuclear plants including Sequoyah, Watts Bar, and Yellow Creek. He again entered the University of Tennessee in September 1987, to pursue the Master of Science Degree in Electrical Engineering. Pursuit of his advanced degree was interrupted in June 1988, when he was transferred to Sequoyah Nuclear Plant in Soddy-Daisy, Tennessee to perform instrumentation and control system designs and instrument accuracy calculations in support of plant operations. In January 1990, he went to work at the High Flux Isotope Reactor at Oak Ridge National Laboratory as an instrumentation and controls engineer for Martin Marietta Energy Systems. In January 1992, he returned to the University of Tennessee to complete the masters program. The masters degree was received May 1997.

He is presently working for Lockheed Martin Energy Research Corporation as the Instrumentation and Controls Maintenance Manager for the High Flux Isotope Reactor, located at Oak Ridge National Laboratory in Oak Ridge, Tennessee.