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Aidong Wei

Date

April 20, 1991

**ANALYSIS AND CONDITION MONITORING
OF AIR-OPERATED VALVES**

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

Aidong Wei

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Special thanks to my wife, Y. Feng.

ABSTRACT

Air-Operated Valves (AOVs) are used in almost all fluid systems in a Nuclear Power Plant. Their failures have resulted in increased plant maintenance. More importantly, the operational readiness of nuclear plant safety-related systems has often been affected by AOV degradation and failure. Thus, in recent years AOVs have received considerable attention by the U. S. Nuclear Regulatory Commission and the nuclear power industry.

A comprehensive study of AOV operations was performed and the results of this study are presented in this thesis. The primary objective of this research is to identify valve related variables that are sensitive to valve anomalies. This was accomplished through experimental and theoretical analysis of AOV dynamics. The AOV related events in nuclear power plants were studied and typical AOV anomalies are summarized. A test loop has been constructed to develop AOV condition monitoring techniques. The operations of the test AOV were monitored by measuring air supply line pressure, stem position, and pressure drop across the valve. These three sensors were interfaced with a data acquisition system. The acquired data were studied both in the time domain and in the frequency domain. An extensive experimental study of the valve operations and both transient and steady-state signal analysis were performed. An analytical model of a typical AOV was developed. Simulations of the model responses for normal operations and with anomalies were performed and compared with experimental data.

Two important signals, stem motion as measured by a linear variable

differential transformer and the valve differential pressure, reflect the changes in the valve behavior and can be monitored using these two measurements. Both transient valve stroking signatures and normal fluctuations may be processed for establishing valve operating characteristics. The results of valve dynamic modeling confirm experimental observation. A set of guidelines for implementing this monitoring technique in power plants is outlined.

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CHAPTER I

INTRODUCTION

1.1 Air-operated Valves in Power Plants

Air-operated valves are widely used in control and protection systems in nuclear power plants and other process industries. Most safety systems in nuclear plants are required to use air-operated valves (AOVs). Other power plant systems also utilize AOVs, especially in inaccessible locations. The applications of AOVs in nuclear power plants are listed in Table 1.1 [1].

Proper functioning of air-operated valves is necessary to ensure uninterrupted plant operation. In nuclear power plants, additional concerns of radiological exposure, fast emergency operation, and safe shutdown procedures require reliable operation of AOVs.

Usually in nuclear power plants, most of the valves are tested and checked at periodic intervals. If a valve degrades, and is not serviced carefully, the degradation may continue and result in serious consequences. About 20% of the unscheduled shutdowns in power plants are due to valve-related problems [2]. Therefore, analyses and condition monitoring of all kinds of valves have been the subjects of extensive studies over the past ten years. A technique based on motor current signature analysis for condition monitoring of motor-operated valves was developed by Oak Ridge National Laboratory [3,4]. Techniques for condition monitoring of AOVs are not available in the open literature.

Table 1.1 Some Applications of AOVs at Nuclear Power Plants [1]

For Pressurized Water Reactors (PWRs)	For Boiling Water Reactors (BWRs)	For Pressurized Heavy Water Reactors
Instrument Air System	Standby Gas Treatment System	On-Line Fueling System
Hydrogen Sampling Isolation	Chilled Water	Primary Coolant Feed/Bleed Control
Pressurizer Head Vent	Condensate and Demineralization Water System	Fast Acting Emergency Shut-Down System
RC Loop Hot Leg Sampling	Instrument Air System	
RHR System Sample	Water Positive Seal System	
Steam Generator Liquid Sample	Waste Disposal System	
Steam Generator Down Sample	Alternate Rod Insertion System	
Continuous Mini-Purge	Containment Monitoring	
Diesel Generator Air Start	RHR Process Sampling	
Emergency Boration System	Nitrogen System	
Chemical and Volume Control System	Reactor Coolant Drains, Vents and Miscellaneous Piping System	
Reactor Coolant System	Leak Detection	
Emergency Charging System	Diesel Fuel Transfer Pump Suction	
Auxiliary Steam System		
Component Cooling Water		
Turbine Trip System		

An AOV analysis project was initiated in the Nuclear Engineering Department of the University of Tennessee to develop new condition monitoring techniques for AOVs.

1.2 Review of Problems Related to Air-Operated Valves

Failures of air-operated valves contribute to many unscheduled shut-downs of power plants. The Nuclear Regulatory Commission (NRC) lists a total of 520 AOV related events from 1984 to 1989 in the Licensee Event Reports [5]. A typical set of AOV failures is listed in Appendix A. A detailed review of this report shows that most AOV abnormalities are due to the following causes.

1. Air leakage through valve diaphragm and air supply line
2. Changes in valve spring properties
3. Variations in valve stem friction and seal packing tightness
4. Variations in valve plug seating
5. Bent valve stems
6. Separation of actuator and valve
7. I/P (current-to-pressure) converter problems.

A detailed compilation of anomalies of pneumatically operated valves is given in the Nuclear Plant Reliability Data System (NPRDS) general report [6]. It lists a total of 288 valve related events. Air leakage through diaphragm due to age and cycle fatigue results in most of the AOV-related events in power plants. Other most common AOV anomalies are given in Table 1.2.

Table 1.2 Most common AOV anomalies

Air leakage through stem seal or gasket damage	Valve packing being dried out
Broken pneumatic line or fittings	Air leakage through the valve yoke
Actuator upper or lower cover leakage due to age and fatigue	Actuator casing and yoke bushing wear
Water in the actuator due to accumulation of condensed water	Stripping and backing out of mounting screw threads, causing punctured diaphragm
Corrosion in the spring or the stem	Stem and diaphragm binding
Damaged diaphragm due to improper tightening of the diaphragm bolts	Diaphragm damage due to system transient over-pressure
Loose screws in the diaphragm resulting in the air leakage	Wear and aging of diaphragm base plate
Leak around actuator-to-diaphragm seal	Air leakage in the I/P convertor or air volume booster for the valve actuator
Air leakage due to the loss of fittings caused by line vibration	Large buildup of rust on the air pressure adjustment nut
	Air pressure regulator failure

The root causes of air-operated valve failures have been studied by the Institute for Nuclear Power Operations (INPO). The development of condition monitoring techniques is emphasized in the present research.

1.3 Overview of Previous Work in AOV Monitoring

The importance of developing methodologies for monitoring air-operated valves has been emphasized in the NRC information notice IE Notice 88-24 [8], titled "Failure of Air Operated Valves Affecting Safety Related Systems," and General Issue 43, titled "Air System Reliability." Air-operated valves and their diagnostics are not as well-understood as motor-driven valves. Some vendors have initiated AOV monitoring programs, including PC-based diagnostic test systems, to resolve problems and improve operability. One of these products is the "AOV Diagnostic System" developed by ABB Combustion Engineering [9]. The measurements used in their study are supply pressure, valve stroke, body strain, and system acoustics. The system can be used for benchmarking and troubleshooting. The studies performed using this system are limited to valve stroke analysis only. Detailed and well-developed techniques, including both time- and frequency-domain analyses, are presently not available.

1.4 Objectives of the Present Research

The objectives of this research are to understand the dynamic behavior of air-

operated valves, identify the measurements that are sensitive to changes in AOV operation, and to use these results for condition monitoring of AOVs. The following tasks were identified to accomplish these objectives.

1. Review AOV related events in nuclear power plants.
2. Construct a test loop for studying AOV operation.
3. Install and calibrate necessary instrumentation.
4. Identify valve-related measurements that are sensitive to valve anomalies.
5. Study valve dynamics by performing transient analysis.
6. Determine the diagnostic features of steady-state operational data using digital signal processing techniques.
7. Develop a mathematical model of a typical AOV and compare its response to actual measurements with and without valve degradation.
8. Establish guidelines for developing an on-line AOV monitoring system.

Three measurements were used to monitor the operation of AOVs. These were valve differential pressure, stem motion, and air supply line pressure. Both transient studies and steady-state signal analysis were performed for a normal AOV and for the cases with anomalies. The measurements were compared to determine their sensitivities to valve degradation. The analytical studies of the AOV dynamics using a mathematical model were compared with experimental data obtained from the test loop. Based on this study, a set of guidelines has been developed for implementation of this technique for on-line AOV monitoring.

1.5 Contributions of the Thesis

This thesis is composed of three parts: valve transient analysis, random signal analysis, and analytical model development and simulation.

A comprehensive study was made of the valve dynamic behavior during open-to-close and close-to-open strokes. Step, ramp, and sinusoidal perturbations were used to exercise the valve actuator for both normal and abnormal valve conditions. The studies showed that valve differential pressure, stem motion (as measured by a linear variable differential transformer (LVDT)), and air line supply pressure are the three important signals that reflect valve condition. This study is very comprehensive and clearly indicates the relationship between valve differential pressure and stem motion. The complete stroke cycle shows a hysteresis effect in the valve operation, thus suggesting that the variations in this characteristic are also indicative of valve anomalies.

The second important contribution of this thesis is the application of digital signal processing technique to the analysis of fluctuation (random) data. The stem position is assumed to be stationary during data acquisition. The three signals mentioned above were recorded corresponding to different valve stem positions for both the normal and faulty situations. Air leakage (through actuator diaphragm) and friction effects were simulated experimentally. The results indicated distinct variations in both time-domain and frequency-domain signatures, thus useful for tracking flow and pressure modulation effects.

The third contribution of this thesis is the development of a detailed model

of a typical air-operated valve dynamics. The simulation results of the model to different inputs were studied and compared with experimental results. The model analysis is very useful in establishing the sensitivity of the valve response to variations in valve characteristics.

The results of steady-state and dynamic behavior of AOVs may be combined to develop an on-line condition monitoring system.

1.6 Organization of the Thesis

This thesis is presented in seven chapters and three appendixes. Following this introductory chapter, the development of experimental facility and measurement system are discussed in Chapter 2. The fundamental study of the instrumentation is also included in this chapter. A detailed experimental analysis of a 3/4 inch globe valve and actuator operation is presented in Chapter 3. This includes the simulation of both normal and abnormal operations. Various aspects of measured signatures are discussed. The application of random signal analysis to steady-state valve operation is presented in Chapter 4. Chapter 5 describes an analytical model of a typical AOV. The model simulation results are presented in this chapter. The simulated results are compared with those from the transient analysis. Chapter 6 presents a methodology for monitoring AOVs. Concluding remarks and suggestions for future research are presented in Chapter 7. Typical AOV anomalies in power plants are listed in Appendix A. Appendix B lists additional bibliography. Appendix C describes the specifications of a linear variable differential transformer.

CHAPTER II

DEVELOPMENT OF THE EXPERIMENTAL FACILITY AND THE MEASUREMENT SYSTEM

2.1 Introduction

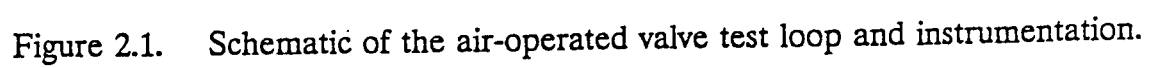
A low pressure hydraulic loop was fabricated to perform valve dynamic analyses, to identify sensitive diagnostic variables, and to develop AOV predictive maintenance techniques. The development of the test facility includes the following tasks.

1. Fabrication of the test loop.
2. Installation and calibration of instrumentations.
3. Interfacing the data acquisition system with the instrumentation.

The sensors were installed and calibrated before the measurements were made. The valve fundamentals and the theory of measurements were also studied. The data acquisition system was interfaced with a personal computer through a patch panel.

2.2 Air-Operated Valve (AOV) Test Loop

A schematic of the test loop is shown in Figure 2.1. The loop consists of a 3/4 inch globe control valve with air-operated actuator, a centrifugal circulation pump,



a heat exchanger, a pressurizer, absolute and several differential pressure gauges, a turbine flow rate meter, and several fittings and connections. The flow rate in the system is 5.6 gallons per minute when the valve is fully open.

The test AOV mainly consists of an actuator and a globe valve. The valve actuator type is air-to-open. There are six springs to push the valve stem to its closed position when the actuating air is lost. A schematic of the test valve and the actuator is given in Figure 2.2. The required AOV actuator operating pressure is from 3 to 18 lb/in² [8]. The outside diameter of the actuator diaphragm is about 12.5 inches. The moving parts weigh 2.23 lbs.

2.2.1 Test Valve Characteristics [10-15]

When a fluid flows steadily through a valve, a pressure difference is produced due to the friction and the viscosity distribution across the flow section. This pressure drop can be determined either analytically or experimentally. Figure 2.3 shows how the pressure drop across the valve is determined by the experimental approach for various valve sizes. The pressure drop across the valve is a function of the flow rate through the valve only if the valve is kept fully open. This figure is not applicable while the valve open-close status varies. Fortunately, the pressure drop across the valve can be calculated using analytical methods. It is clear that flow through a valve and fittings in the pipe line cause a reduction in static head h_i , which may be expressed in terms of flow velocity through the valve:

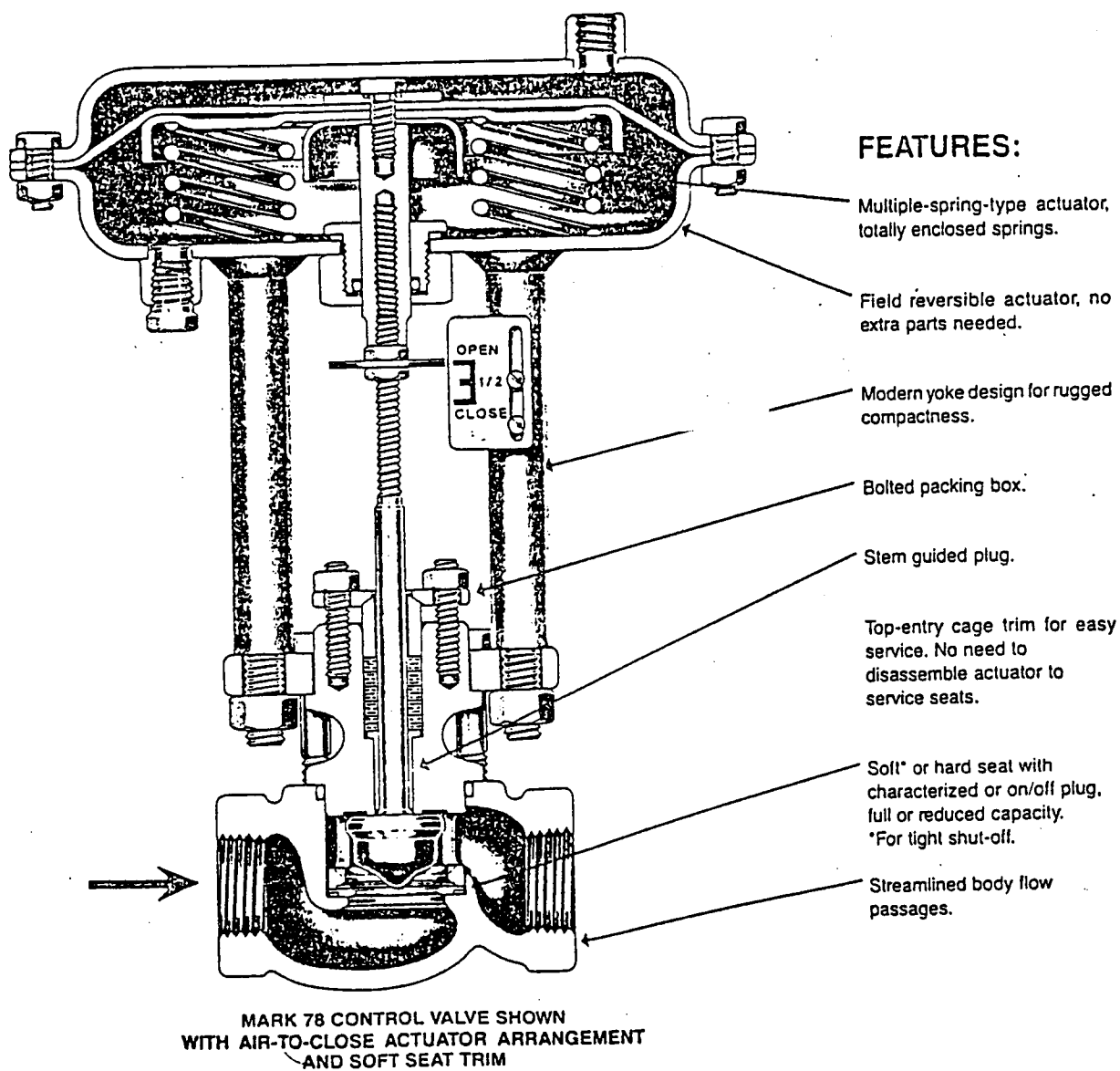
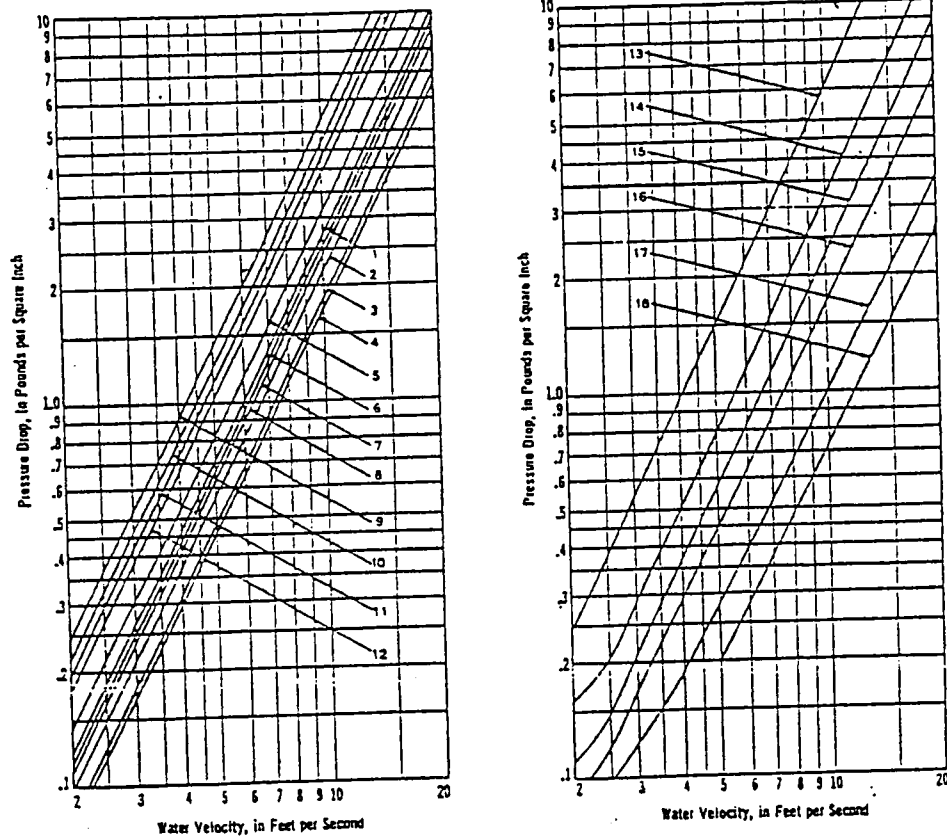


Figure 2.2. Schematic of the test valve and the actuator [9].

Crane Water Flow Tests



Water Flow Tests — Curves 1 to 18

Fluid	Figure No.	Curve No.	Size, Inches	Valve Type*
Water		1	3/4	150-Pound Cast Iron Y-Pattern Globe Valve, Flat Seat
		2	2	
		3	4	
		4	6	
		5	1 1/2	150-Pound Brass Angle Valve with Composition Disc, Flat Seat
		6	2	
		7	2 1/2	
		8	3	
		9	1 1/2	150-Pound Brass Conventional Globe Valve With Composition Disc—Flat Seat
		10	2	
		11	2 1/2	
		12	3	
		13	3/4	200-Pound Brass Swing Check Valve
		14	1/2	
		15	3/4	
		16	1 1/4	
		17	2	125-Pound Iron Body Swing Check Valve
		18	6	

*Except for check valves at lower velocities where curves (14 to 17) bend, all valves were tested with disc fully lifted.

Figure 2.3. Water velocity through the valve versus the pressure loss across the valve [10].

$$h_1 = k \frac{v^2}{2g}. \quad (2.1)$$

The resistance coefficient k in the equation, therefore, is defined as the number of velocity heads lost due to the valve and fittings. The coefficient k is

$$k = f \frac{L}{D}, \quad (2.2)$$

where the L/D refers to the equivalent length of resistance to flow in pipe diameters and can be determined from Table 2.1. The friction factor f is plotted in Figure 2.4 on the basis of relative roughness of the pipe and the Reynolds number. For laminar flow, f can be expressed as

$$f = \frac{64}{Re}, \quad (2.3)$$

where Re is the Reynolds number. This is calculated from

$$Re = \frac{Dv\rho}{\mu_e}, \quad (2.4)$$

where D is the nominal diameter of the valve or tube (m), v is the flow velocity through the valve (m/s), and ρ is the water density (kg/m^3), and μ_e is water viscosity (kg/m-s). For turbulent flow (normally $Re \geq 1000$), f is given by

$$f = 0.079 Re^{-\frac{1}{4}}. \quad (2.5)$$

It is more convenient to express the valve flow capacity and the valve flow characteristics in terms of the valve flow coefficient C_v . The C_v coefficient is defined

Table 2.1 Representative equivalent length in pipe diameters (L/D) of various valves and fittings [10].

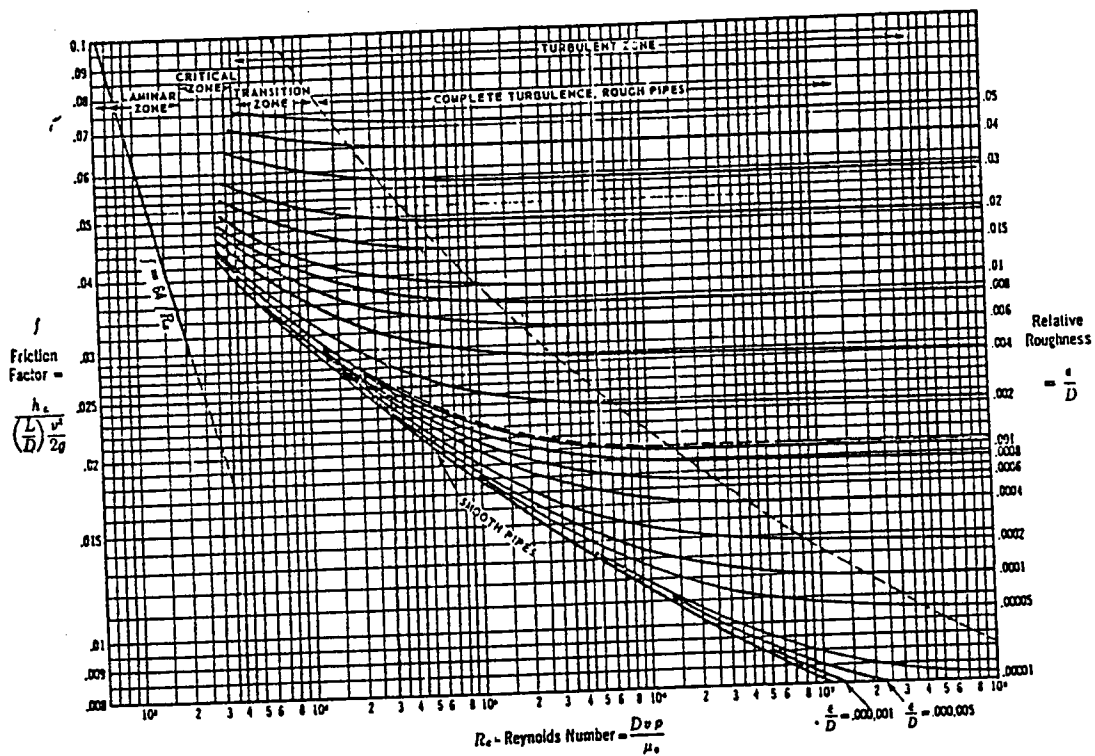
Description of Product				Equivalent Length in Pipe Diameters (L/D)
Globe Valves	Stem Perpendicular to Run	With no obstruction in flat, bevel, or plug type seat	Fully open	340
		With wing or pin guided disc	Fully open	450
	Y-Pattern	(No obstruction in flat, bevel, or plug type seat)		
		- With stem 60 degrees from run of pipe line	Fully open	175
Angle Valves		- With stem 45 degrees from run of pipe line	Fully open	145
		With no obstruction in flat, bevel, or plug type seat	Fully open	145
		With wing or pin guided disc	Fully open	200
Gate Valves	Wedge, Disc, Double Disc, or Plug Type		Fully open	13
			Three-quarters open	35
			One-half open	160
			One-quarter open	900
	Pulp Stock		Fully open	17
			Three-quarters open	50
			One-half open	260
			One-quarter open	1200
	Conduit Pipe Line		Fully open	3"
	Check Valves	Conventional Swing	0.5 ft... Fully open	135
Clearway Swing		0.5 ft... Fully open	50	
Globe Lift or Stop; Stem Perpendicular to Run or Y-Pattern		2.0 ft... Fully open	Same as Globe	
Angle Lift or Stop		2.0 ft... Fully open	Same as Angle	
In-Line Ball		2.5 vertical and 0.25 horizontal... Fully open	150	
Foot Valves with Strainer	With poppet lift-type disc	0.3 ft... Fully open	420	
	With leather-hinged disc	0.4 ft... Fully open	75	
Butterfly Valves (8-inch and larger)			Fully open	40
Cocks	Straight-Through	Rectangular plug port area equal to 100% of pipe area	Fully open	18
	Three-Way	Rectangular plug port area equal to 80% of pipe area (fully open)	Flow straight through	44
			Flow through branch	140
Fittings	90 Degree Standard Elbow			30
	45 Degree Standard Elbow			16
	90 Degree Long Radius Elbow			20
	90 Degree Street Elbow			60
	45 Degree Street Elbow			26
	Square Corner Elbow			57
	Standard Tee	With flow through run		20
		With flow through branch		60
	Close Pattern Return Bend			50
Pipe	90 Degree Pipe Bends			See Page A-27
	Miter Bends			See Page A-27
	Sudden Enlargements and Contractions			See Page A-26
	Entrance and Exit Losses			See Page A-26

*Exact equivalent length is equal to the length between flange faces or welding ends.

†Minimum calculated pressure drop (psi) across valve to provide sufficient flow to lift disc fully.

‡For limitations, see page 2-11. For effect of end connections, see page 2-10.

For resistance factor "K", equivalent length in feet of pipe, and equivalent flow coefficient "Cv", see pages A-31 and A-32.



Problem:
Determine the friction factor for 10-inch cast iron pipe (10.16" I.D.) at a Reynolds number flow of 30,000.
Solution: The relative roughness (see page A-13) is 0.001. Then, the friction factor (f) equals 0.016.

For other forms of the f , equation, see page 3-2.
Data extracted from *Friction Factors for Pipe Flow* by L. F. Mc
with permission of the publisher, The American Society of Me-
chanical Engineers, 29 West 39th Street, New York 18, N. Y.

Figure 2.4. Friction factor for any type of commercial pipe [10].

as the flow of water at 60 °F (15.56 °C), in gallons per minute, at a pressure drop of one lb/in² (6.895 kpa) across the valve. So, the relationship between k and C_v is given by the following equation

$$C_v = 29.9 \frac{d^2}{\sqrt{k}} \quad (2.6)$$

C_v can be found from Figure 2.5 if the valve resistance coefficient k and the friction factor f are known.

For turbulent flow, the flow through the valve in gallons of water per minute at 60 °F can be determined from

$$Q = 7.9 \times C_v \times \sqrt{\frac{\Delta P}{\rho}} \quad (2.7)$$

where ρ is the water density. For different flow rates the pressure drop across the valve can be computed from the same formula arranged as follows

$$\Delta P = \frac{\rho}{62.4} \left(\frac{Q}{C_v} \right)^2 \quad (2.8)$$

The head loss and pressure drop with laminar flow ($2000 \geq Re \geq 1000$) is

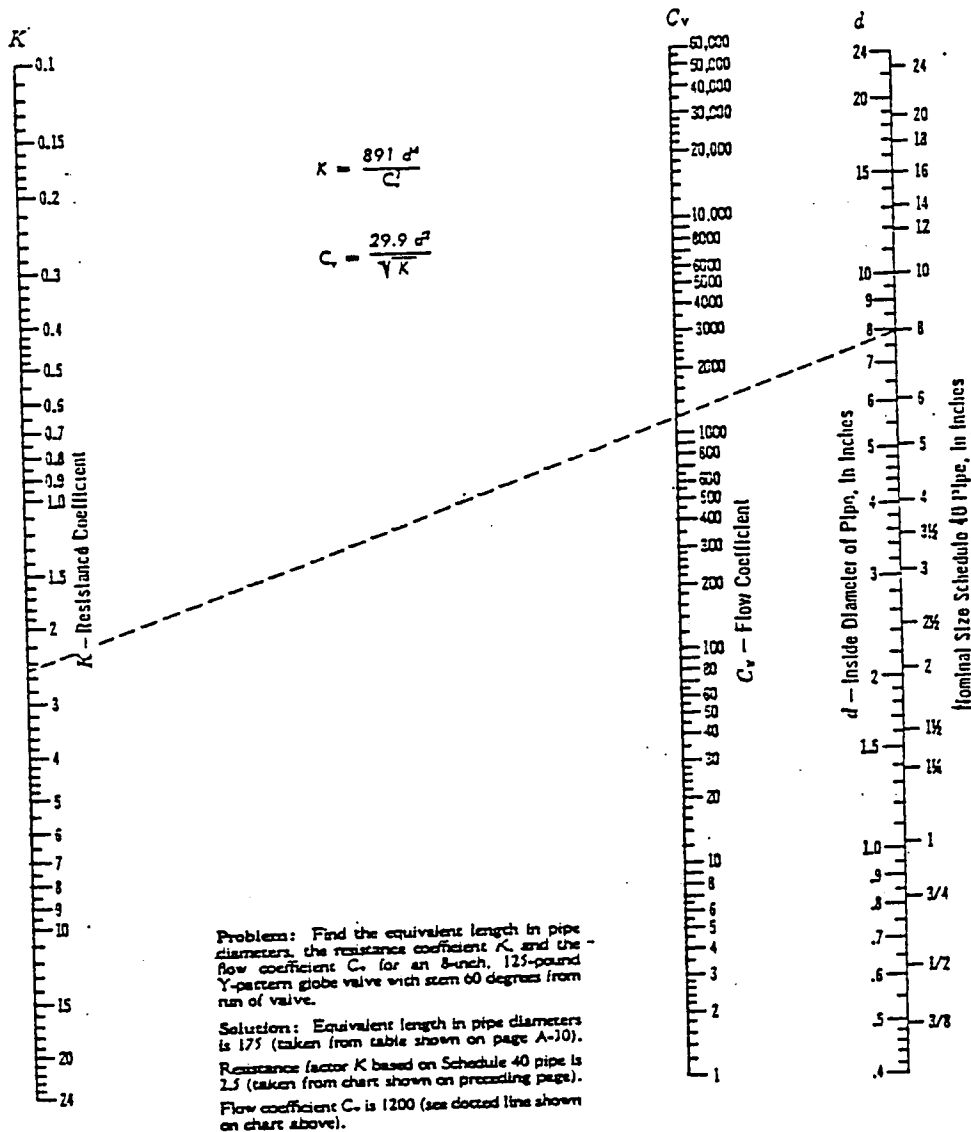
$$\Delta P = 2.28 \times 10^{-5} \left(\frac{L}{D} \right) \frac{\mu Q}{d^3} \quad (2.9)$$

If $Re \leq 1000$, an equivalent length correction factor is required

$$\left(\frac{L}{D} \right)_s = \frac{Re}{1000} \left(\frac{L}{D} \right)_r \quad (2.10)$$

For the test valve, when the valve is fully open, the value of C_v is equal to 8.0.

***Equivalents of Resistance Coefficient K
And Flow Coefficient C_v**



*For limitations, see page 2-17.

Figure 2.5. Equivalents of resistance coefficient K and flow coefficient C_v , [10,13,14,15].

The relationship between the stem position and the flow rate is shown in Figure 2.6. An exponential expression of the flow rate vs. stem position is given based on the data from Figure 2.6.

$$Q = 5.614(1 - e^{-0.03956 \times L}) \text{ (gal/min)}. \quad (2.11)$$

where L is the fraction of stem travel. This was established through a mathematical fit to the data.

The pressure drop for turbulent flow across the valve is given as a function of the stem travel. For turbulent flow

$$\Delta P = 0.5051 \times \rho \frac{(1 - e^{-0.03956 \times L})^2}{C_v^2} \quad (2.12)$$

For laminar flow

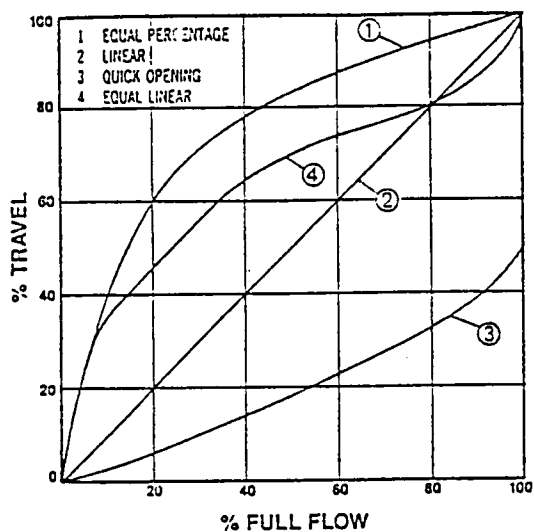
$$\Delta P = 1.28 \times 10^{-4} \left(\frac{L}{D} \right) \frac{\mu (1 - e^{-0.03956 \times L})}{d^3} \quad (2.13)$$

where the equivalent length correction factor is calculated using Equation (2.10) if R_e is less than 1000, and C_v changes as a function of stem travel. Both Equations (2.12) and (2.13) show that the pressure drop across the valve is a function of the stem travel, C_v , and flow properties.

2.2.2 Centrifugal Recycling Pump [16,17]

The centrifugal recycling pump used in the test loop is driven by a 1/2 HP AC motor. The principle of the centrifugal pump is shown in Figure 2.7. Flow enters at

FLOW CHARACTERISTIC: Equal-percentage, linear, quick-opening or equal/linear combination.



RANGEABILITY: 60:1 with Equal Percentage or Equal/linear Trim
30:1 with Linear Trim

VALVE CAPACITY:

Plug Style	C _v per Valve Seat Size*				
	1/2"	3/4"	1"	1-1/2"	2"
Equal Percentage	4.4	8.0	11.5	30	45
Linear	4.4	8.0	12.2	30	45
Equal/Linear	4.4	8.0	12.2	35	50
Quick-Opening	4.6	9.0	12.7	40	60

Low Flow Trim Available

Figure 2.6. Flow characteristic and C_v per valve seat size [9].

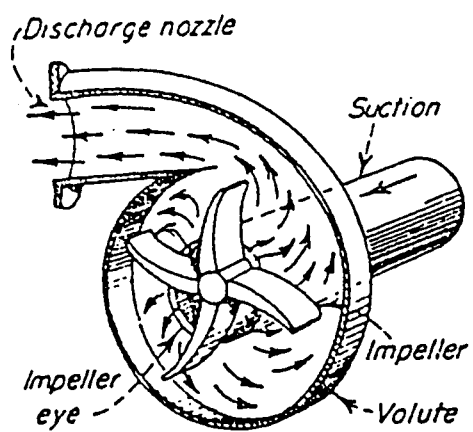


Figure 2.7. Principle of a centrifugal pump [16,17].

the center of a rotating impeller, is accelerated to high velocity, and is thrown by the vanes radially outward into a passage which converts most of the velocity energy into additional pressure. The conversion between the velocity and the pressure is related to the system flow path resistance. When the valve stem position changes, the system resistance is changed, so the flow rate of the pump is also changed. When the valve is fully open, the pump can provide the system a flow rate of 5.6 gallons per minute. When the valve is totally closed, the highest pressure head is 40 feet of water or 17.3 psig. The specification and performance of the test loop pump is shown in Figure 2.8.

2.2.3 Heat Exchanger

When the system is used for a period of time, the temperature will increase due to flow friction. A heat exchanger is installed in the test loop to keep the system at a constant temperature. The system flow goes through the tube side of the exchanger and the cooling water flows in the shell side.

2.2.4 Pressurizer

A pressurizer is connected to the test loop to maintain the system at a constant pressure (around 40 psig) with a small range of fluctuation. The bottom of the pressurizer is connected to the test loop. A constant pressure is supplied to the top of the pressurizer using a nitrogen cylinder. About 60 percent of the total

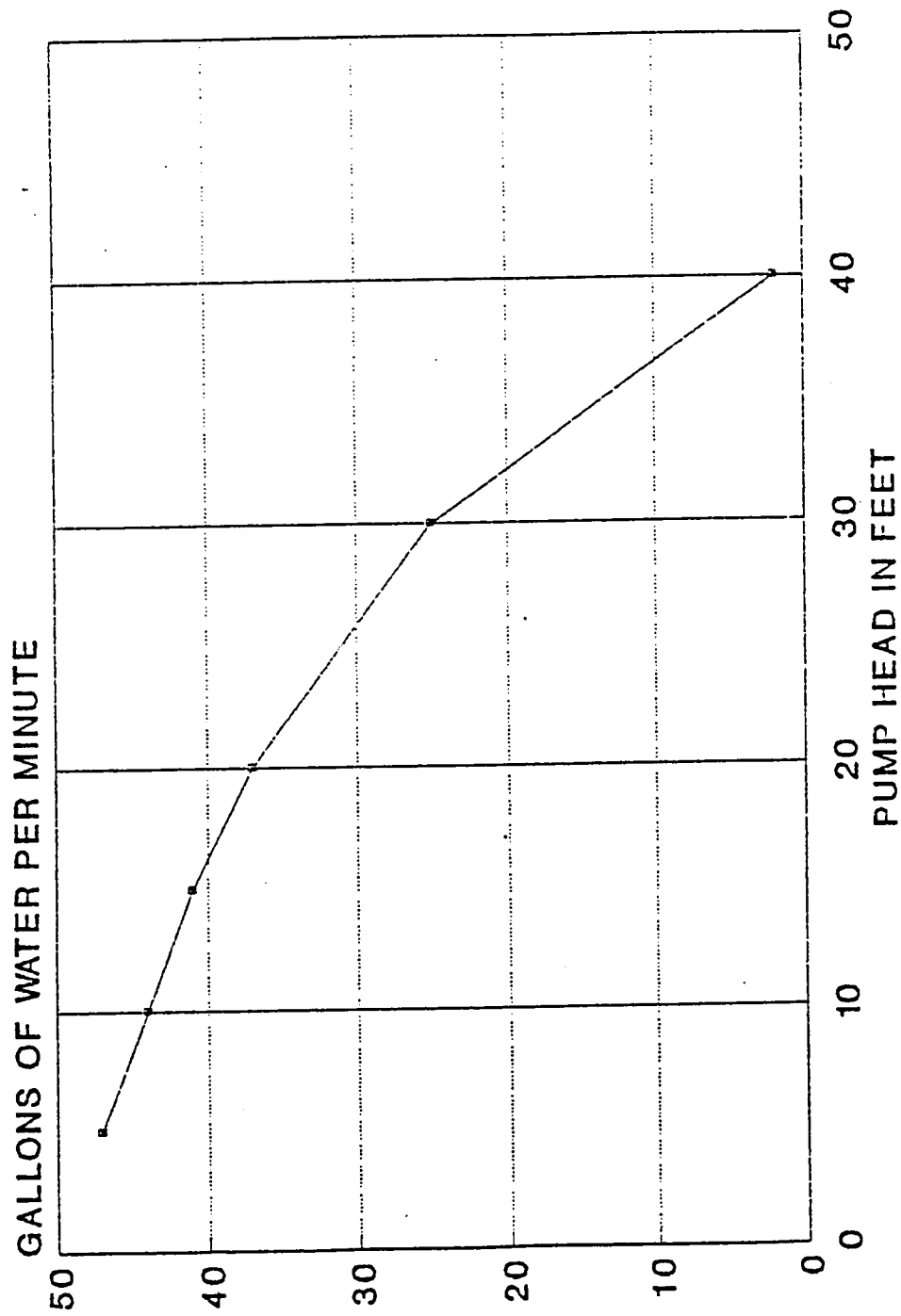


Figure 2.8. Pump specification and performance as operated in the test loop.

volume is occupied by water. The diameter of the pressurizer is about 30 cm and the height is about 45 cm. A level indicator is also connected to the pressurizer.

2.2.5 Current-to-Pressure (I/P) Convertor [18]

The I/P (current-to-pressure) convertor is used to control the pressure applied to the actuator diaphragm by adjusting its electric current input. A schematic of the I/P convertor is given in Figure 2.9. When an electric current passes through the coil of the I/P transducer, a magnetic field is created which acts on the magnet assembly, causing the anvil to which it is attached to move away from or towards the nozzle depending on the direction of the current flow. Motion at the nozzle assembly is proportional to the amount of current passing through the coil. If the anvil moves away from the nozzle, reduced pressure on top of the diaphragm assembly causes it to move in the upward direction and the supply valve closes. If the anvil moves towards the nozzle, increased pressure on top of the diaphragm assembly causes it to move in the downward direction and the supply valve opens. So, the supply valve will throttle to keep the downstream pressure at a value proportional to the current flowing through the coil.

2.3 Test Loop Instrumentation

The following measurements were used in order to monitor the valve operation: 1) pressure on the actuator diaphragm, 2) valve stem position, 3) pressure

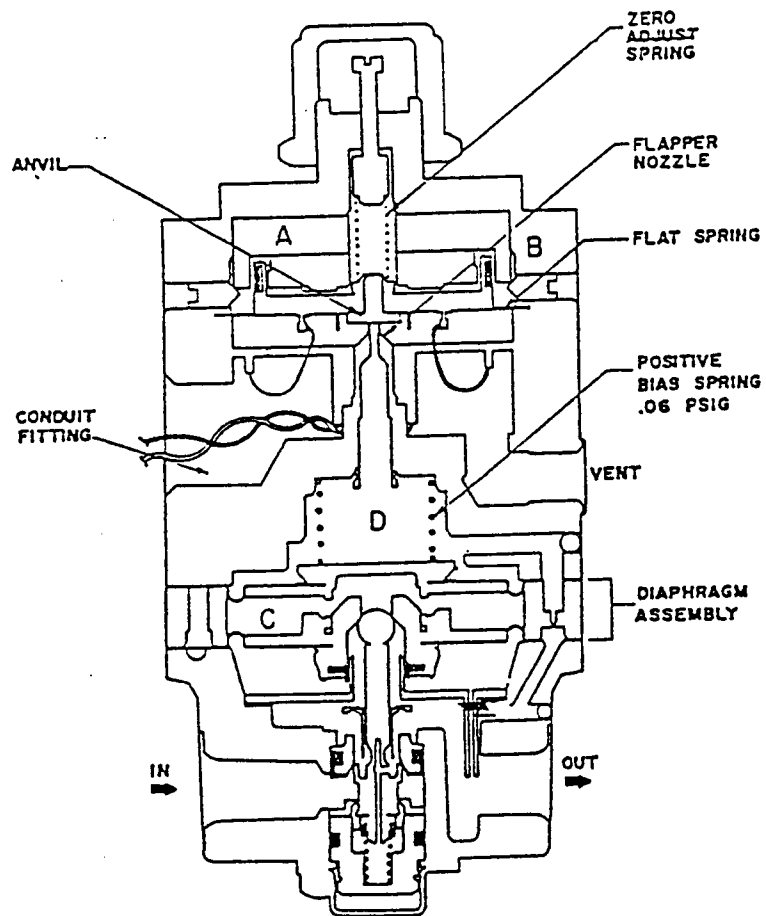


Figure 2.9. Schematic of I/P (current-to-pressure) convertor [18].

drop across the valve, and 4) water flow rate in the test loop. The test valve loop instrument includes a linear variable differential transformer (LVDT) to measure stem motion, absolute and differential pressure transducers, and a turbine flow rate meter.

2.3.1 Absolute and Differential Pressure Gauges [19]

Both absolute pressure gauges and differential pressure gauges were used in the AOV test measurements. An absolute pressure gauge was used to measure the air line pressure on the actuator diaphragm. The differential pressure transducers measure the pressure difference both across the test valve and the system line pressure drops. These pressure gauges are very precise and give reliable indication. The differential pressure transmitters have a variable capacitance sensing element, called the delta cell (Figure 2.10). Differential capacitance between the sensing diaphragm and the sensing plates is converted into an electrical signal. Process pressure is transmitted through an isolation diaphragm and silicon oil-fill fluid to a sensing diaphragm in the center of the delta cell. The position of the sensing diaphragm is detected by the capacitance plates on both sides of the sensing diaphragm. The sensor is driven by an oscillator. The signal is then rectified through a demodulator to obtain the DC signal using a diode bridge. The DC current through the transformer winding is directly proportional to the system pressure. A schematic of the circuit diagram is given in Figure 2.11. For absolute pressure transmitters, only one capacitor is used, however, both pressure gauges work on the

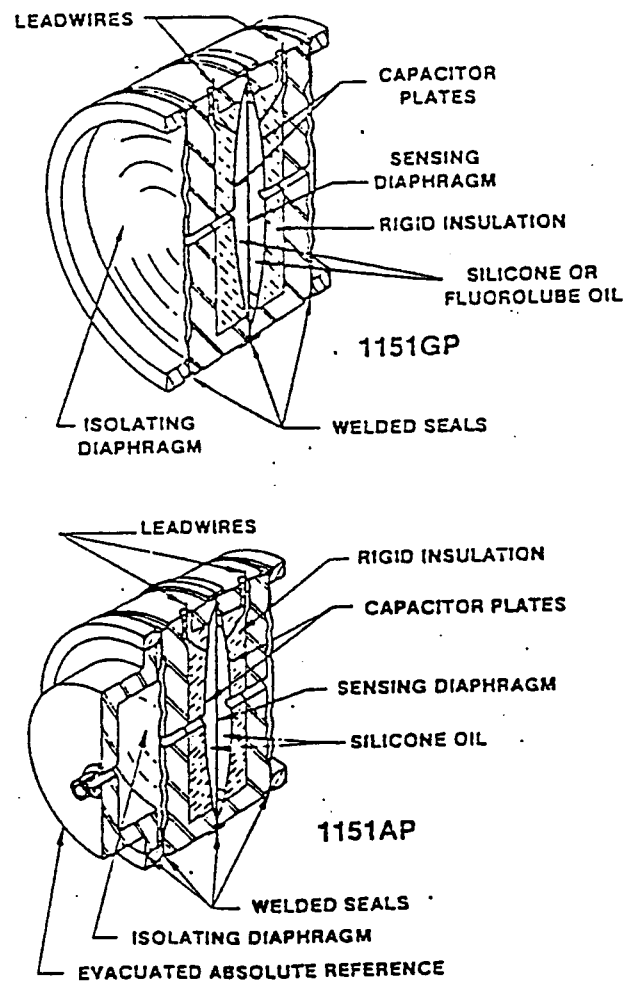


Figure 2.10. The Delta cell of a Rosemount pressure gauge [19].

same principle.

2.3.2 Linear Variable Differential Transformer [20]

The linear variable differential transformer (LVDT) consists of a core and a coil. The coil is mounted on top of the upper cover of the valve actuator. The core of LVDT is directly connected to the valve stem. The LVDT is used to measure the valve stem travel. The transducer produces an electrical output which is proportional to the displacement of the movable core. The electrical circuit consists of a primary coil and two secondary coils symmetrically spaced on a cylindrical form sensing the core movement. A free moving rod-shaped magnetic core inside the coil assembly provides a path for the magnetic flux linking the coils. A cross section of the LVDT and a plot of its operating characteristics are shown in Figures 2.12 a) and 2.12 b), respectively. Figure 2.13 shows the dimensions of LVDT used in this project. When the primary coil is energized by an external AC source, voltages are induced in the two secondary coils. These are connected series opposing so the two voltages are of opposite polarity. Therefore, the net output of the transducer is the difference between these voltages, which is zero when the core is at the center or null position. When the core is moved from the null position, the induced voltage in the coil toward which the core is moved increases, while the induced voltage output in the opposite coil decrease. This action produces a difference voltage output that varies linearly with the changes in the core position. Further details of the LVDT are given in Appendix C.

ELECTRICAL BLOCK DIAGRAM

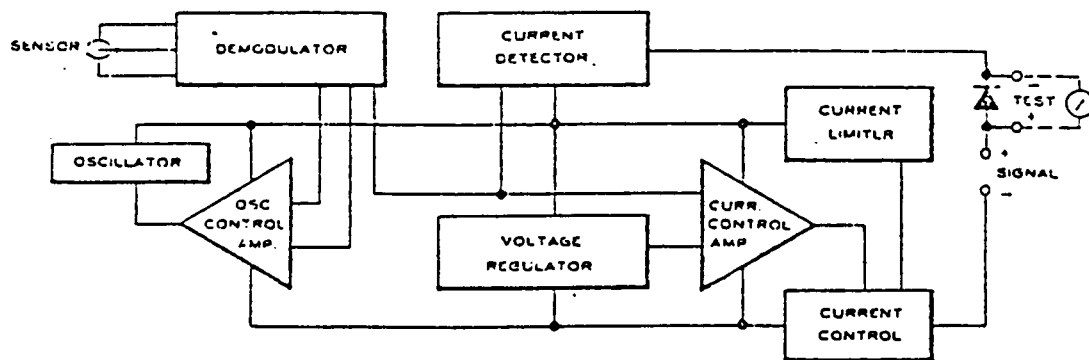
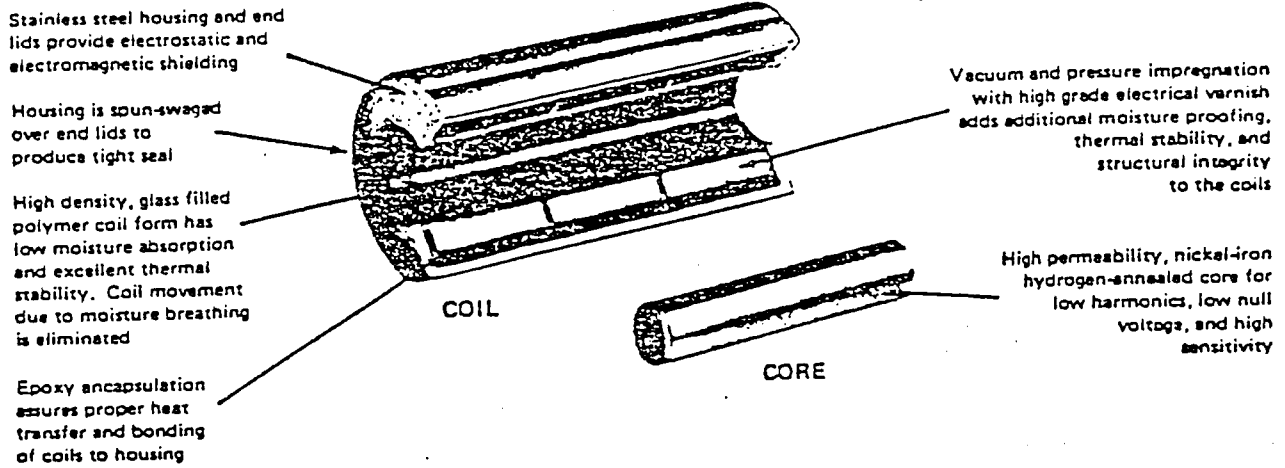
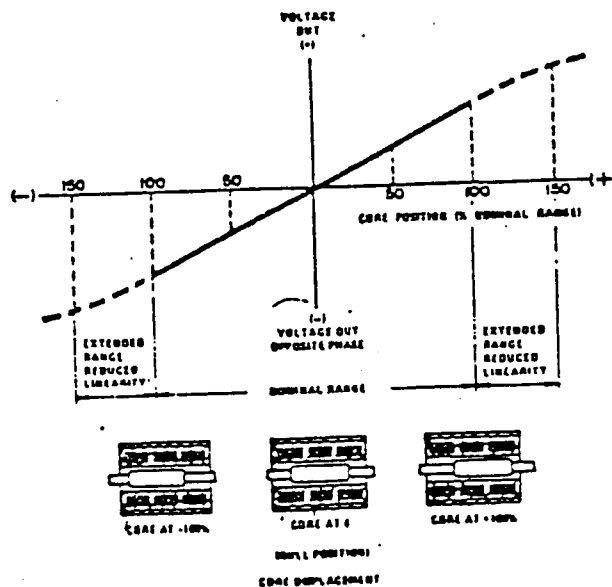


Figure 2.11. Schematic of a Rosemount pressure gauge circuit [19].



a) Cross section of the linear variable differential transformer (LVDT) [20].



b) Plot of a typical LVDT operating characteristics [20].

Figure 2.12. Cross section and operating characteristics of the linear variable differential transformer

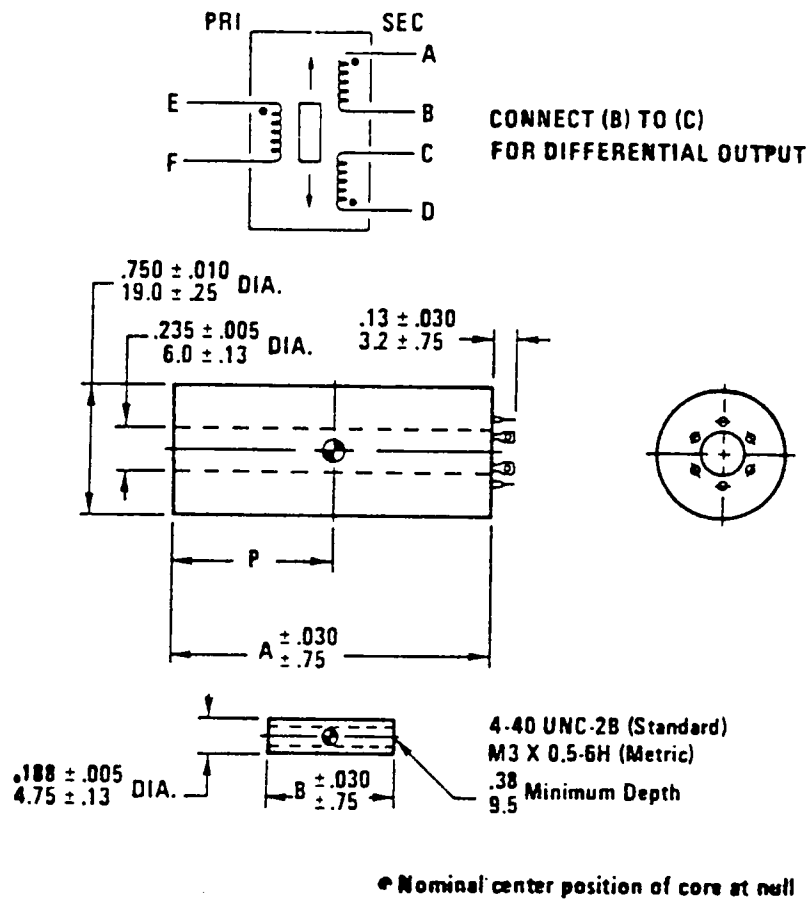


Figure 2.13. Dimensions of LVDT used in this project

2.3.3 Flow Rate Meter [21,22]

The system flow rate is measured using a turbine flow rate meter. The meter consists of a blade rotor, a pick-off, and a meter housing. The rotation of freely suspended blade rotor is precisely proportional to the volume of fluid passing through the transducer. A pick-off is attached to the flow rate meter housing, close to the turning rotor. When the flow passes through the blades of the turbine rotor, the rotor produces an impulse, which is collected and is presented as a frequency of rotation. Each pulse represents a discrete volume of fluid. The output rotation frequency is linearly proportional to the flow rate. A schematic of the turbine flow rate meter is given in Figure 2.14.

2.4 Calibration of Loop Measurements

Calibration of these instruments is necessary to ensure the accuracy of measurements and conversion to engineering units. Various instrumentation calibration graphs are shown in Figure 2.15 through Figure 2.18.

Figure 2.15 shows the relationship between I/P convertor voltage input and the pressure output. When the input voltage to the I/P convertor varies from negative to positive ten volts, the output pressure varies from 3 to 18 psig. The relationship between the voltage and pressure is almost linear except at the lower pressure range (less than 5 psig).

The absolute pressure gauge provides a voltage readout. The actual pressure

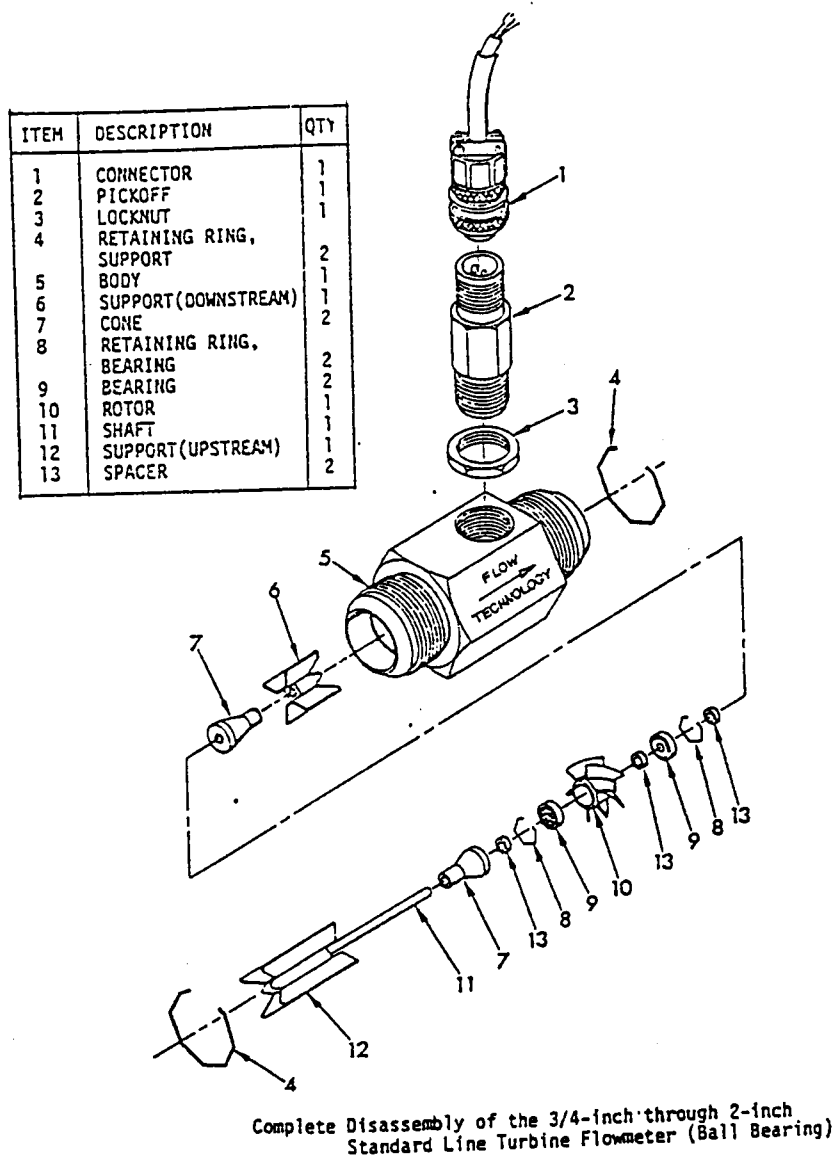


Figure 2.14. Schematic of the turbine flow rate meter [21,22].

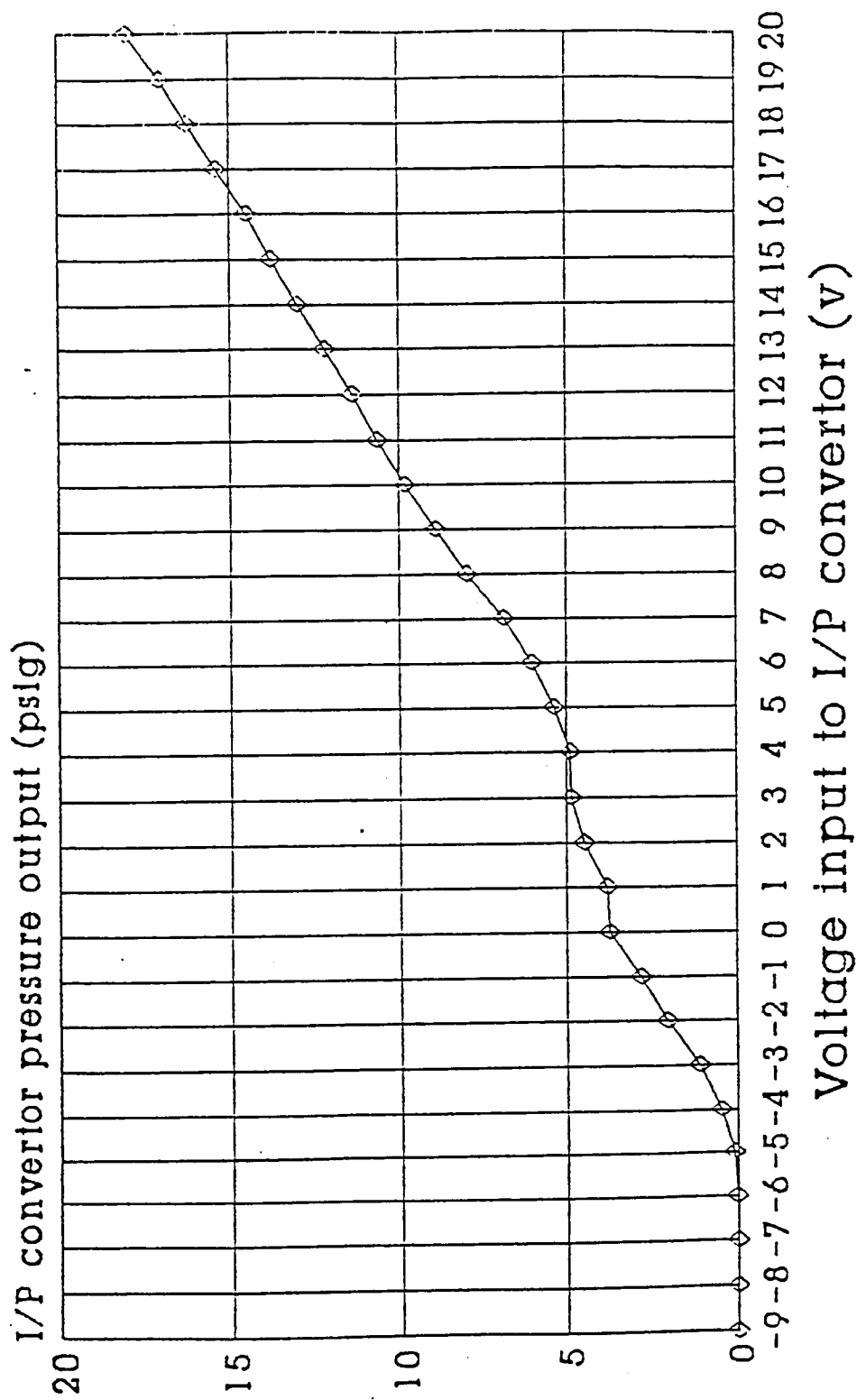


Figure 2.15. Relationship between I/P convertor voltage input and the pressure output.

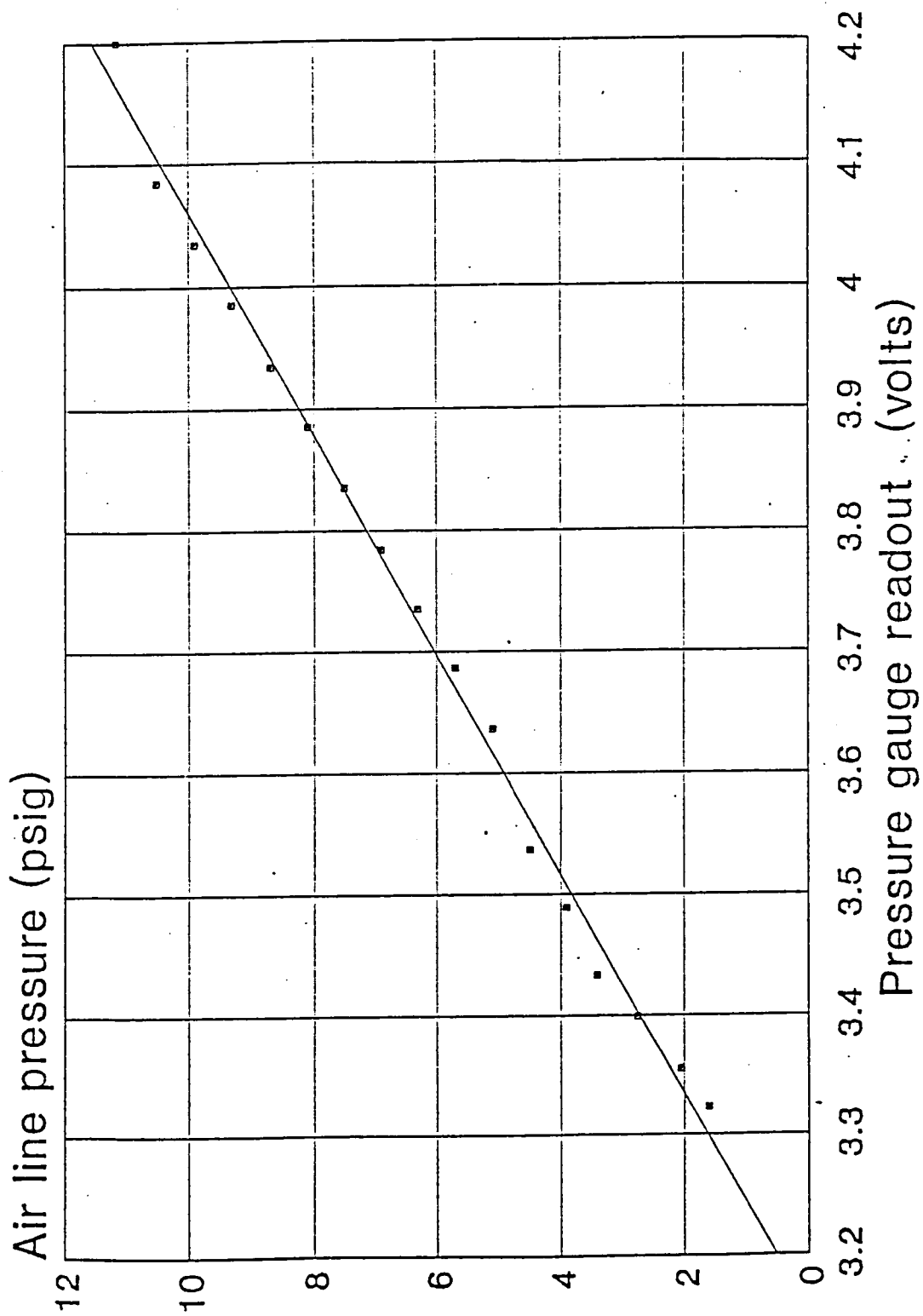


Figure 2.16. Actual pressure versus the voltage readout of the absolute pressure gauge.

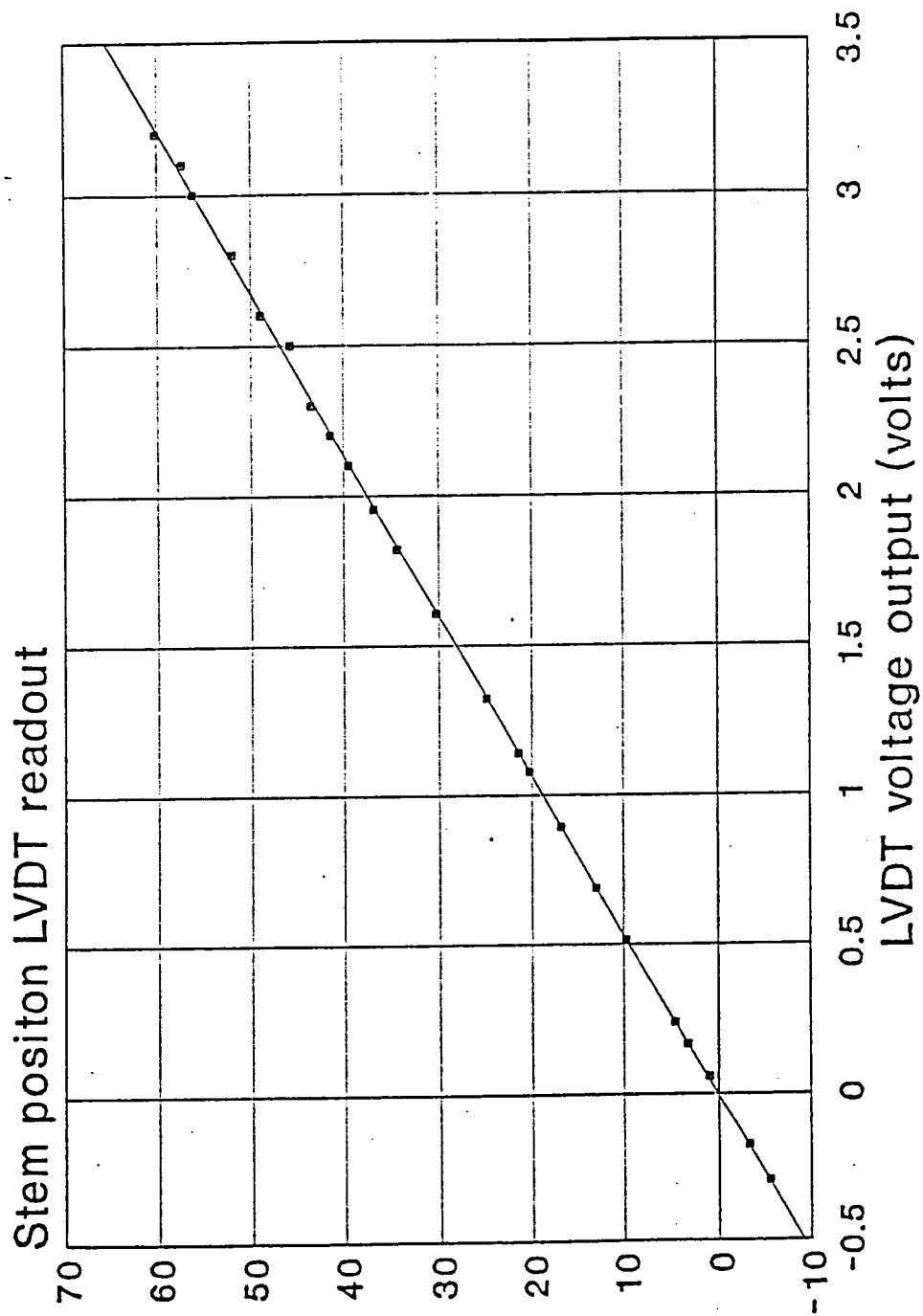


Figure 2.17. Linear variable differential transformer (LVDT) calibration data.

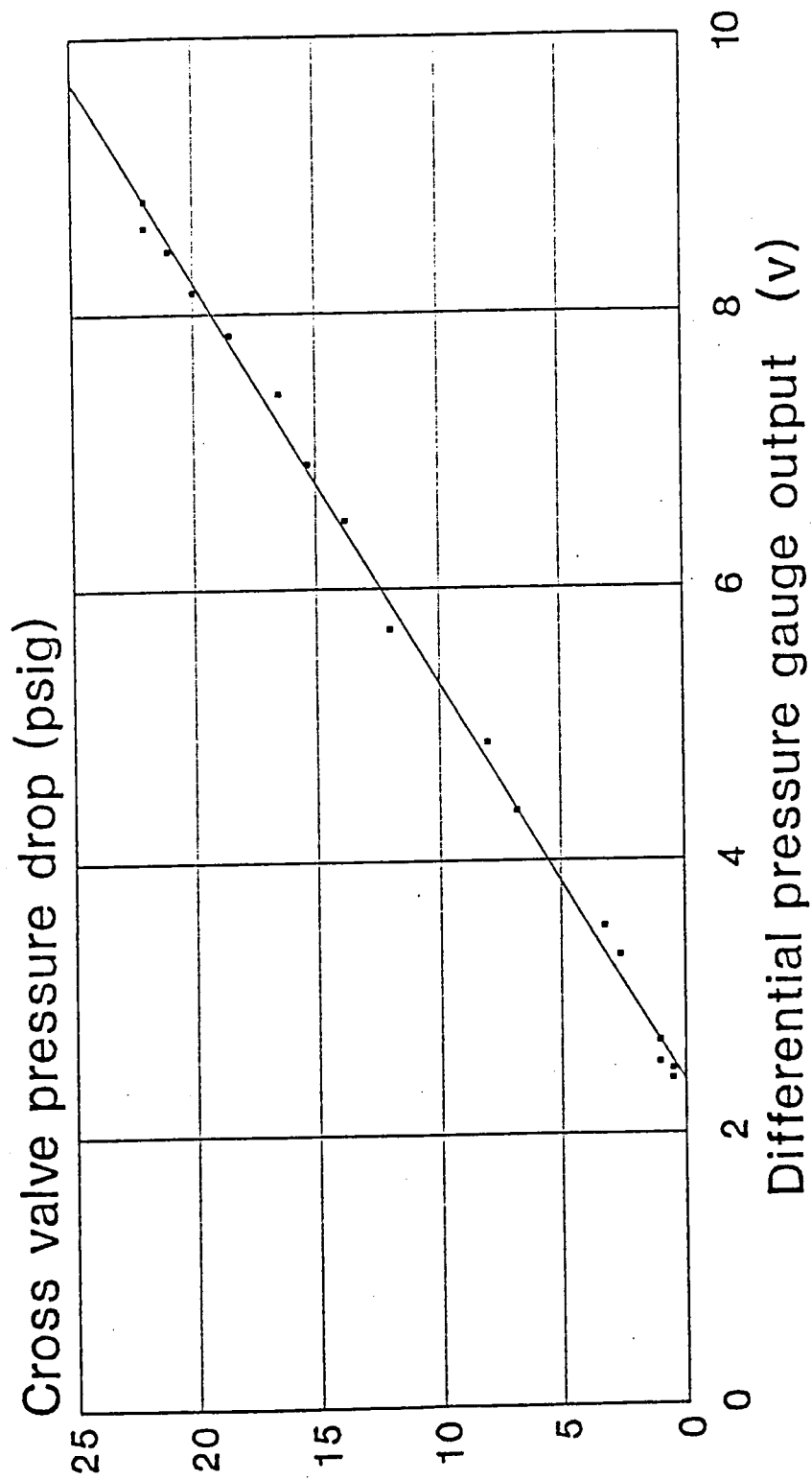


Figure 2.18. Pressure difference across the valve versus gauge voltage readout.

versus the voltage readout is shown in Figure 2.16. The voltage readout is linearly proportional to the pressure in the gauge delta cell. The voltage output of the gauge ranges between 3.2 and 4.2 volts corresponding to pressure changes from 3 psig to 12 psig.

The linear variable differential transformer (LVDT) calibration data is shown in Figure 2.17. The position of the LVDT core is linearly proportional to the voltage output. The digital readout of its position indicator and the voltage output vary from -10 to 65, and -0.5 to 3.5 volts, respectively. The power supply readout can be adjusted to a zero reading when the valve stem is either in the open or closed position.

Figure 2.18 shows the pressure difference across the valve versus gauge voltage readout. A linear relationship between the pressure drop across the valve and the gauge readout is generated using this data. When the actuator position varies from fully open to fully closed, the pressure difference changes from one psig to a maximum value of 27 psig. The voltage output of the gauge is from 2.4 volts to about 9 volts when the actuator position varies from fully open to fully closed.

The calibration data of the turbine flow rate meter is plotted in Figure 2.19. The rotational frequency is linearly proportional to the flow rate and is expressed as [20]

$$\text{Flow rate (Gal/min)} = 2.8071 \times 10^{-2} F \quad (2.14)$$

where F is the blade rotation frequency of the rotor in Hz.

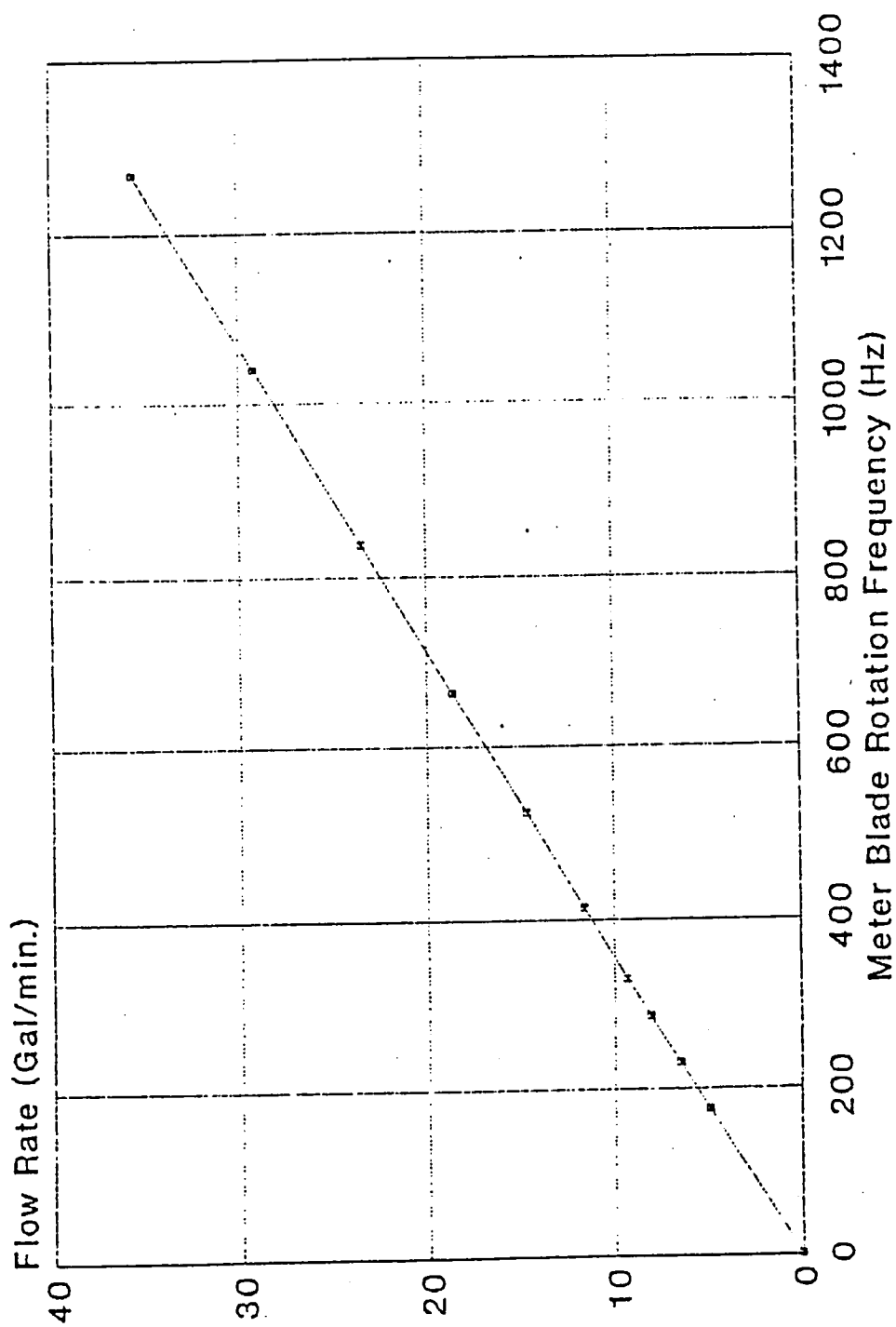


Figure 2.19. Calibration data of the turbine flow rate meter.

2.5 Summary

The calibration of the measurements was thoroughly checked to ensure their accuracy. If the signals sampled by the computer have to be transferred to experimental units, these conversion factors can be used easily to convert the voltage readouts to actual experimental values.

CHAPTER III

ANALYSIS OF TRANSIENT BEHAVIOR OF AIR-OPERATED VALVES (AOVs)

3.1 Introduction

The performance of installed AOVs is monitored periodically by examining valve stroke time, stem seating position, and air line pressure. But a slowly developing anomaly may not be easily detected.

In this research, a detailed transient analysis of the test valve was conducted to identify measurements that are sensitive to valve anomalies and to provide guideline for on-line transient condition monitoring techniques for AOVs. Transient analysis refers to actuation of the valve with different pressure perturbations. The valve was actuated by varying the air line pressure, and the corresponding responses were measured. Air line leakage and friction increase were introduced to study the responses of the system to the anomalies. The following tests were conducted at normal operating conditions and with introduced anomalies:

1. Response to step change in the air line pressure.
2. Response to ramp change in the air line pressure.
3. Response to sinusoidal change in the air line pressure.

3.2 Data Acquisition System

The interface between the measurements and the personal computer is illustrated in Figure 3.1. The instrument outputs are connected to the PC using a patch panel. The measurements are air line pressure, stem motion, and valve differential pressure. Many pressure perturbations may be employed for valve actuation. In this research step pressure input, ramp pressure input, and sinusoidal pressure input were used.

The data acquisition and some of the analysis were performed using the Multivariate Autoregression modeling for System Surveillance and Analysis (MARSSR) software system [23]. The plots were made using PC-MATLAB [25].

3.3 Transient Analysis

Due to the test valve dynamics, the responses to different inputs vary during the transient analysis (open-to-close and close-to-open strokes). This difference could be used to study the performance of air-operated valves.

The valve transient analyses were also conducted by introducing certain anomalies to the valve. The introduced anomalies were diaphragm and air line leakage and changes in stem friction. Problems such as broken air line, lost bolt on the diaphragm fastener, and pressure variations are considered as part of the air leakage anomaly. Bent stem and binding of the stem and its packing were simulated as a change in the friction force. The problem of valve seating was not simulated due

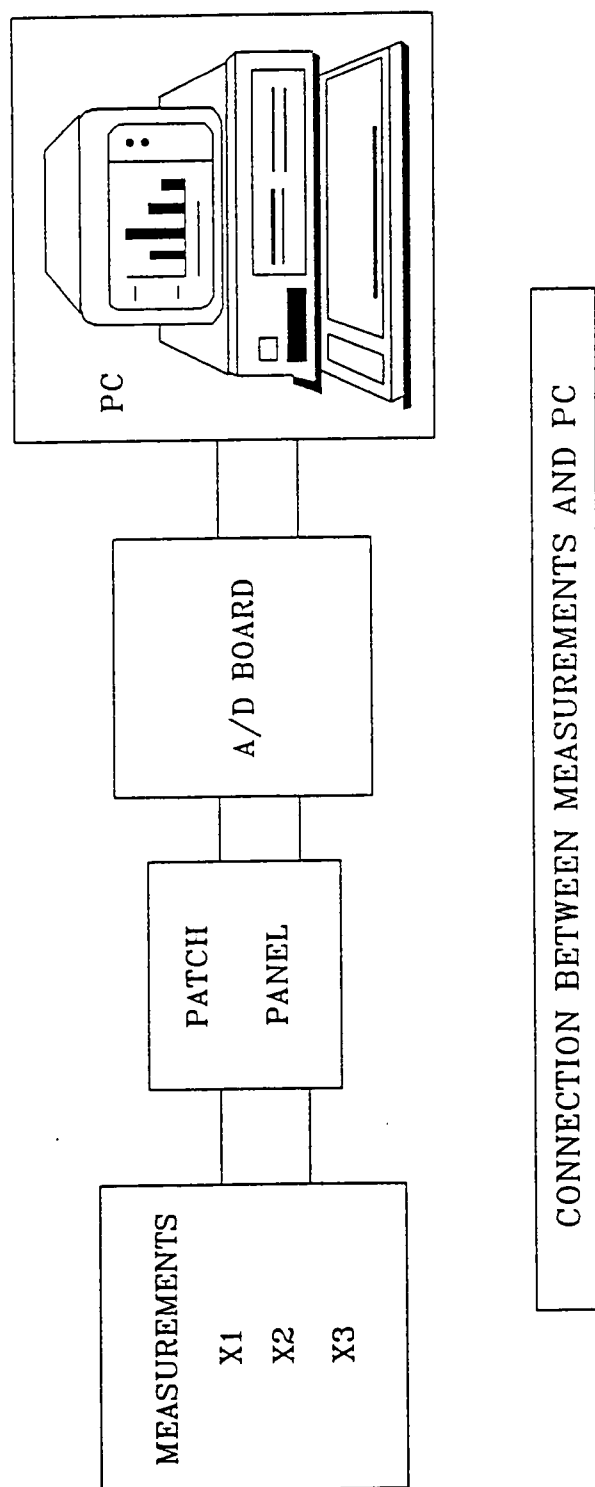


Figure 3.1. Interface between the measurements (sensors) and the data acquisition PC.

to the difficulty of introducing this anomaly.

The first anomaly studied using this approach was diaphragm and air line leakage. The anomaly was simulated by partly disconnecting the air supply line after the I/P convertor. The amount of leakage was limited so that a full valve stroke could be completed.

The other introduced anomaly was the stem friction variation. This anomaly was simulated by holding the valve stem with a clamp. The holding forces were carefully adjusted so that a full stroke of the test valve can be performed.

3.3.1 Response to Step Changes in Air Line Pressure

The step perturbation to the valve was introduced by changing the electric current input to I/P convertor from a positive to a negative voltage. When the current input to I/P convertor changes suddenly, the pressure output of the I/P convertor, corresponding to this current changes approximately like a step variation, thus applying a step pressure input on the diaphragm.

3.3.1.1 Response to Step Input at Normal Operation

The input and response during normal operation are plotted in Figure 3.2. The range of the current input to I/P convertor is from -4.5 volts to +4.5 volts and the pressure in the air supply line varies from 3 to 15.5 psig. Based on this pressure range, a valve close-open stroke is completed. It can be seen that there is very little

time delay between the voltage input and the pressure gauge readouts.

The LVDT measurement shows the valve stem position. As the valve approaches either open or close position, the response of the valve stem becomes sluggish or slow. When the valve is close to its open position, the resistance of the spring force reaches its maximum so that the movement of the stem is more restrained. On the other hand, as the valve approaches its close position, the flow dynamic forces dominate and the spring force effect becomes smaller, and the movement of the stem is slowed.

Figure 3.2 also shows the response of the valve differential pressure output to this step input. The pressure drop across the valve is directly related to the valve stem position. This relationship is also verified using a mathematical model of AOV dynamics. This pressure drop across the valve is plotted for a full stroke of the test valve.

The relationships between the measurements are plotted in Figures 3.3 through 3.5. The air line pressure vs. LVDT position is shown in Figure 3.3. The time delay in the valve stroke can be clearly seen (Figure 3.3). The slopes and the graphs of the plots for close-to-open and open-to-close strokes are different because of the valve dynamics. A hysteresis effect is seen in this plot. The air pressure was applied to the valve during the open-to-close stroke and was simply released during the close-to-open stroke. The full stroke is completed as the air line pressure varies from 3.7 to 4.2 volts which corresponds to the stem position varying from 1.35 to 3.65 volts. When the air line pressure was applied, the stem does not move quickly.

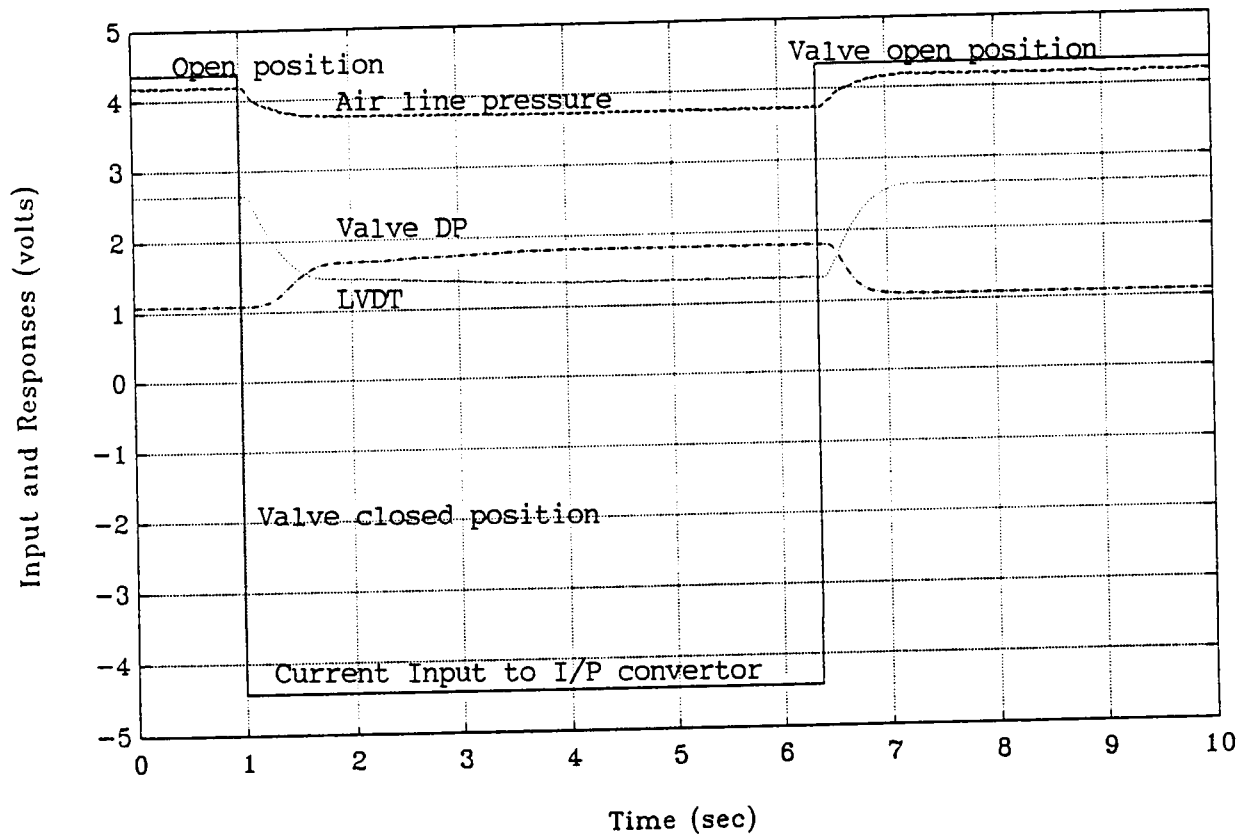


Figure 3.2. Step input and the responses of valve system variables.

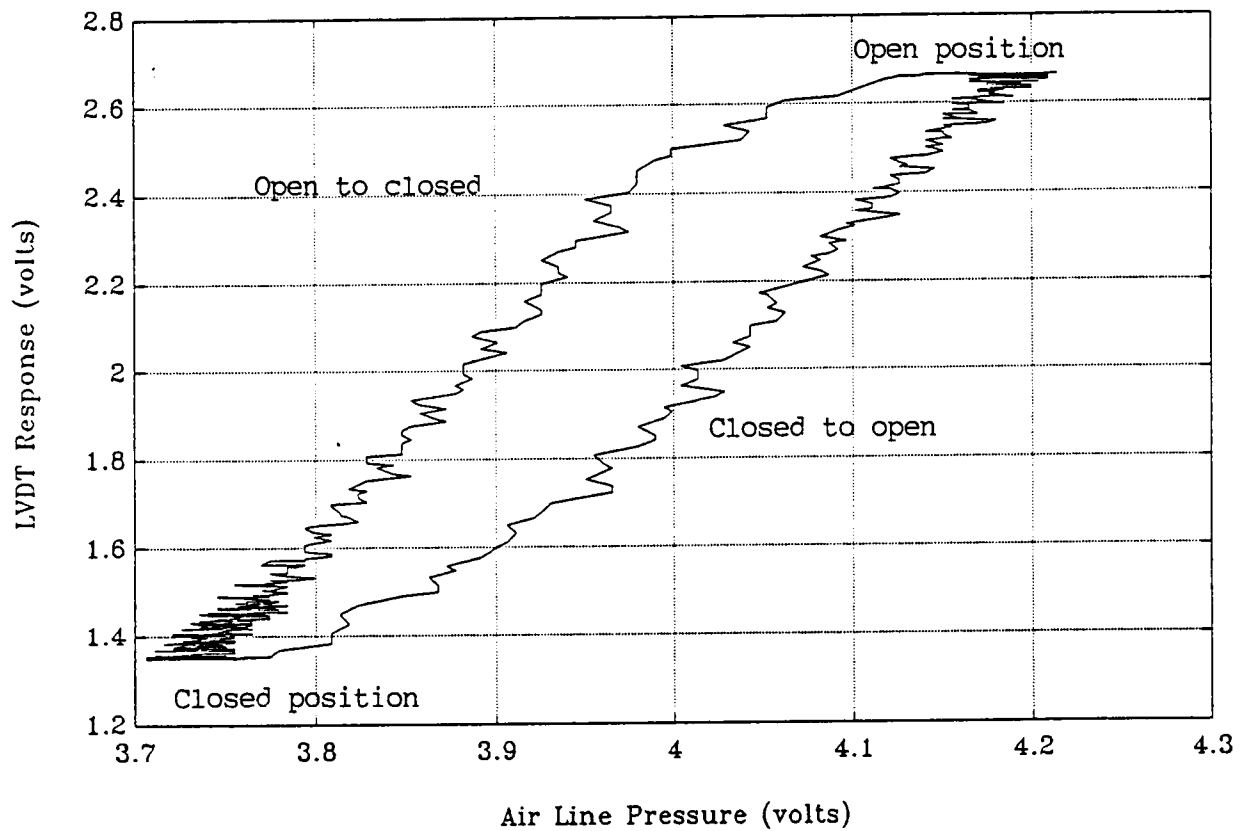


Figure 3.3. Air line pressure vs. LVDT for a step input response.

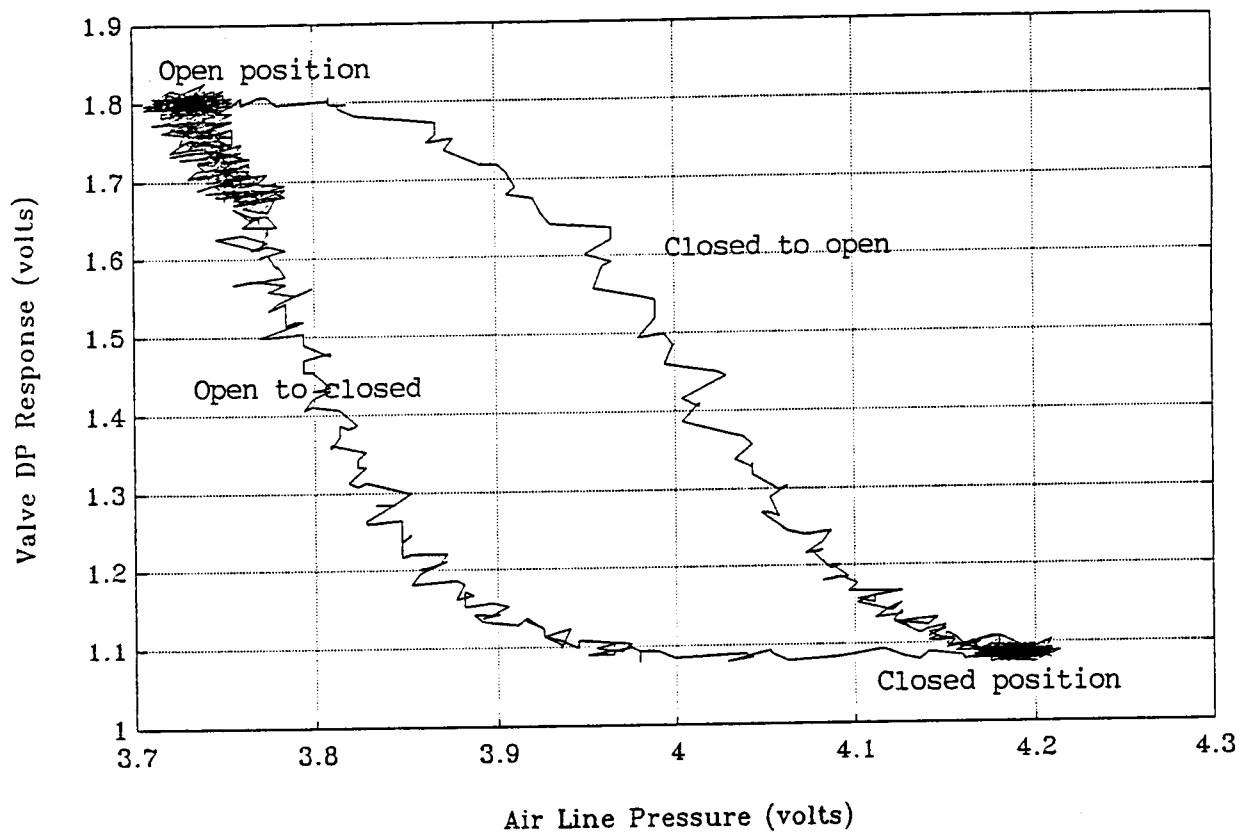


Figure 3.4. Air line pressure vs. valve differential pressure for a step response.

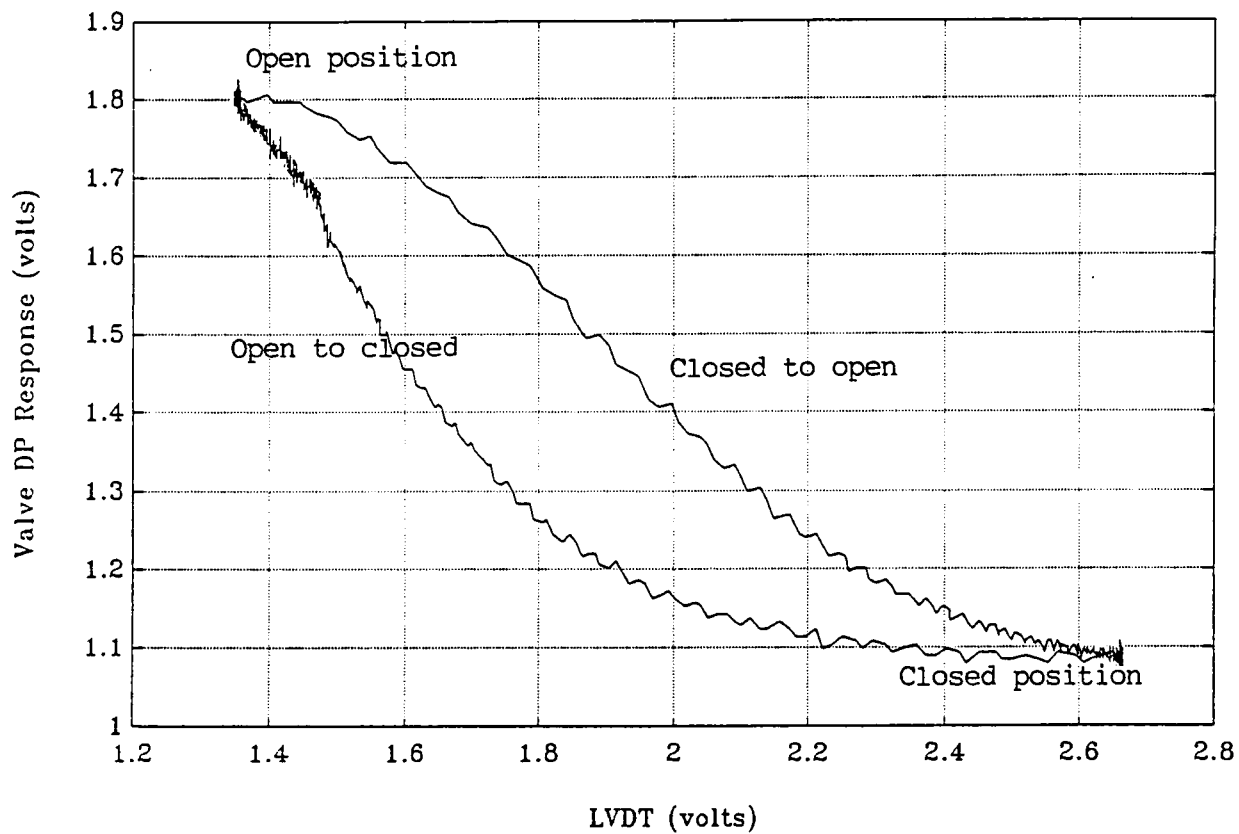


Figure 3.5. Valve differential pressure versus stem position (LVDT) for a step response.

A time delay is seen in the response. During the open-to-close stroke the pressure was first released and then the stem starts to move. Some discontinuity is noticed in the plots due to air line pressure fluctuations and stem travel variations. The slopes and the discontinuity are useful parameters and can be used to predict valve operating conditions.

Figure 3.4 shows the relationship between the air line pressure and the valve differential pressure. The time delay and changing slopes are again observed. The pressure difference across the test valve is from 1.1 to 1.8 volts. The difference between the traces for the same air line pressure would indicate any changes in the valve operation for the perturbation considered in the experiments.

The valve differential pressure versus the stem position behavior is shown in Figure 3.5. During the close-to-open stroke, the pressure difference is exponentially related to the stem position. This represents the valve property of equal percentage flow. The shape of the plot represents the normal operating condition of the valve. Changes in rate of response and time delay may be monitored during valve stroking.

3.3.1.2 Response to Step Input with Air Leakage

The step input and the responses of the test valve with air leakage are shown in Figure 3.6. No obvious changes are noticed in this plot for a step response with air leakage.

Figure 3.7 shows the stem position (LVDT) versus air line pressure for a step disturbance with air leakage. The delay time is increased slightly and the shape of

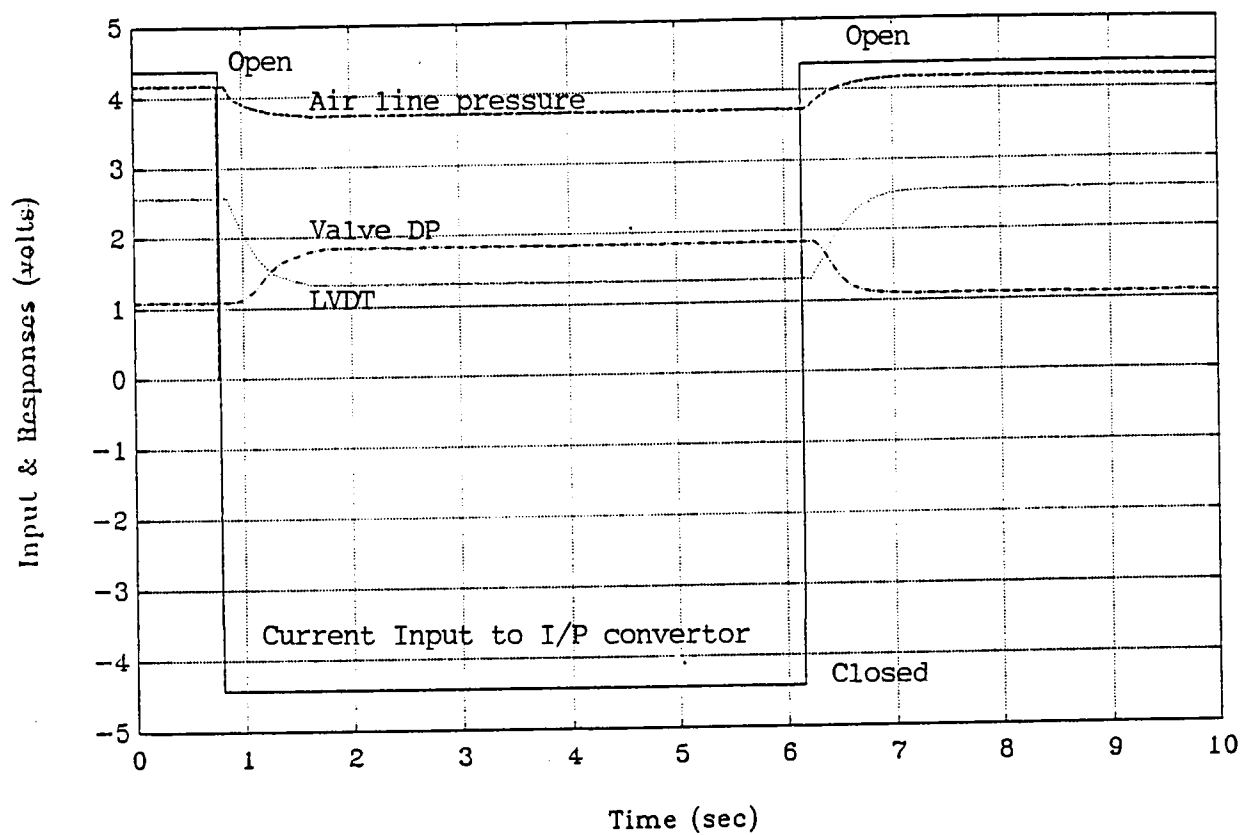


Figure 3.6. Step Responses of the measurements with air leakage.

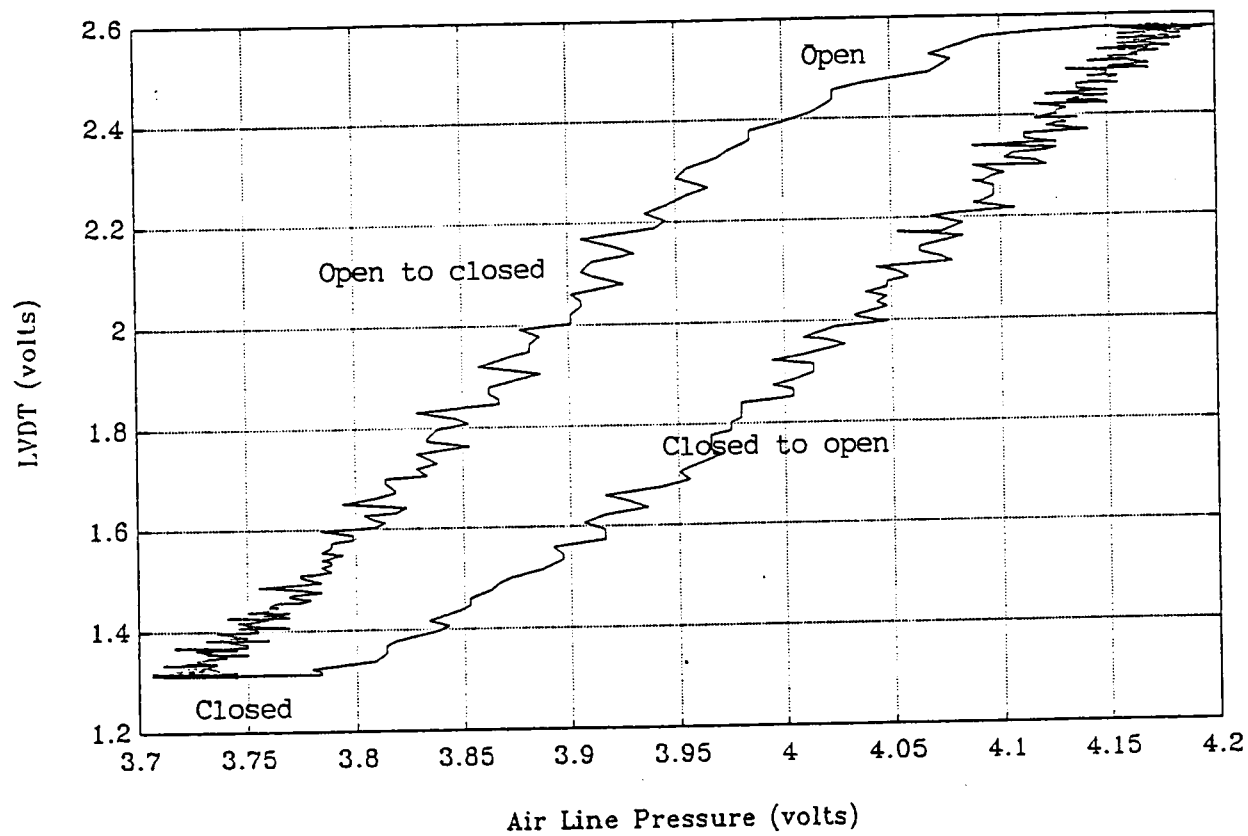


Figure 3.7. Stem Position versus air line pressure for a step response with air leakage.

the slope at the start of motion has been affected. A slower action of the stem travel can be noticed.

Figure 3.8 shows the relationship between the air line pressure and the valve differential pressure signals with air leakage. The delay time increases compared with the normal operation. The slopes of both measurements are not changed as the air leakage is introduced.

The valve differential pressure versus stem position is plotted in Figure 3.9. The shape and the slope are not changed compared with the normal operation except a slight increase in the delay time.

3.3.1.3 Response to Step Input with Increased Stem Friction

The step change in the actuating pressure and the responses of the test valve with friction increase are shown in Figure 3.10. No obvious changes in the step input and responses are found with increased friction.

Figure 3.11 shows the stem position (LVDT) vs. the air line pressure to a step input for the case of increased friction. The delay time is changed and so is the shape of the slopes at the starting points of motion. A greater discontinuity is noticed at the end of the stroke. The close-to-open and open-to-close strokes differ more compared to the normal case. The stem position after a stroke is slightly changed due to the friction changes. As the valve approaches the seating position, more time is needed for full seating.

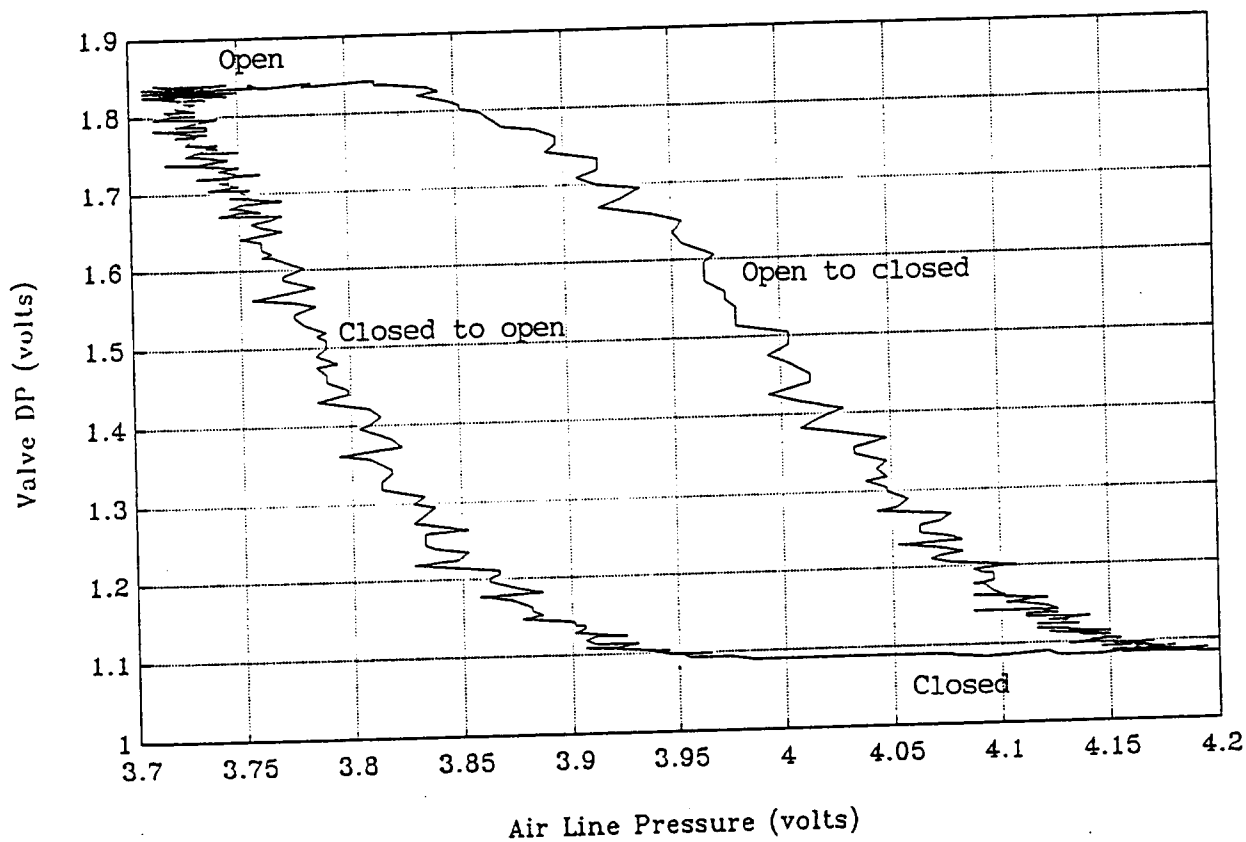


Figure 3.8. Air line pressure versus valve differential pressure to step response with air leakage.

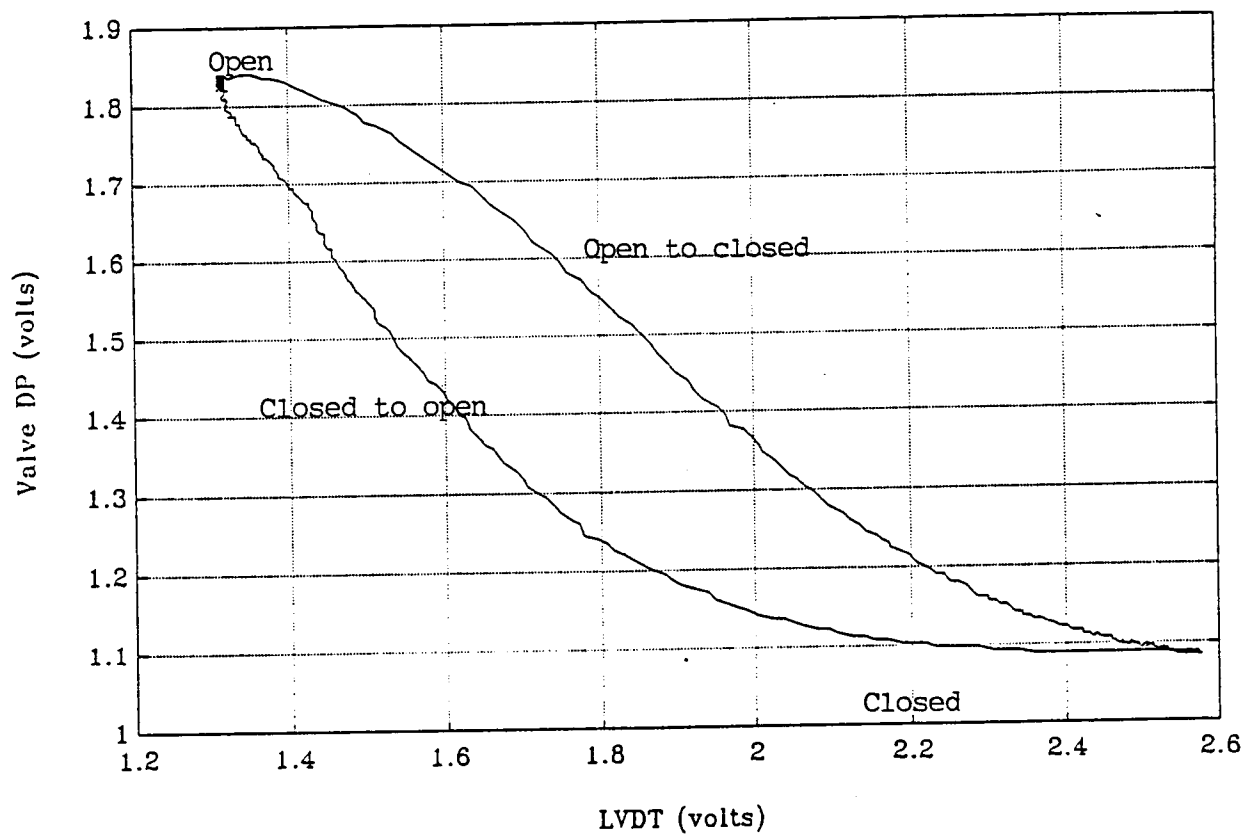


Figure 3.9. LVDT vs. valve differential pressure to step response with air leakage.

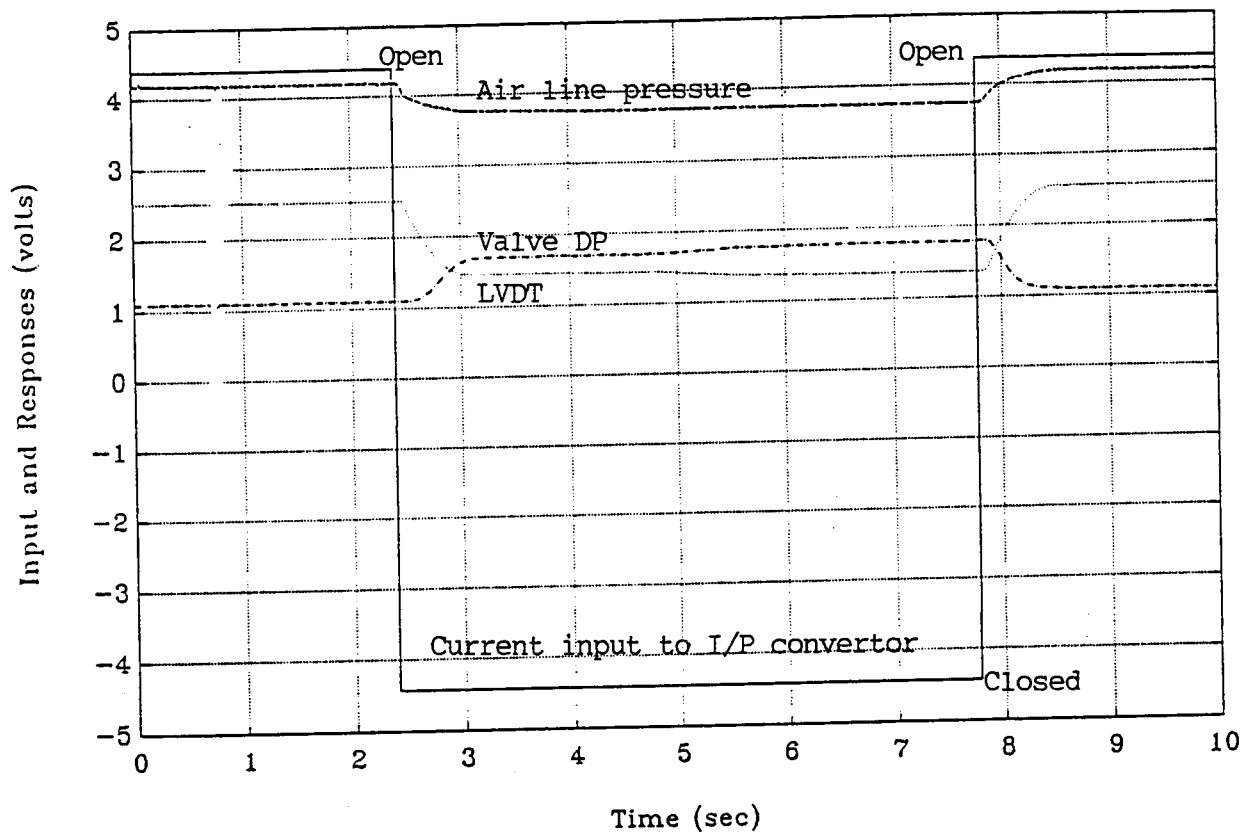


Figure 3.10. Step Responses of the measurements with increased stem friction.

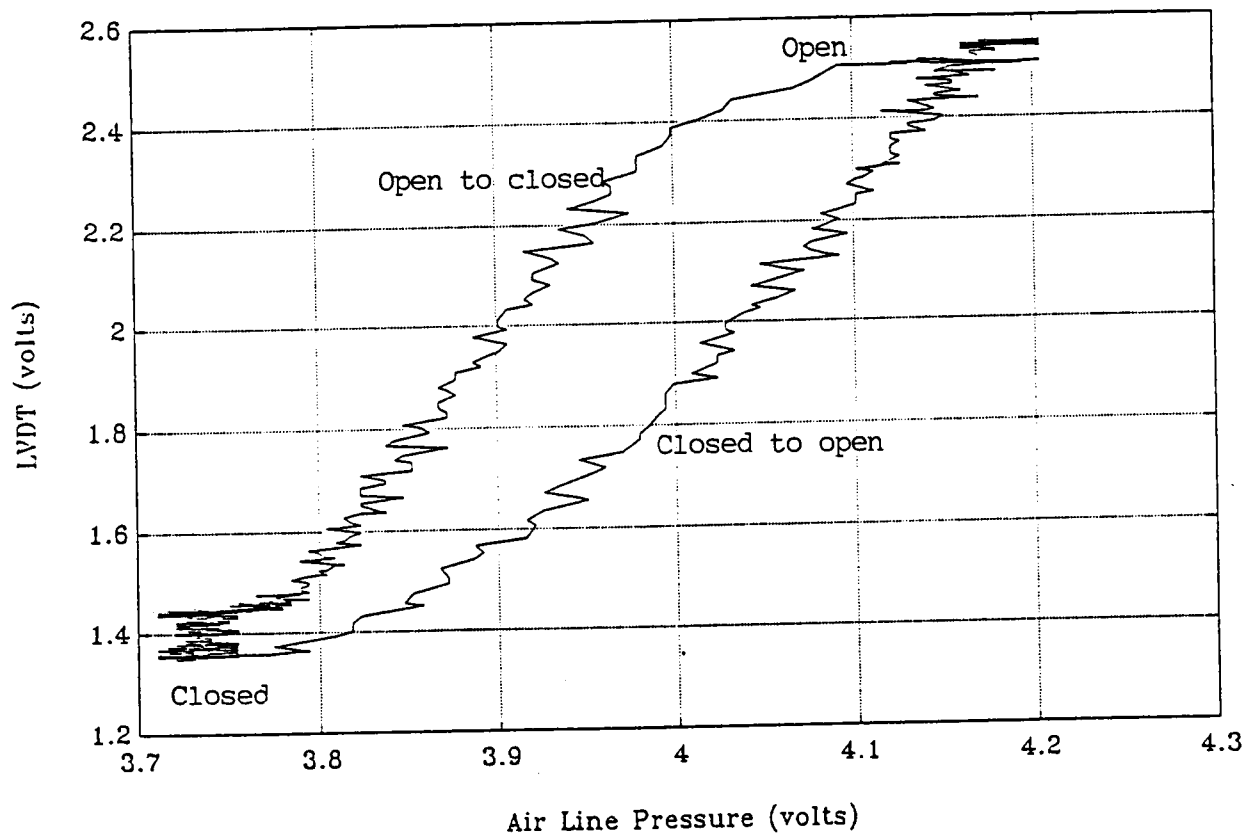


Figure 3.11. Valve stem position (LVDT) vs. air line pressure to a step response with increased stem friction.

Figure 3.12 shows the relationship between air line pressure and valve differential pressure signals for the case of increased friction. The delay time increases compared with other operating conditions. The slopes of both measurements are not changed but more time is spent to complete the valve seating.

The valve differential pressure versus stem position is shown in Figure 3.13. The shape and the slope are changed compared with the normal operation and with air leakage. The discontinuity occurs more with friction increase.

3.3.2 Response to Ramp Changes in Air Line Pressure

Ramp pressure perturbation input is introduced using a function generator. As the current is gradually changed, the air pressure out of the I/P convertor changes proportionally to the current.

3.3.2.1 Response to Ramp Input at Normal Operation

Figure 3.14 shows the ramp input to the system and the responses of the test valve system. A pressure ramp from 3 to 15.5 psig was gradually applied in 10 seconds during the close-to-open stroke. During the open-to-close stroke, the pressure reduces gradually from 15.5 to 3 psig in 10 seconds.

Figure 3.15 shows a full stroke of the air line pressure vs. LVDT signal. The slopes of both traces are very linear except for the starting points. Discontinuities are seen on the plot. The pressure from the I/P convertor is not very stable. The

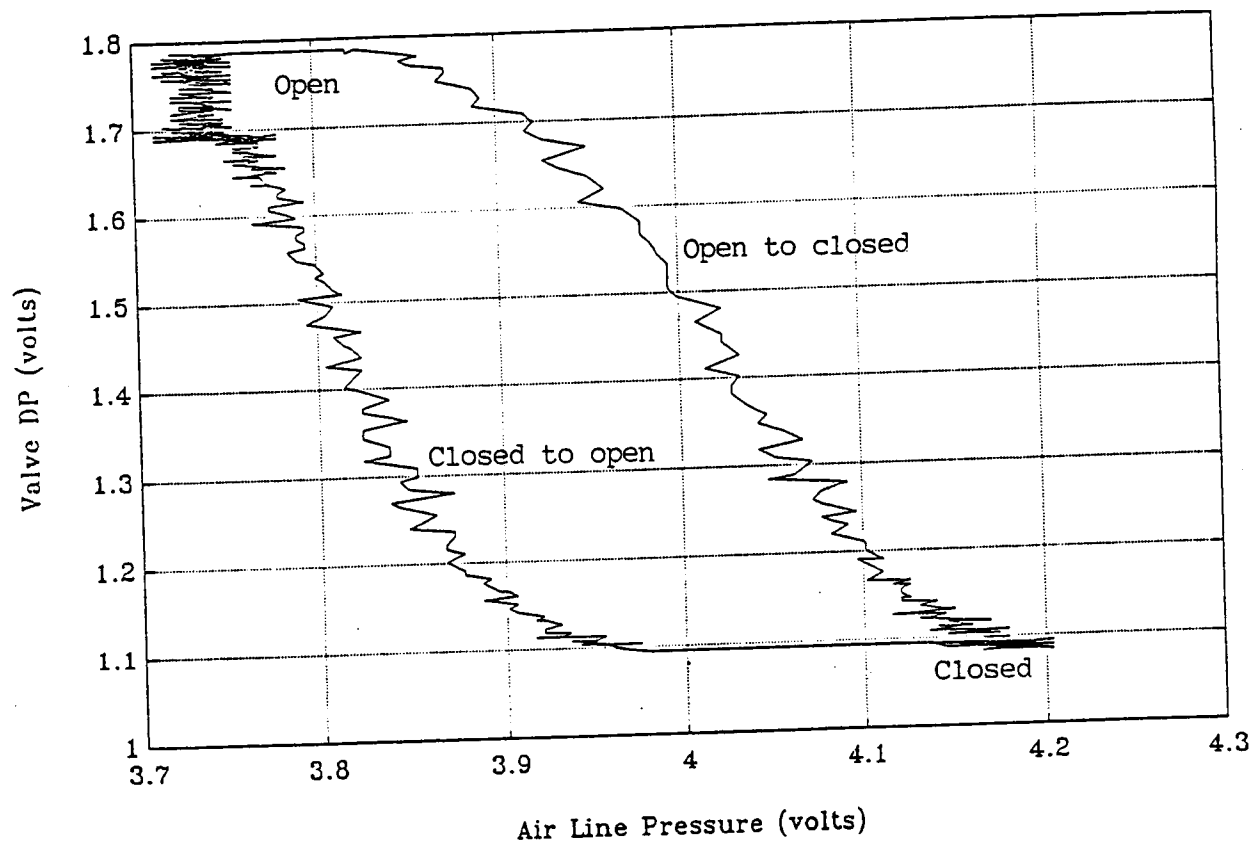


Figure 3.12. Valve differential pressure vs. air line pressure to a step response with increased stem friction.

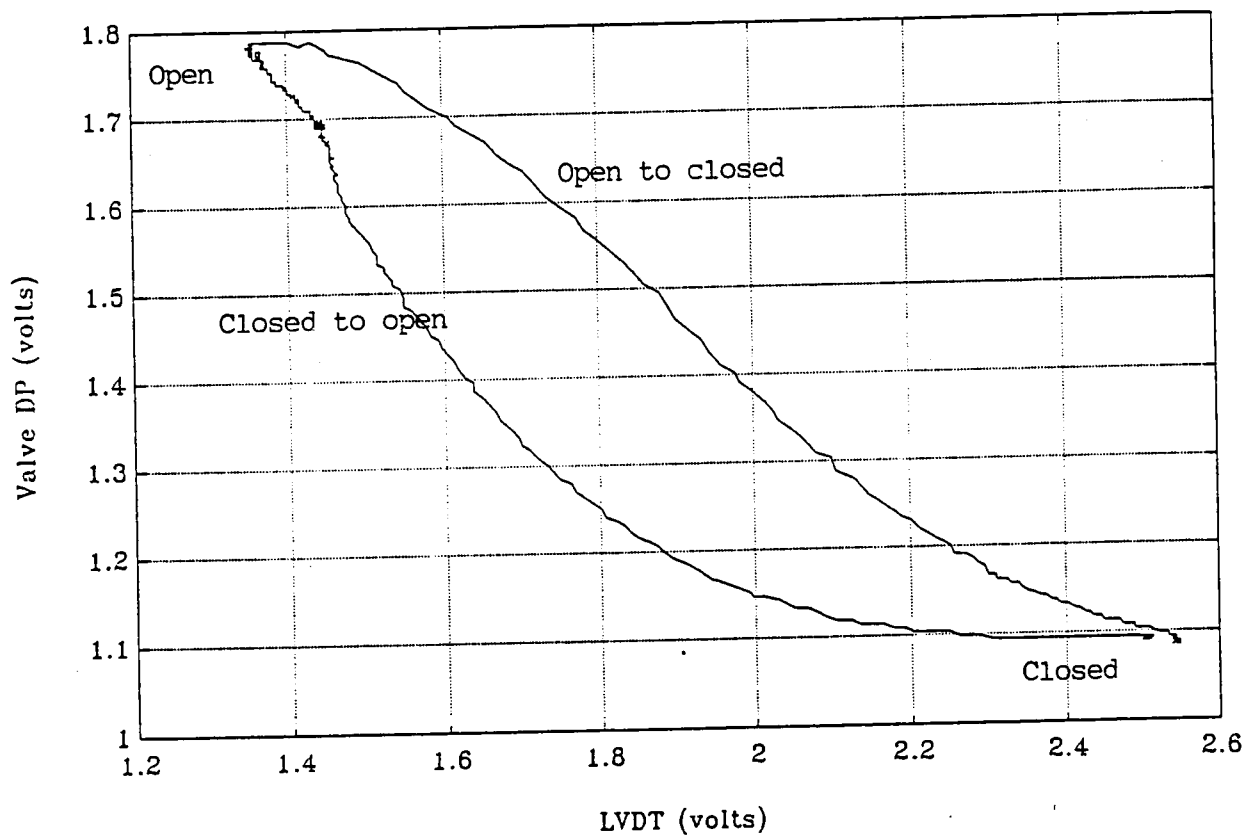


Figure 3.13. Valve differential pressure vs. stem position to a step response with increased stem friction.

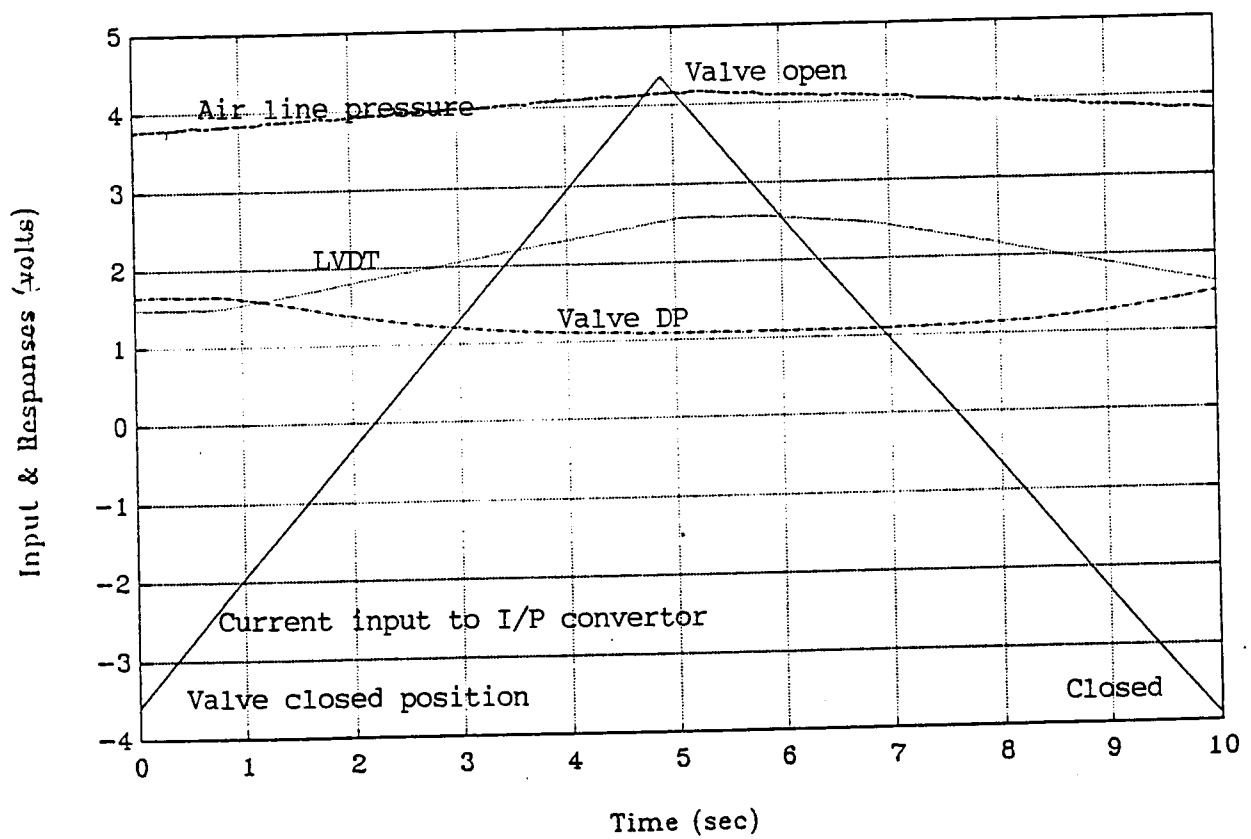


Figure 3.14. Ramp input and responses of the valve variables.

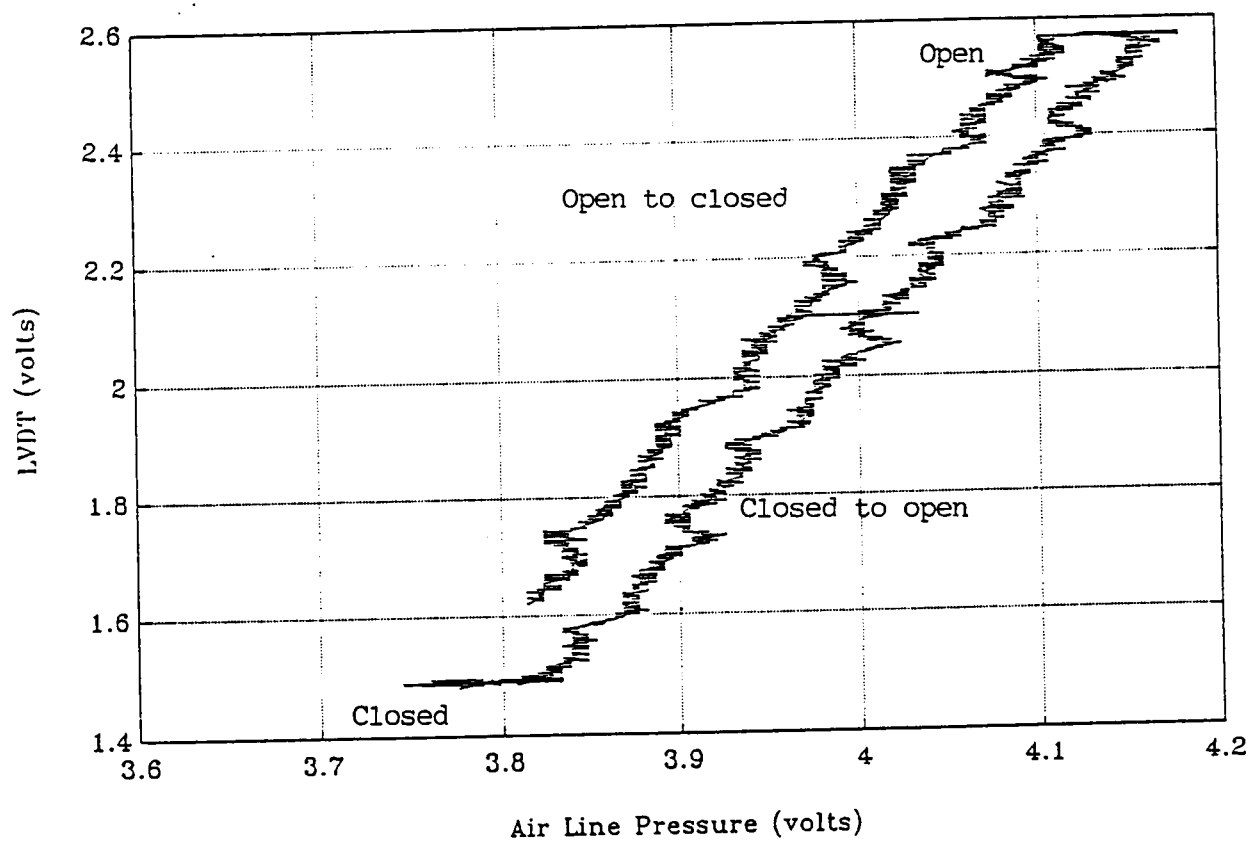


Figure 3.15. Valve stem motion (LVDT) vs. air line pressure for a ramp input.

distance between the traces shows the valve property and the time delay. This distance is much smaller than that for the step response. This is because the valve system has sufficient time to come to equilibrium during the strokes.

Figure 3.16 shows the relationship between the air line pressure and the valve DP. The shape of the ramp response is similar to the step response, but the distance between the two traces is also smaller than that for the step response.

The differential pressure versus stem position graph is given in Figure 3.17. Compared with the step response of the measurements, both traces are similar to each other with almost the same slope. This is because the ramp input is considerably smaller than the step input changes. The discontinuity of the plots shows that the measurements fluctuate. Any anomalies in the valve will result in changes in the characteristics and can be used for detecting incipient changes.

3.3.2.2 Responses to Ramp Input with Air Leakage

The responses of the measurements to a ramp input for the case of air leakage are plotted in Figures 3.18 through 3.21.

Figure 3.18 shows the time plot of the air line pressure, LVDT, and valve differential pressure signals to a ramp input change in the actuating pressure with air leakage. The changes in the plots are negligible. Small air leakages will not be detected by ramp tests.

Figure 3.19 shows the air line pressure vs. LVDT position to ramp input with air leakage. The delay time has increased slightly and the slopes of the plots have

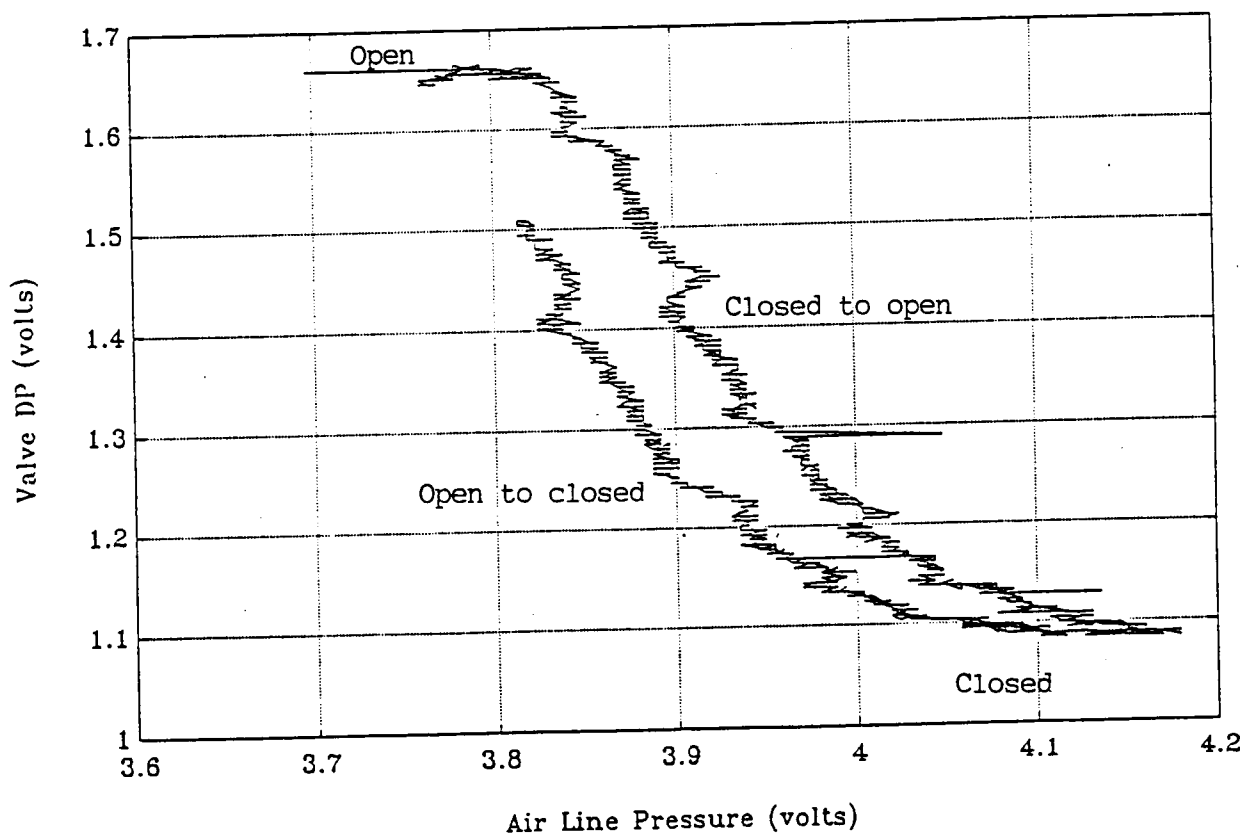


Figure 3.16. Valve differential pressure vs. air line pressure for a ramp input.

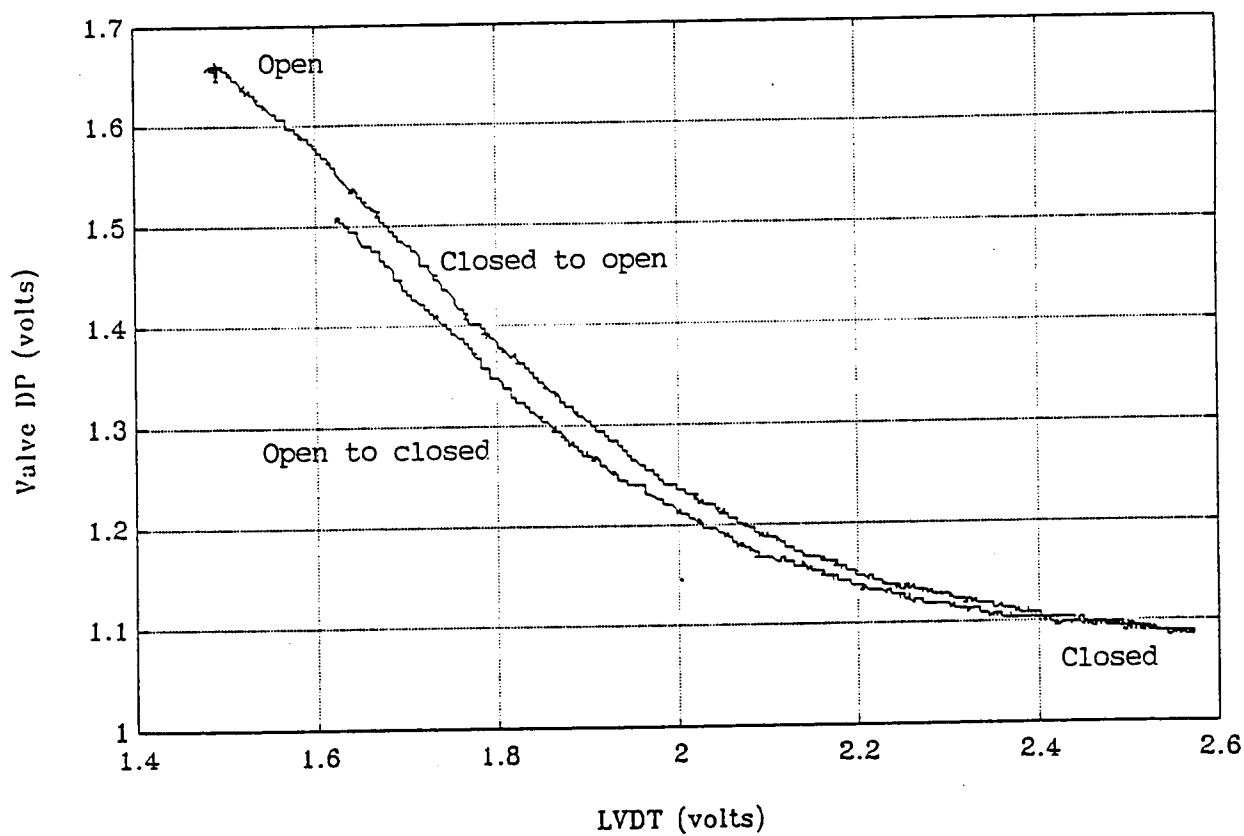


Figure 3.17. Valve differential pressure vs. LVDT for a ramp input.

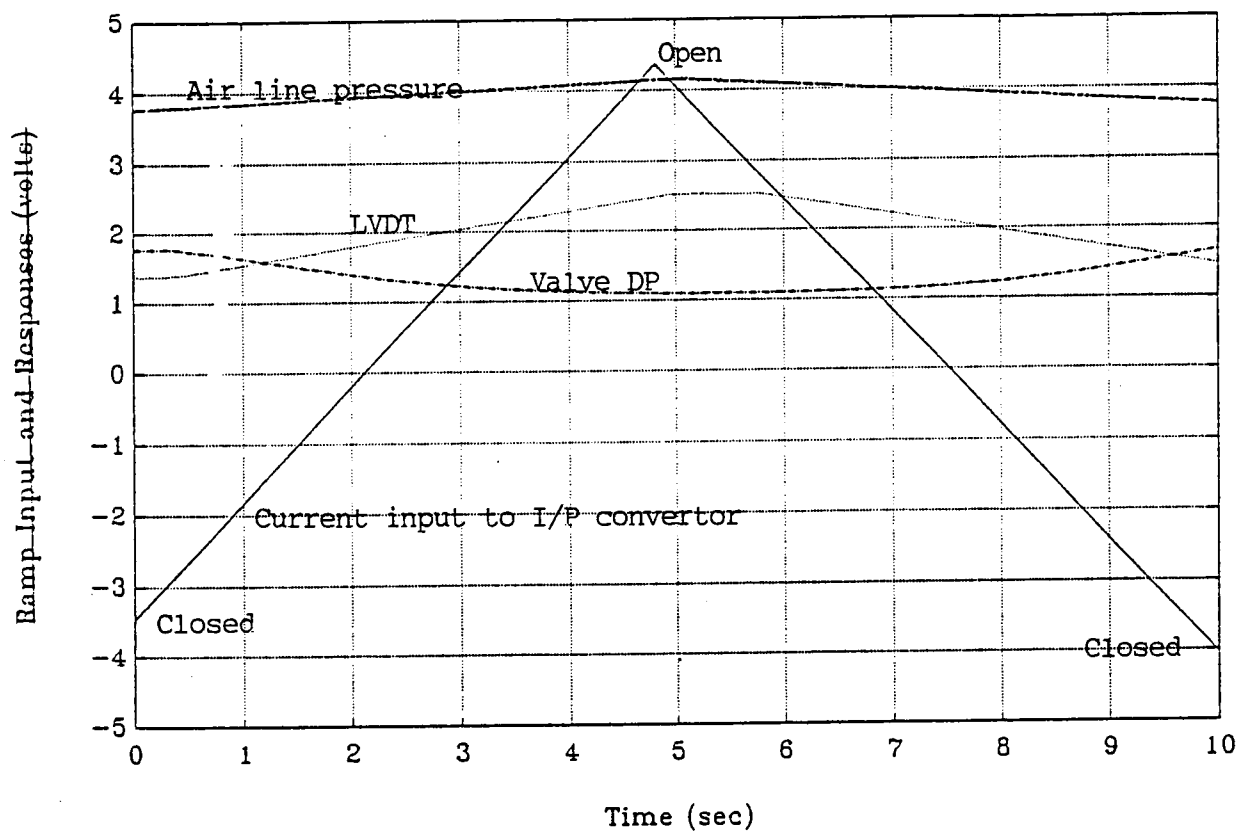


Figure 3.18. Ramp input and responses of the variables with air leakage.

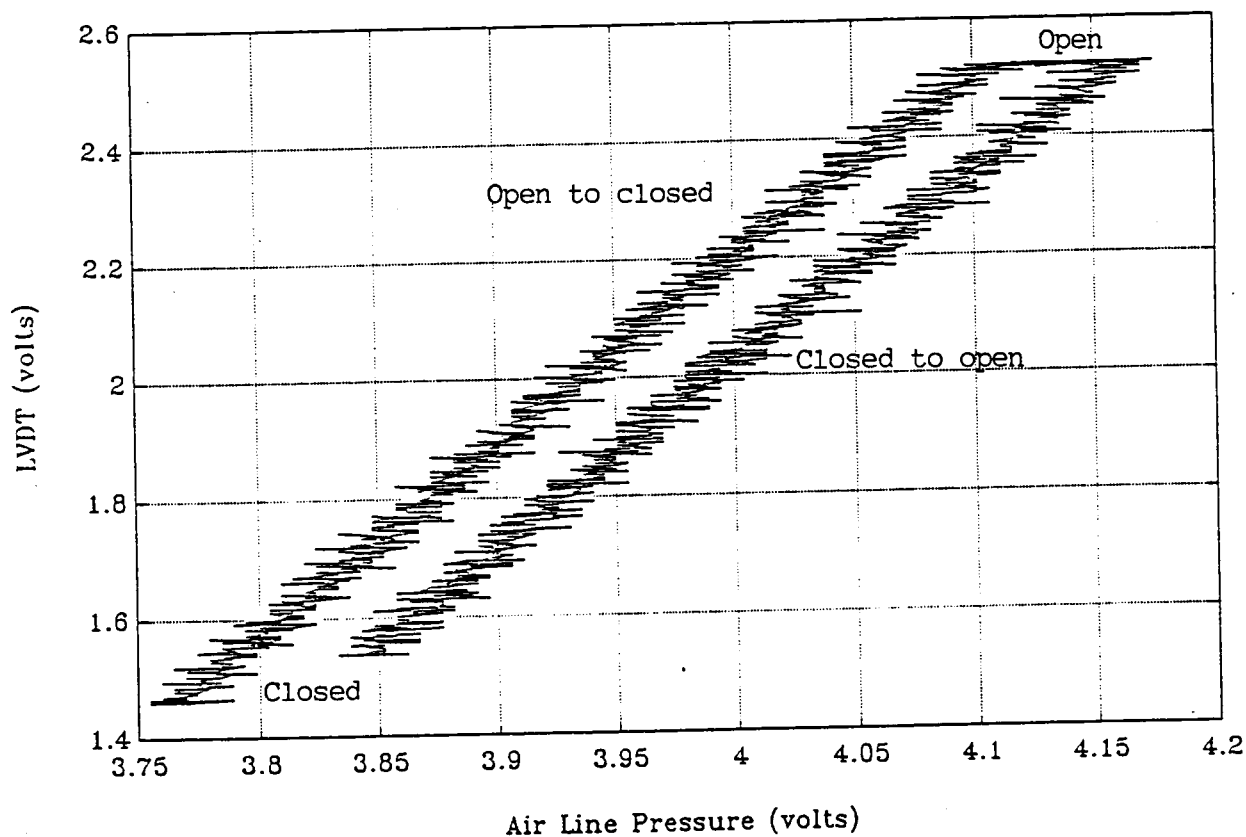


Figure 3.19. Valve stem motion (LVDT) vs. air line pressure for a ramp input with air leakage.

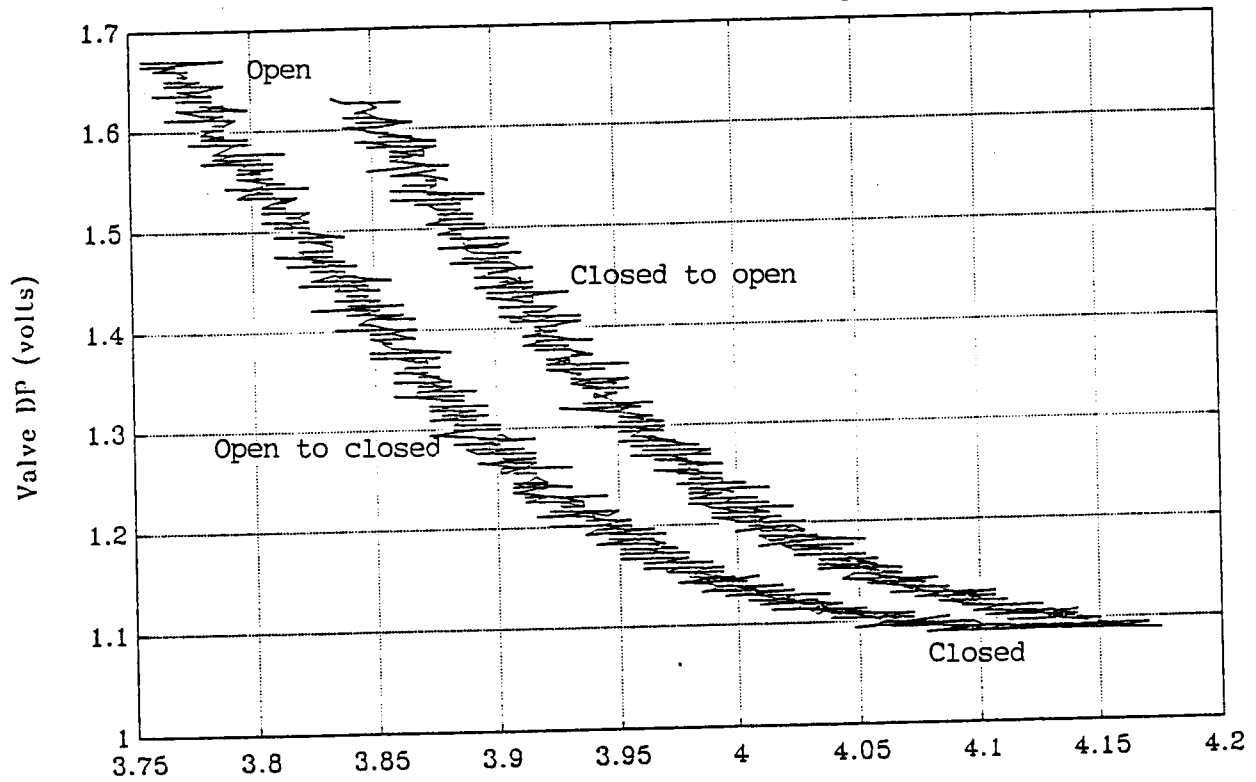


Figure 3.20. Valve differential pressure vs. air line pressure for a ramp input with air leakage.

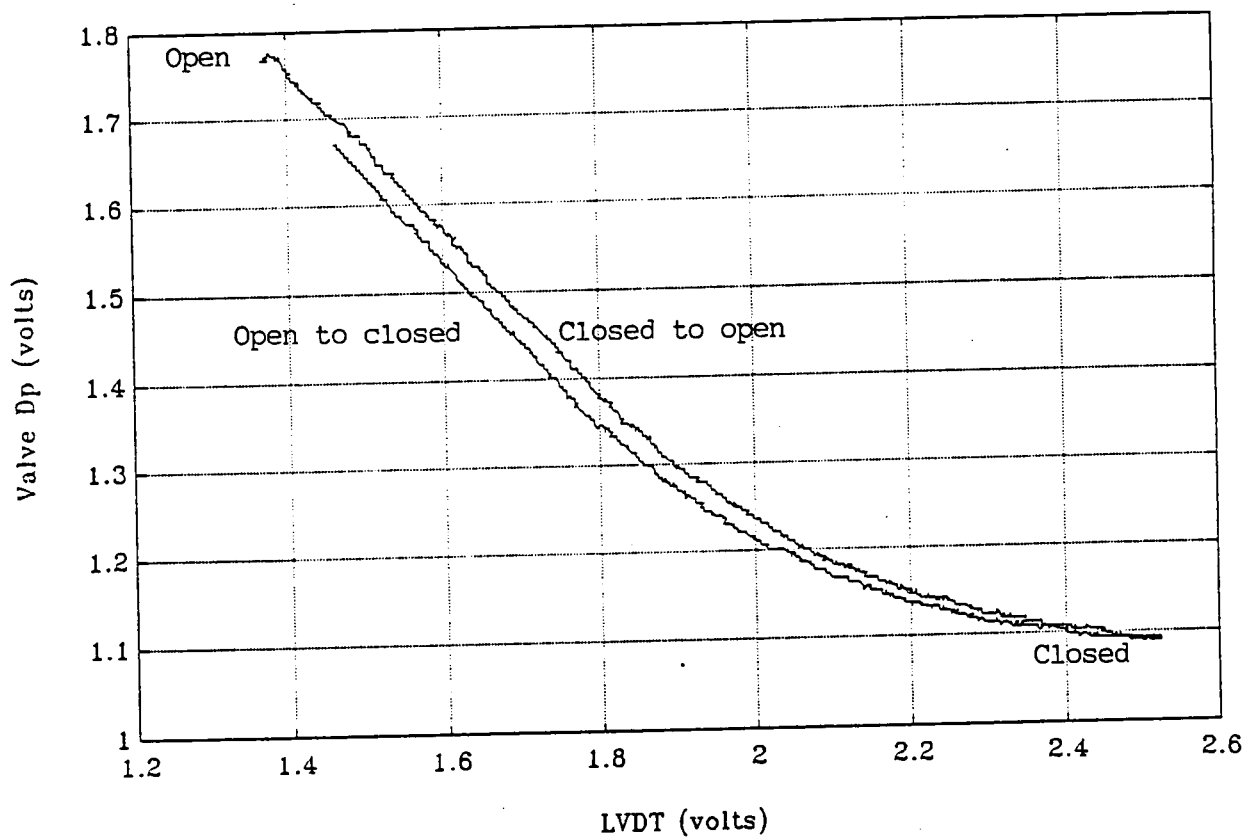


Figure 3.21. Valve differential pressure vs. stem motion for a ramp input with air leakage.

not changed.

Figure 3.20 shows the relationship between the air line pressure and the valve differential pressure signals with air leakage. The slopes of both measurements are not changed as the air leakage is introduced, and the delay time does not show any obvious changes.

The valve differential pressure versus the stem position is plotted in Figure 3.21. The shape and the slope are changed slightly compared with normal operation. The delay time decreases by a small amount compared to the responses during normal operation.

3.3.2.3 Responses to Ramp Input with Increased Stem Friction

The responses of the variables to a ramp input with increased stem friction are plotted in Figures 3.22 through 3.25.

Figure 3.22 shows the plots of the air line pressure, LVDT, and cross valve differential pressure signals to a ramp input with increased friction. The changes in the plots are negligible indicating that the ramp pressure input will not show a small amount of friction variation.

Figure 3.23 shows the air line pressure vs. LVDT signal to a ramp input with increased stem friction. The delay time has increased and the slopes of the relationship have changed. A larger discontinuity has occurred in the plot. The slopes are not always linear for full stroke plot. The friction variation is clearly shown on this plot.

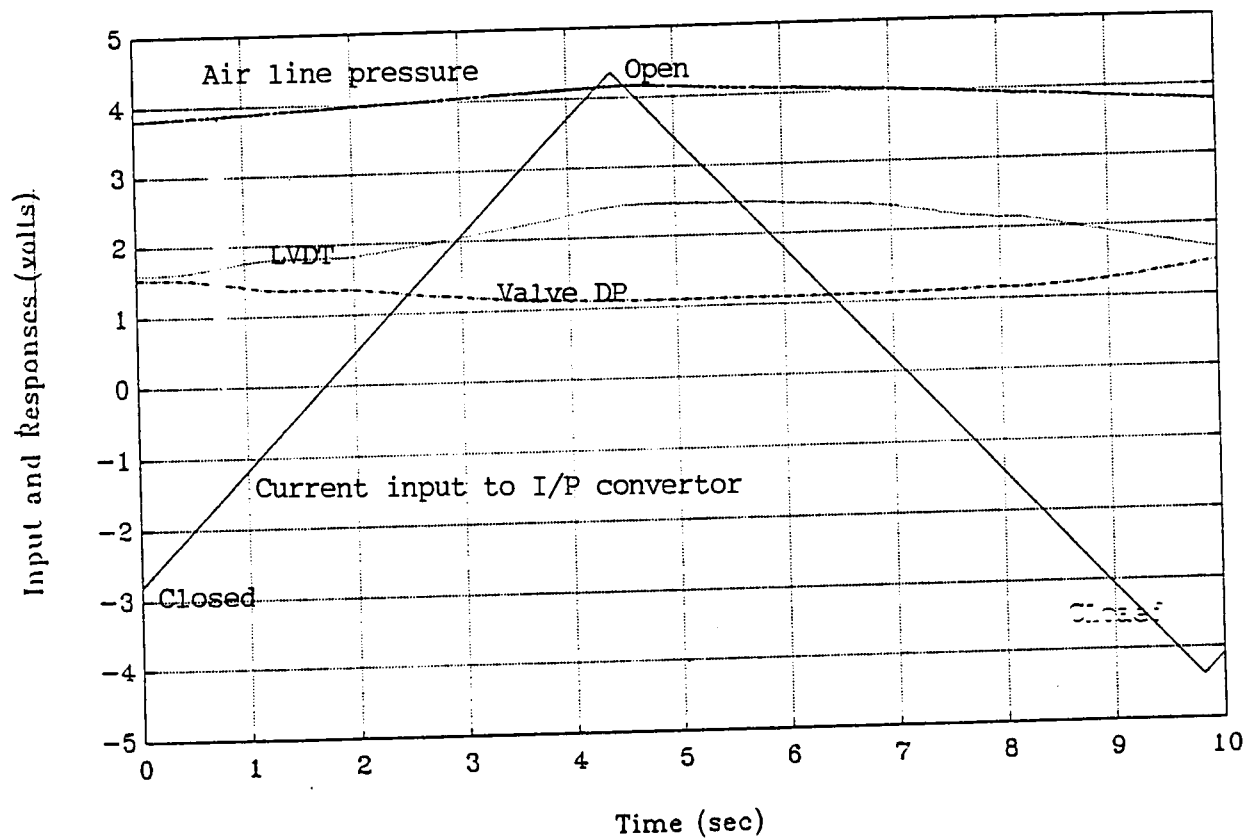


Figure 3.22. Ramp input and responses of the measurements with increased stem friction.

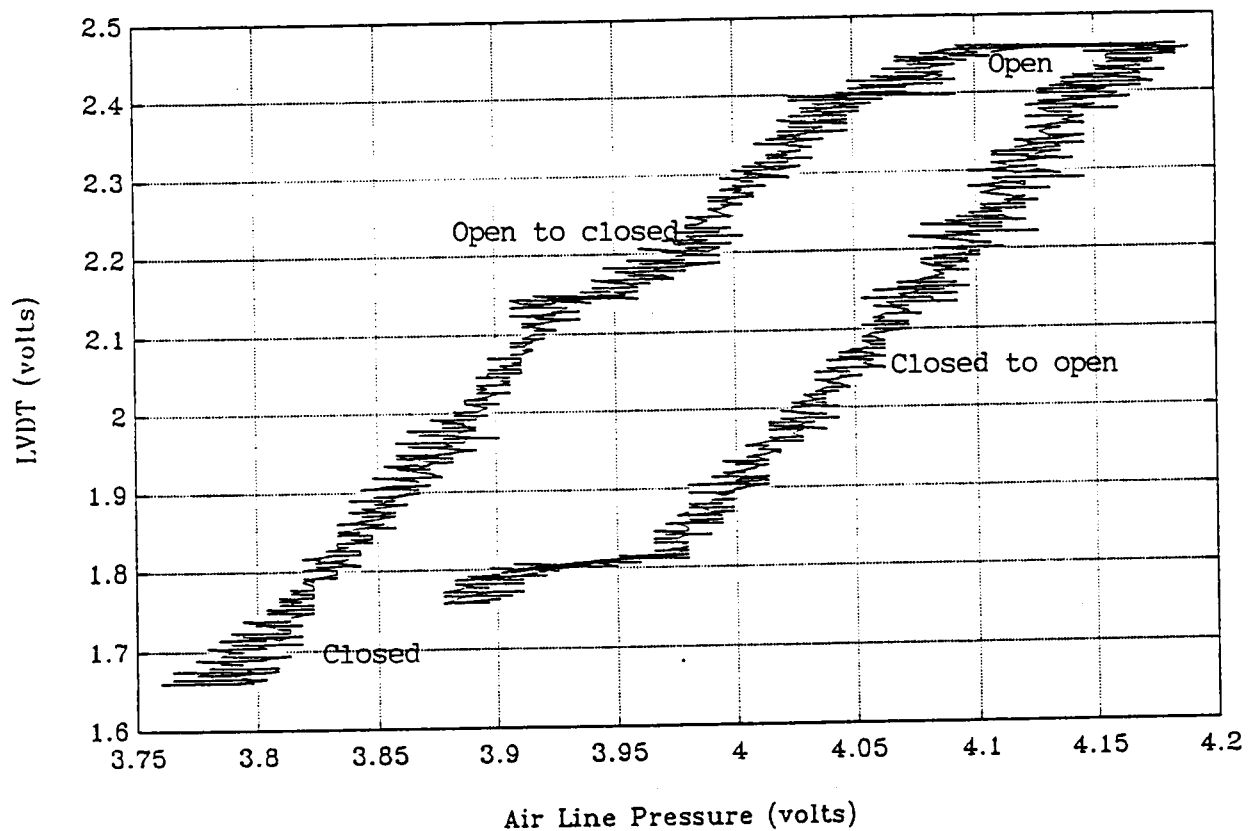


Figure 3.23. Air line pressure vs. LVDT to ramp input with increased stem friction.

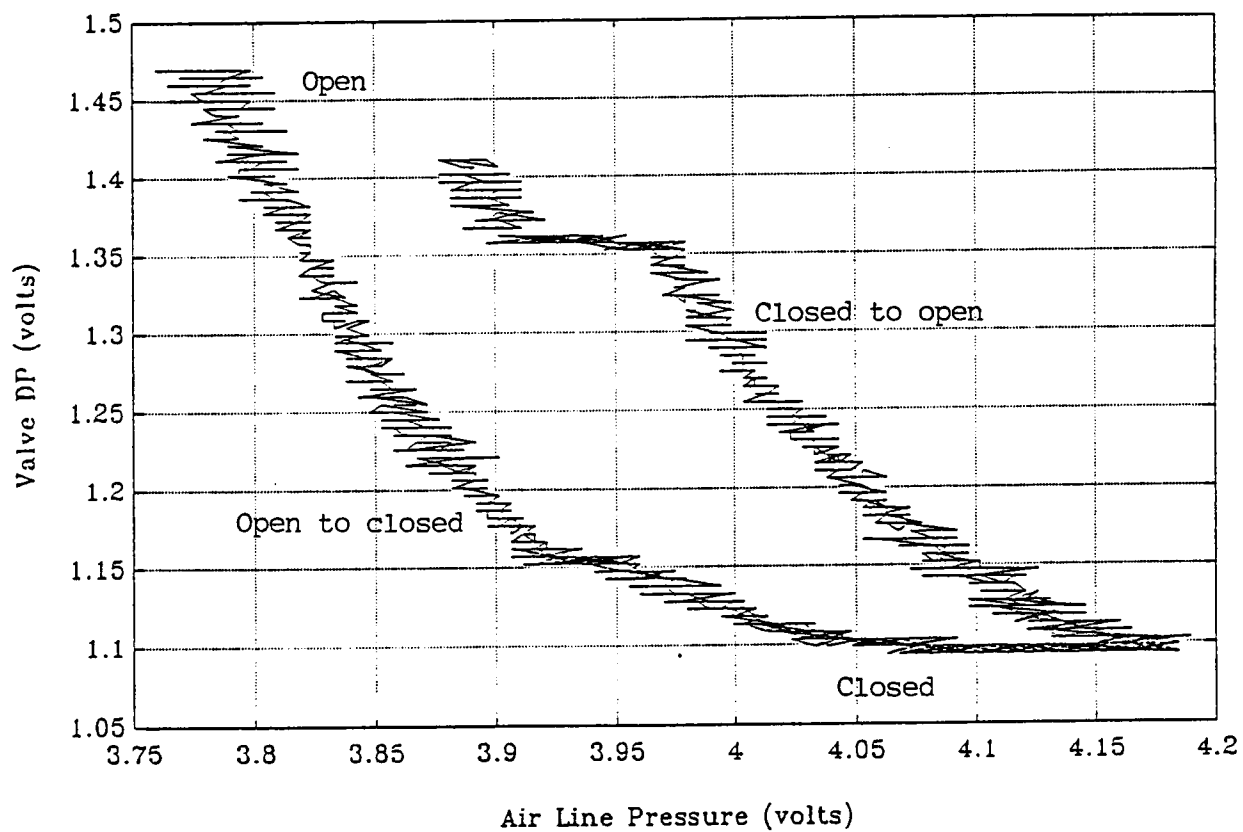


Figure 3.24. Air line pressure vs. cross valve differential pressure to ramp input with increased stem friction.

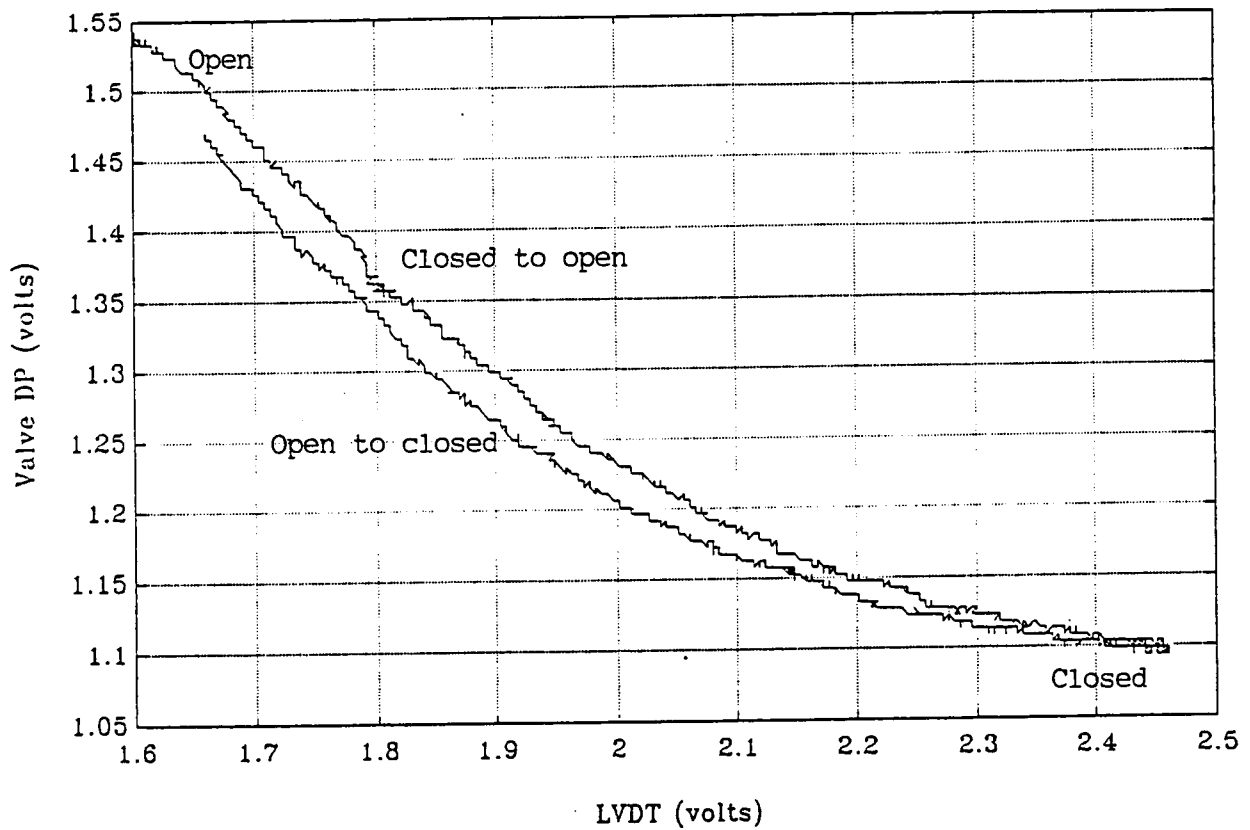


Figure 3.25. LVDT vs. cross valve pressure to ramp input with increased stem friction.

Figure 3.24 shows the relationship between the air line pressure and the valve differential pressure signals with increased stem friction. The shapes of the plots have changed and a discontinuity has occurred as the friction on the stem is increased. The delay time between the air line pressure and the valve differential pressure signals increases considerably. The seating time is also increased as the friction increases.

Figure 3.25 shows the LVDT signal vs. valve differential pressure signal. The shape and the slope have changed slightly compared to the normal operation. The delay time increases by a small amount. More discontinuity is seen in the plot. The effect of friction is clearly represented by these hysteresis plots.

3.3.3 Response to Sinusoidal Changes in Air Line Pressure

Sinusoidal pressure input was also introduced by a function generator. The air line pressure varied from 3 to 15.5 psig. A stroke was completed in about 10 seconds.

3.3.3.1 Response to Sinusoidal Input at Normal Operation

Figure 3.26 shows the sinusoidal pressure input and responses of the variables. The input was slowly applied to allow for the system to respond properly. Figure 3.27 shows the air line pressure vs. LVDT signal to sinusoidal input. Figure 3.28 shows the relationship between the air line pressure and cross valve differential

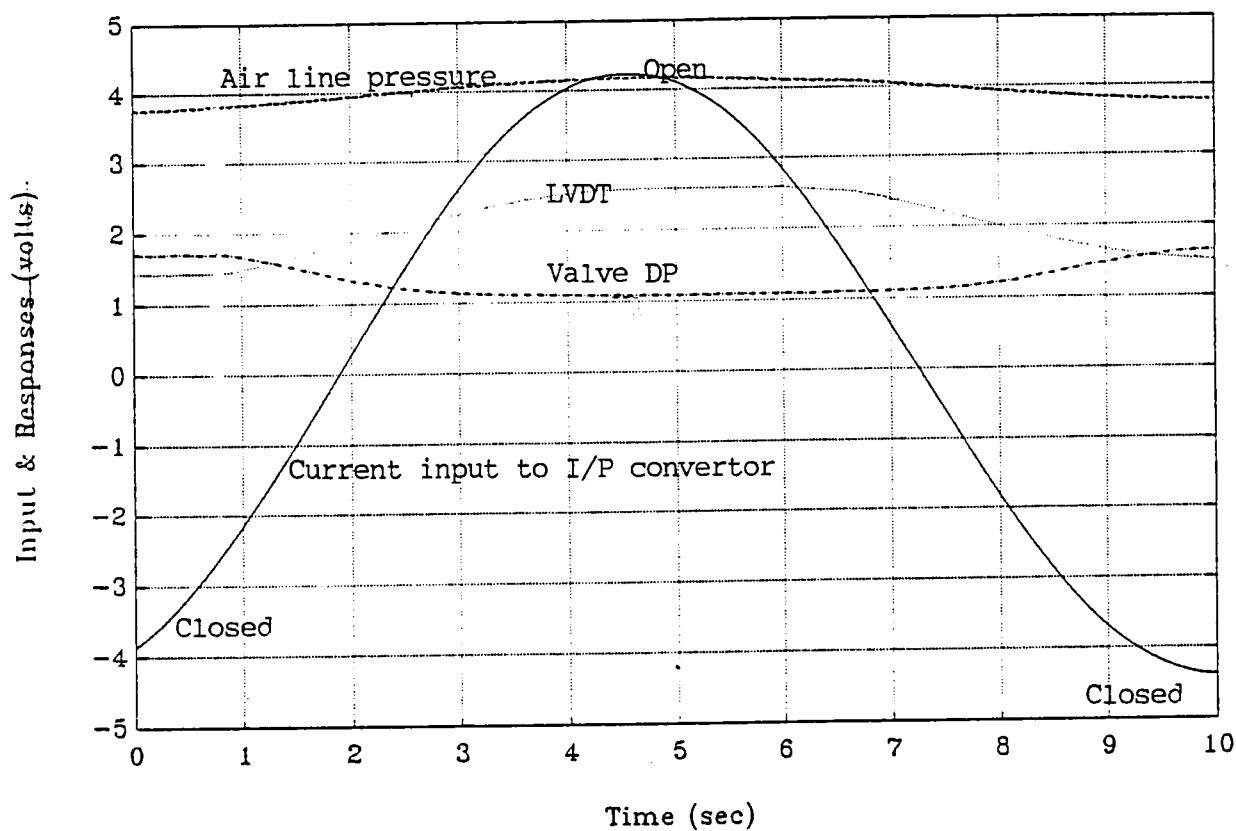


Figure 3.26. Sinusoidal pressure input and responses of the measurements.

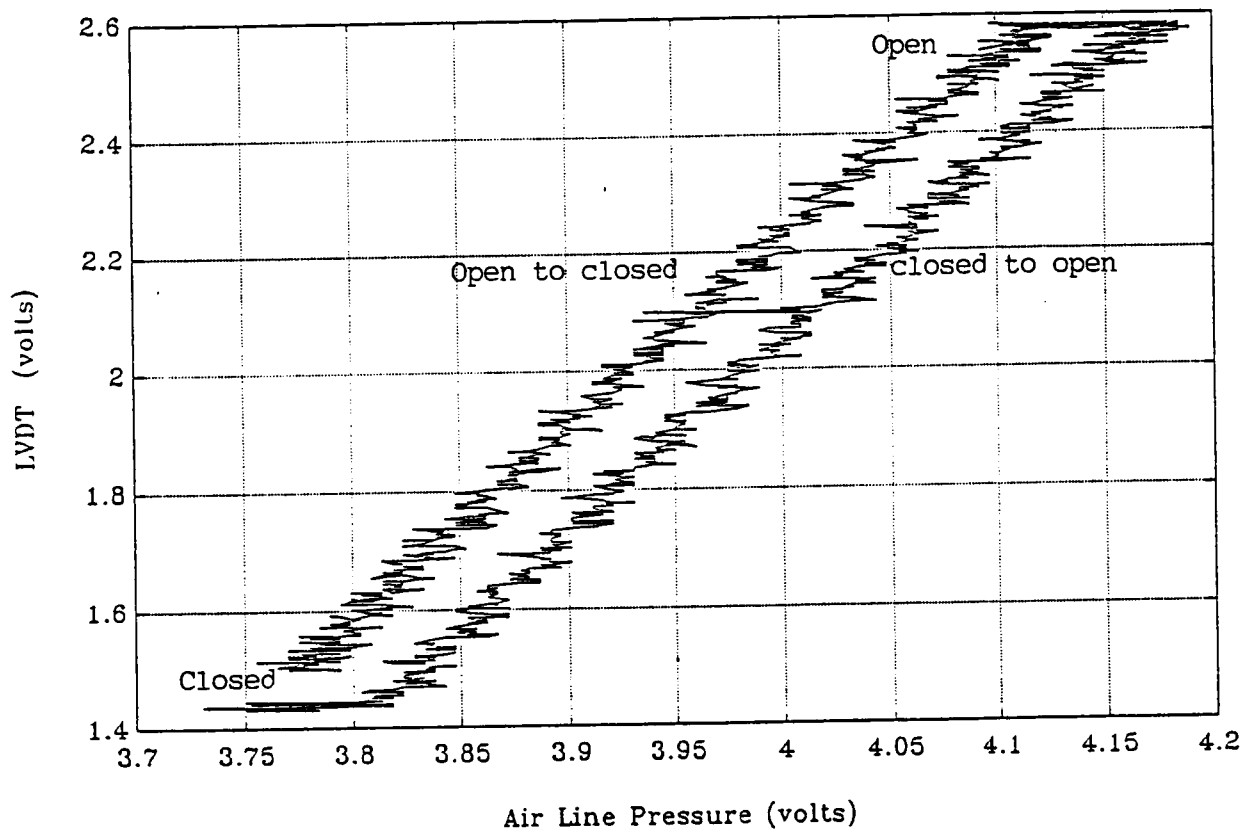


Figure 3.27. Air line pressure vs. LVDT to sinusoidal input.

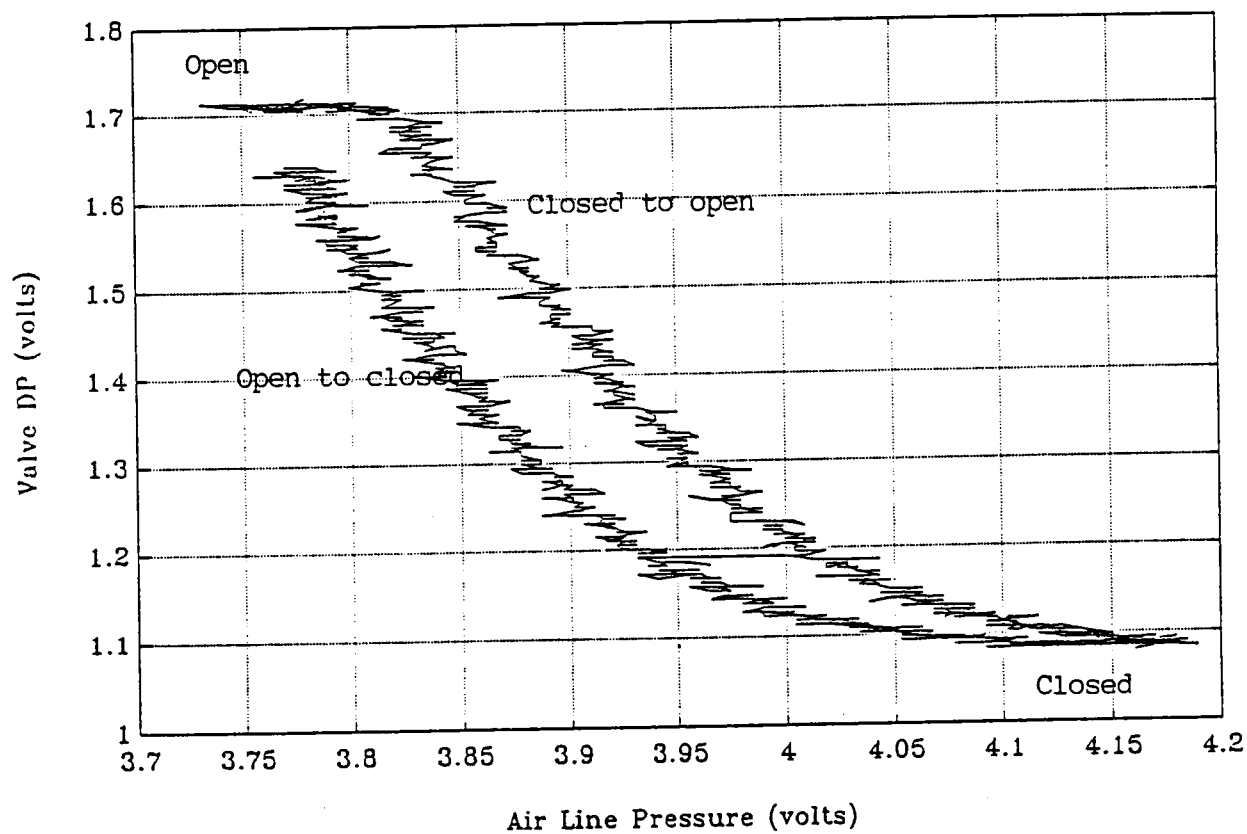


Figure 3.28. Air line pressure vs. cross valve different pressure to sinusoidal input.

pressure signals. The cross valve pressure versus stem position is given in Figure 3.29. The shapes of the responses for the ramp and sinusoidal inputs are almost similar to each other with small variations.

3.3.3.2 Response to Sinusoidal Input with Air Leakage

The responses of the measurements to a sinusoidal pressure input with simulated air leakage are plotted in Figures 3.30 through 3.33.

Figure 3.30 shows the time series plot of the air line pressure, LVDT, and cross valve differential pressure signals to a sinusoidal input with air leakage. Figure 3.31 shows the air line pressure vs. LVDT position indicator for this case. Figures 3.32 and 3.33 show the relationship between air line pressure and valve differential pressure, and LVDT and valve differential pressure signals, respectively. The responses are similar to the normal operation.

3.3.3.3 Response to Sinusoidal Input with Increased Friction

The responses of the variables to a sinusoidal pressure input with increased stem friction are plotted in Figures 3.34 through 3.37. Figure 3.34 shows the transient plot of air line pressure, LVDT signal, and cross valve differential pressure signal for this case. No obvious changes in the time plots are seen. Figure 3.35 shows the air line pressure signal vs. LVDT signal to sinusoidal pressure input with increased stem friction. The slopes of the hysteresis graph are not linear. The effect of friction

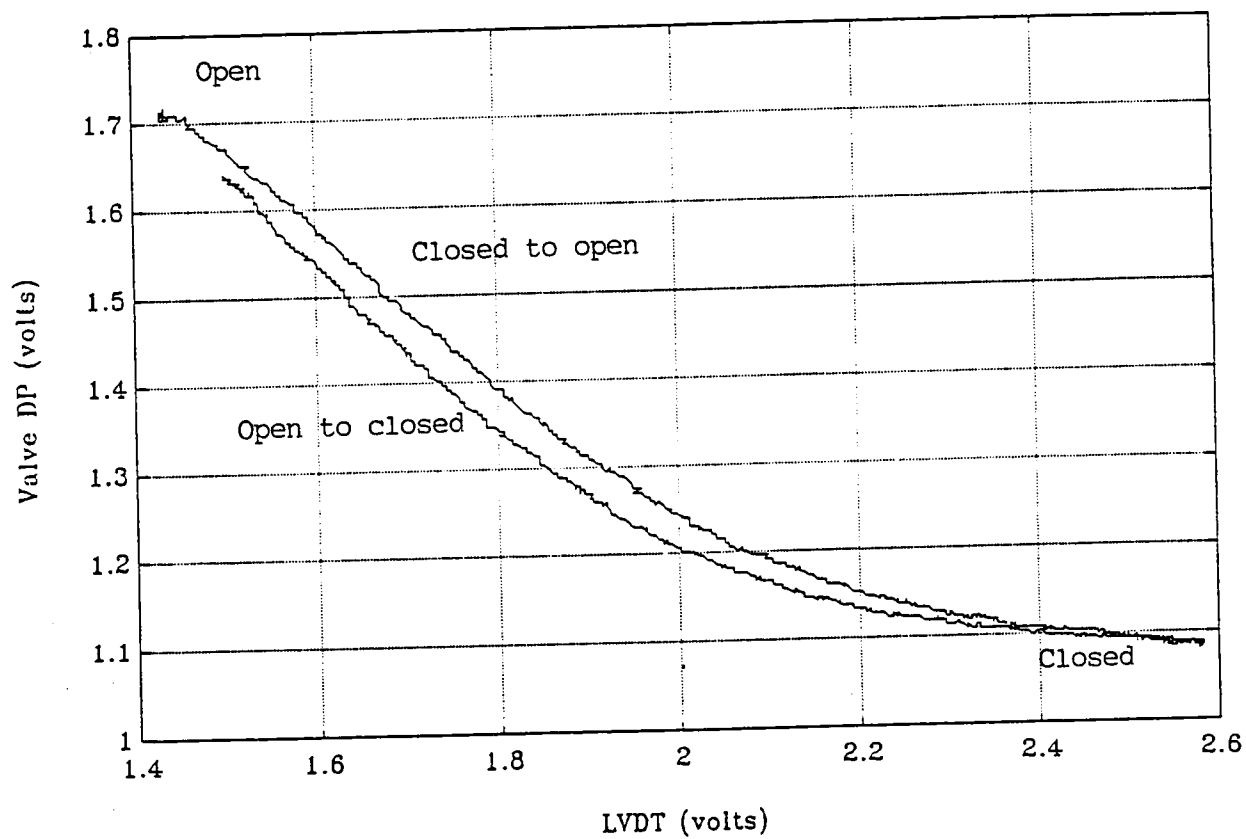


Figure 3.29. LVDT vs. across valve differential pressure to sinusoidal input.

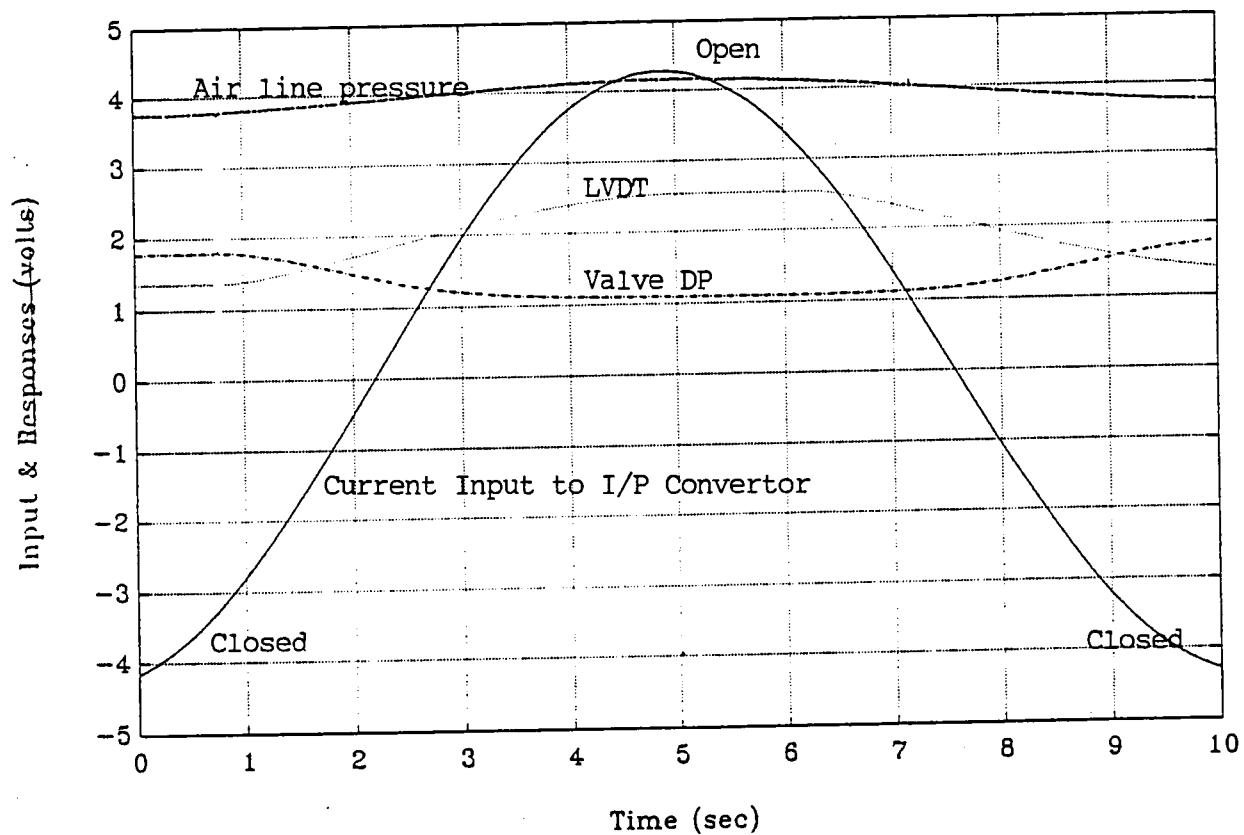


Figure 3.30. Sinusoidal input and the responses of the variables to an air leakage.

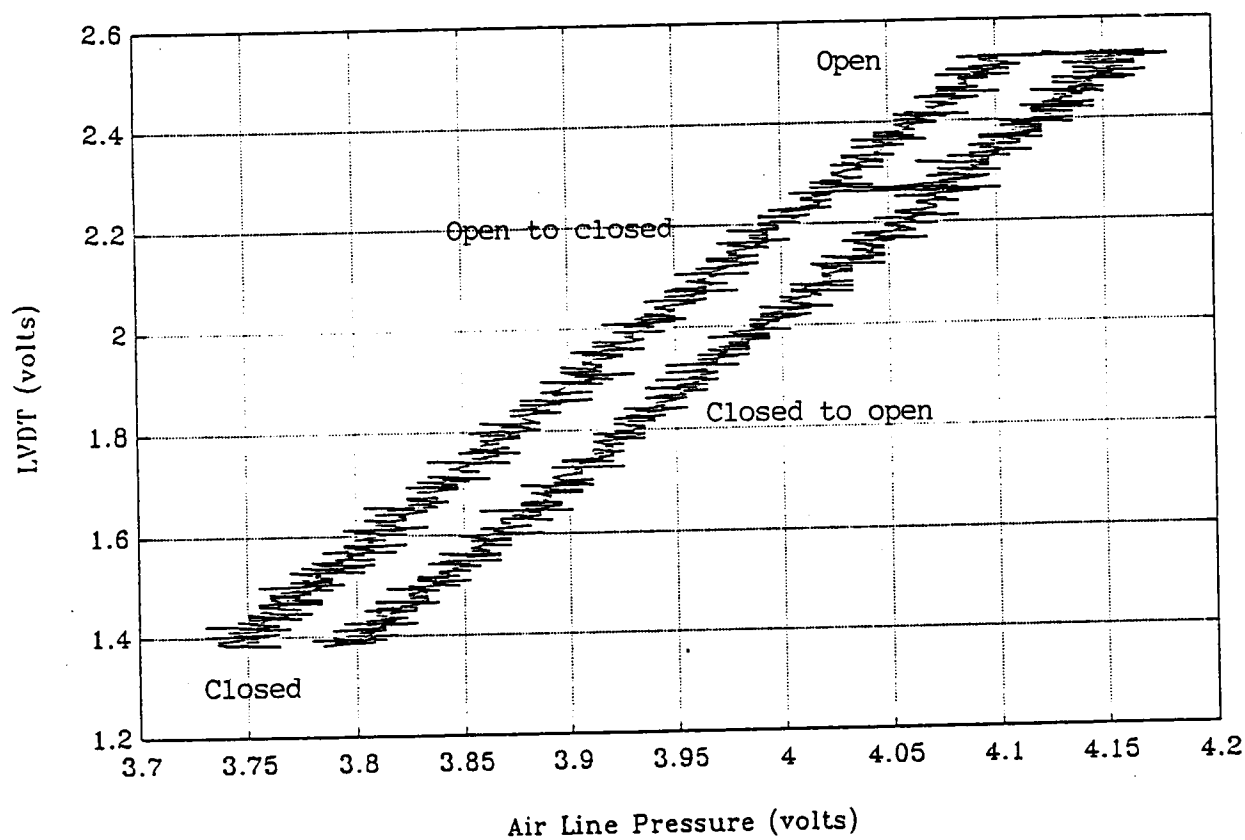


Figure 3.31. Air line pressure vs. LVDT signal to sinusoidal input with air leakage.

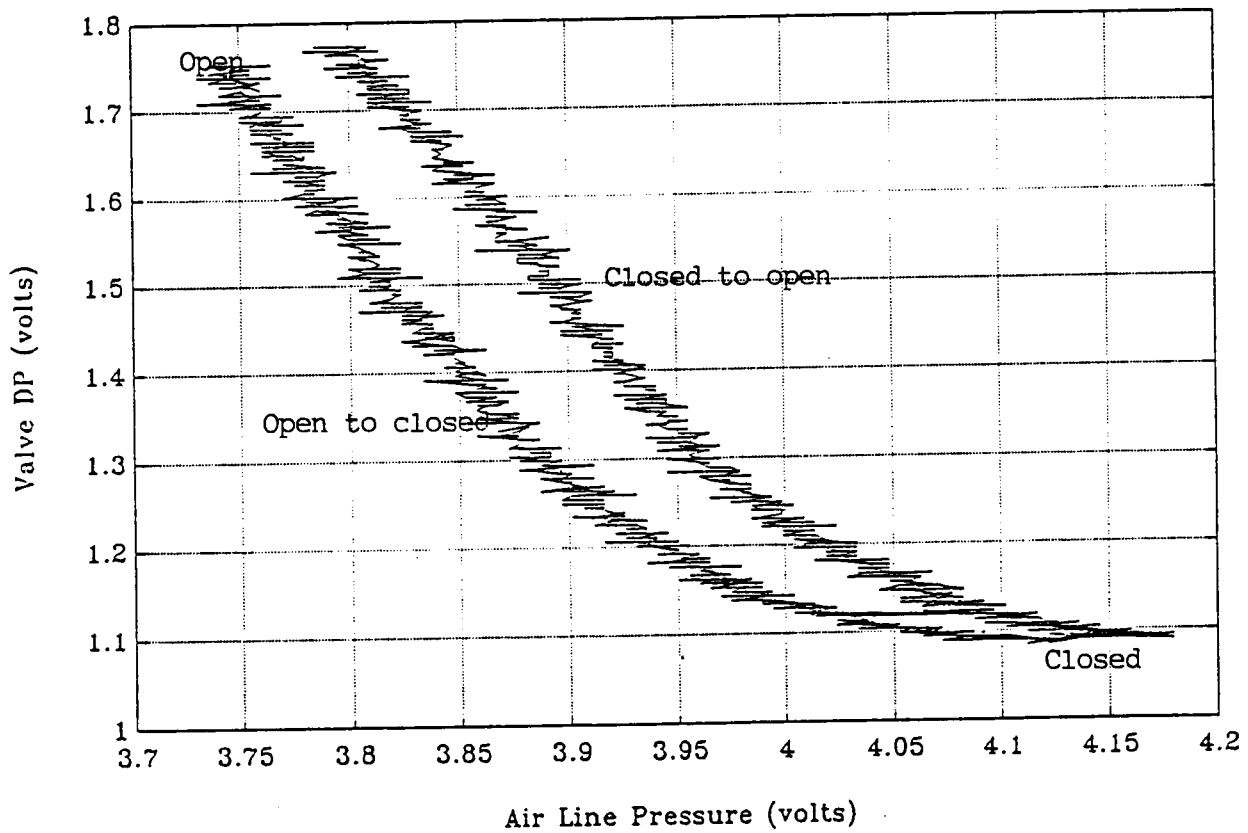


Figure 3.32. Air line pressure vs. cross valve differential pressure to sinusoidal input with air leakage.

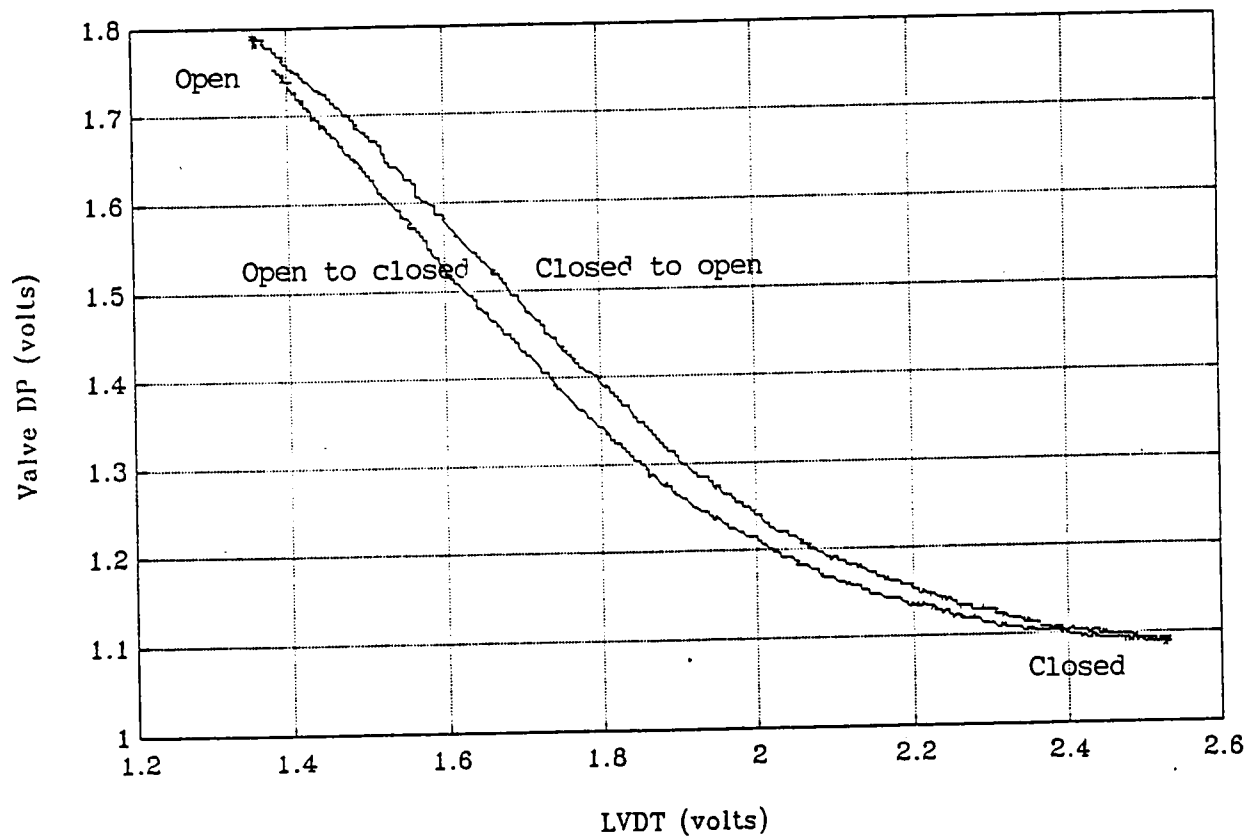


Figure 3.33. LVDT signal vs. cross valve differential pressure to sine wave input with air leakage.

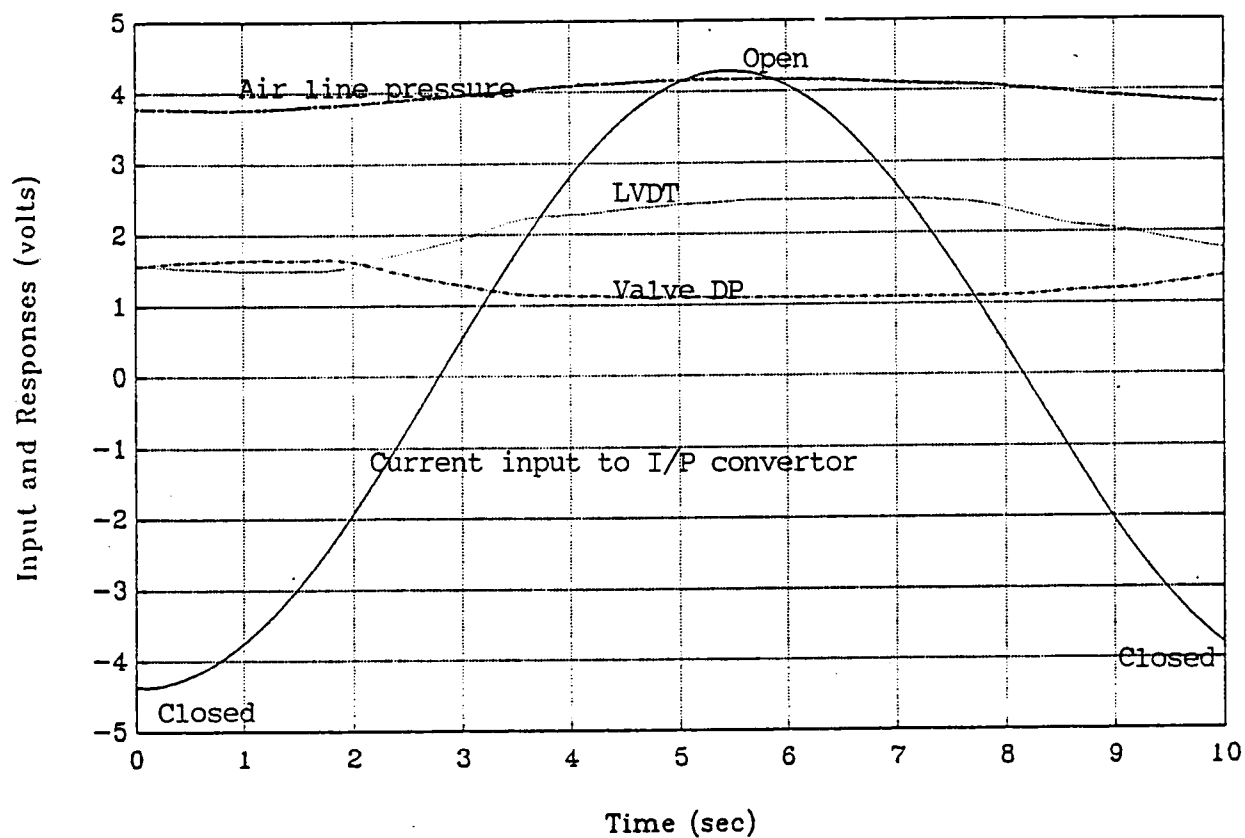


Figure 3.34. Sinusoidal pressure input and response of the measurements with increased stem friction.

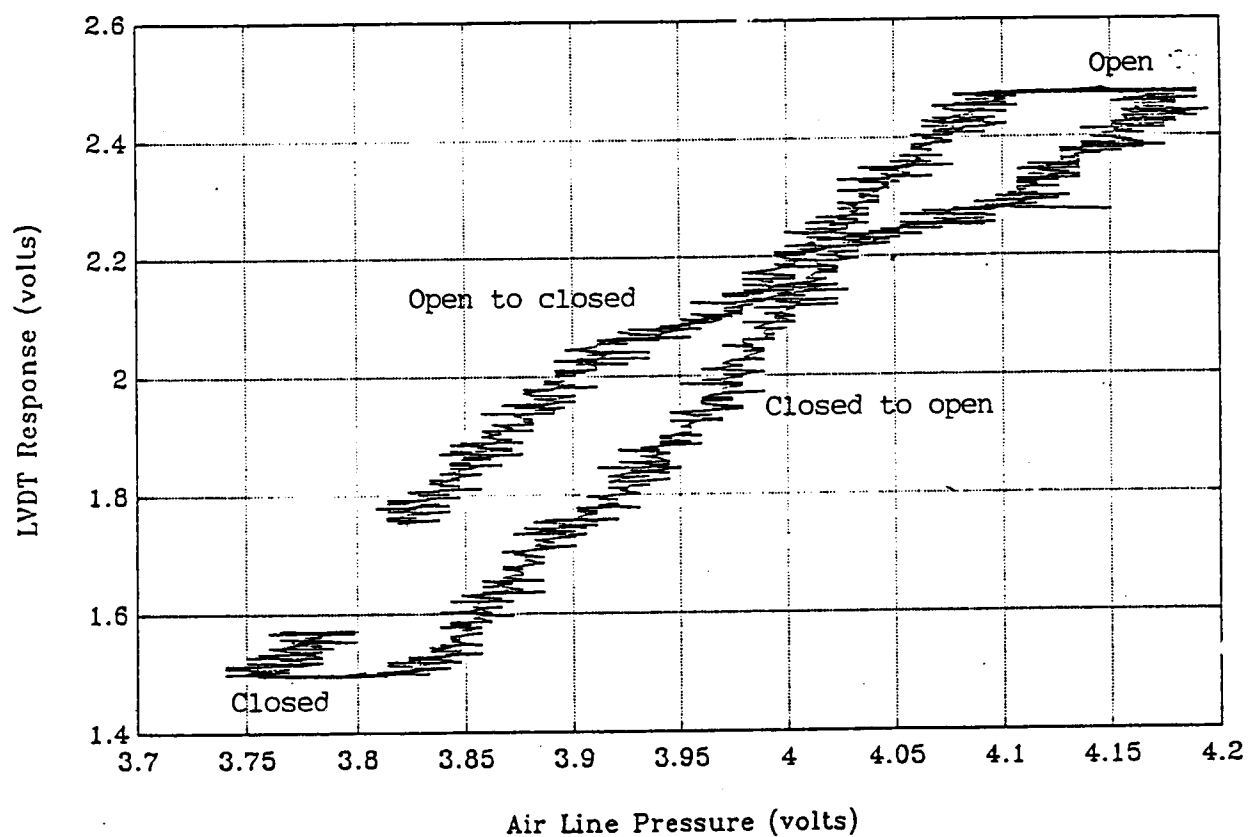


Figure 3.35. Air line pressure vs. LVDT signal for a sinusoidal input with increased stem friction.

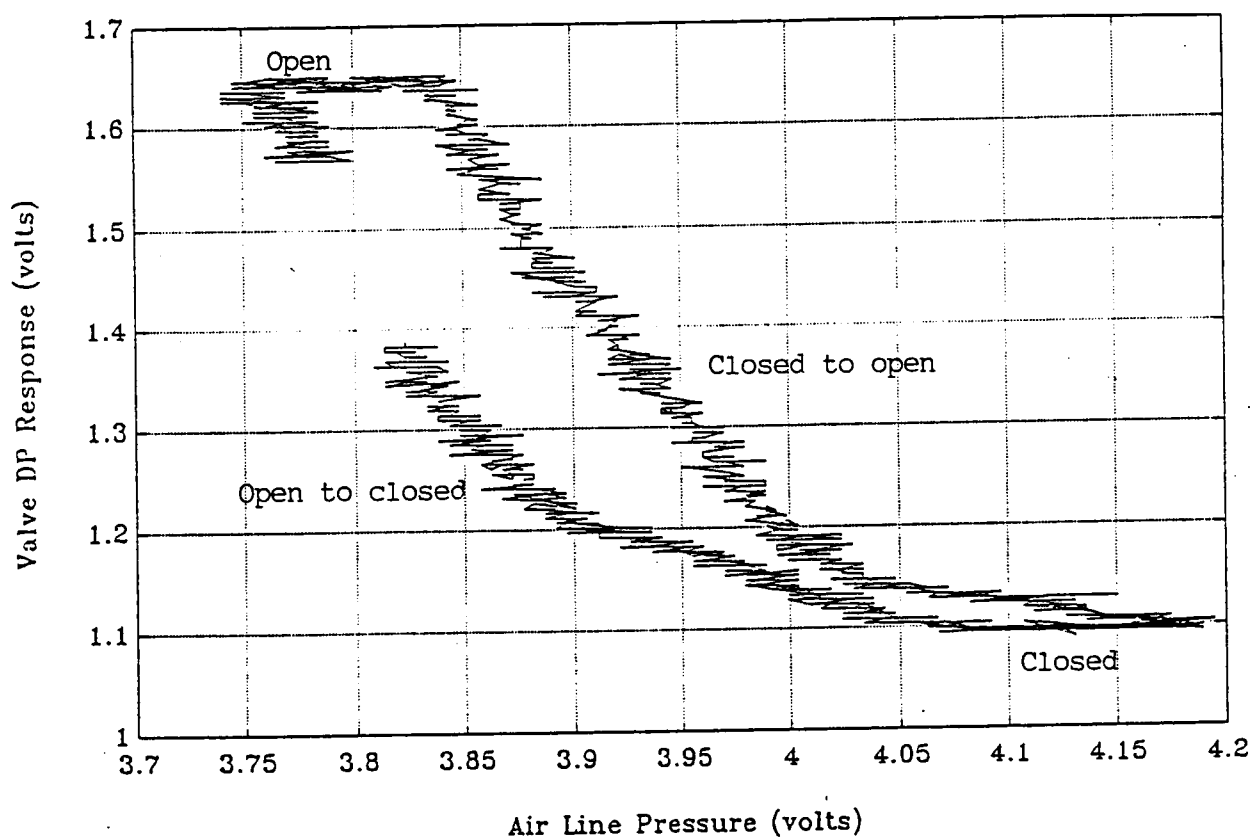


Figure 3.36. Air line pressure vs. cross valve pressure difference to a sinusoidal pressure input with increased stem friction.

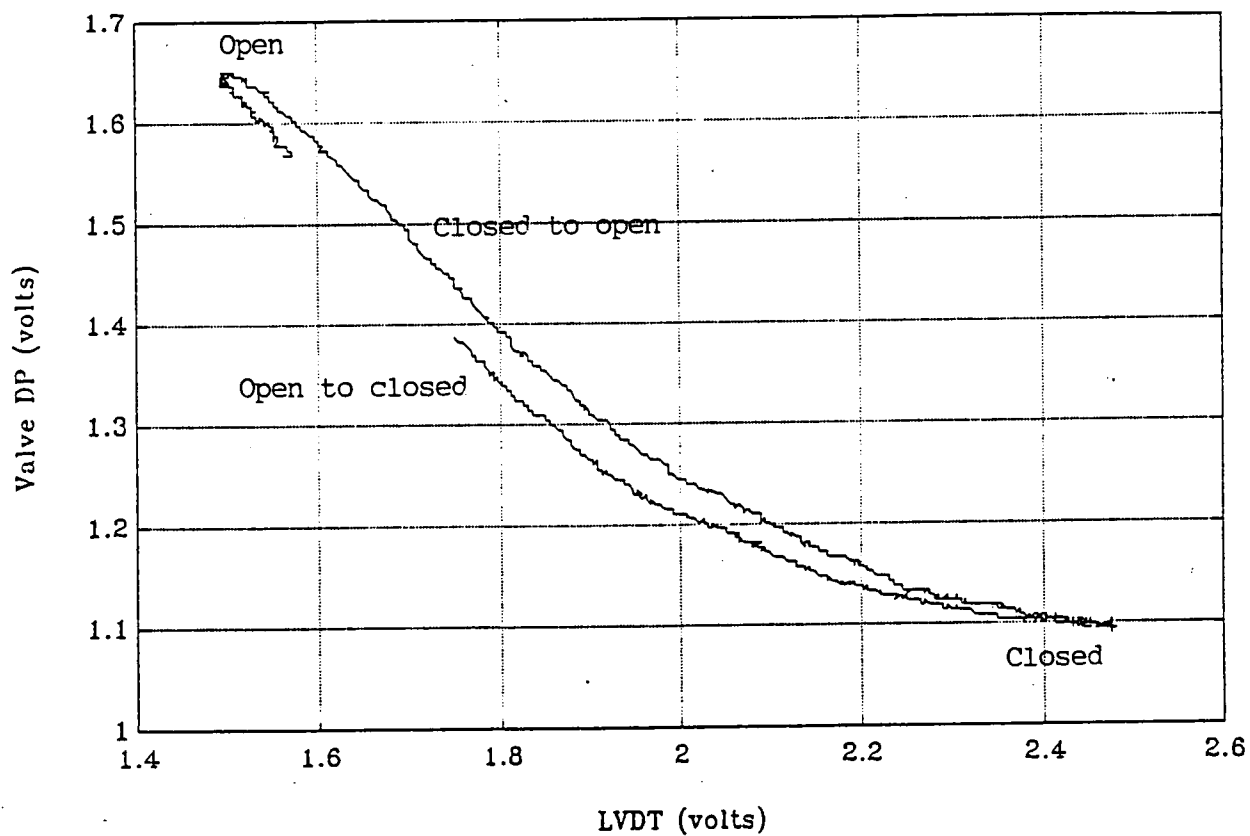


Figure 3.37. LVDT vs. cross valve pressure signal to a sinusoidal pressure input with increased stem friction.

change is clearly shown in the plot.

Figure 3.36 shows that relationship between the air line pressure and cross valve differential pressure signals with the increased stem friction. The hysteresis plots are completely distorted.

Figure 3.37 is the plot of LVDT vs. cross valve differential pressure signals to sinusoidal pressure input with increased stem friction. The overall behavior is similar to the other cases; but the discontinuity and the slopes are different. The effect of friction is clearly seen in this plot.

3.4 Summary of Valve Transient Analysis

Based on the valve transient analysis, the following conclusions may be made.

1. The valve dynamics are clearly represented by the stem motion and valve differential pressure variables during transient valve stroking. Nonlinear effects are seen in the hysteresis plots.
2. The time delay can be seen in the data plots.
3. Since ramp pressure input and sinusoidal pressure input are slowly applied, they give similar responses in the analysis.
4. Step response of the measurements are more sensitive to a small amount of air line leakage than responses to ramp and sinusoidal pressure inputs.
5. The air leakage can be detected by checking the responses to different inputs.
6. Friction variation is easier to detect as indicated by the measurements.
7. The three variables, valve differential pressure, stem motion, and air line pressure are sensitive to the anomalies, but not to the same extent.
8. The shape, slope, and the area of the plots of one variable versus another

can be used as a tool for detecting anomalies and for continuous monitoring.

No attempt has been made to quantify the variations in the stem motion and valve differential pressure signals either for the normal cases or for the cases with anomalies in the valve system. Some guidelines for implementing this approach for AOV condition monitoring are presented in Chapter 6.

CHAPTER IV

ANALYSIS OF FLUCTUATING COMPONENTS OF VALVE SIGNALS

4.1 Introduction

The random fluctuations of process signals often contain information about the dynamic characteristics of components that are being monitored. Baseline data may be acquired and a normal operational signature database may be generated from this information. Any changes in the behavior of these signatures during operation will provide an indication of degradation and changing process conditions.

The random signal analysis was performed to develop AOV on-line monitoring techniques. The valve dynamics were studied at different valve positions. For the test valve loop, the three measurements recorded were the air line pressure, the linear variable differential transformer (LVDT), and the valve differential pressure.

4.2 Data Acquisition System

A PC-based digital data acquisition system was used in the valve test project. The instruments to monitor the valve operation were interfaced with a personal computer using a patch panel and an analog-to-digital (A/D) board. A signal conditioning system was used before digitizing the data. This consists of high-pass

filters, low-pass filters, and amplifiers. The interfacing between the measurements and the PC is shown in Figure 4.1. The data acquisition is performed using an A/D (analog-to-digital) board and a data acquisition software. The measurements were sampled, analyzed, and plotted using both PC-MATLAB and MARSSA (Multivariate Autoregression modeling for System Surveillance and Analysis) software systems. The sampled data are studied by computing the signal variances, correlation functions, power spectral density functions, coherence functions, and autoregression modeling. The methodology used in this study is summarized below.

4.3 Methodology for Random Signal Analysis

The test valve dynamics were studied using various signatures derived from measurements. These studies include (a) time series, (b) correlation functions, (c) signal variances, (d) power spectral density functions, and (e) coherence functions. The studies are completed using PC- MATLAB and MARSSA software programs. In addition, univariate autoregression modeling is also applied to study the signal characteristics. The various methods are briefly discussed.

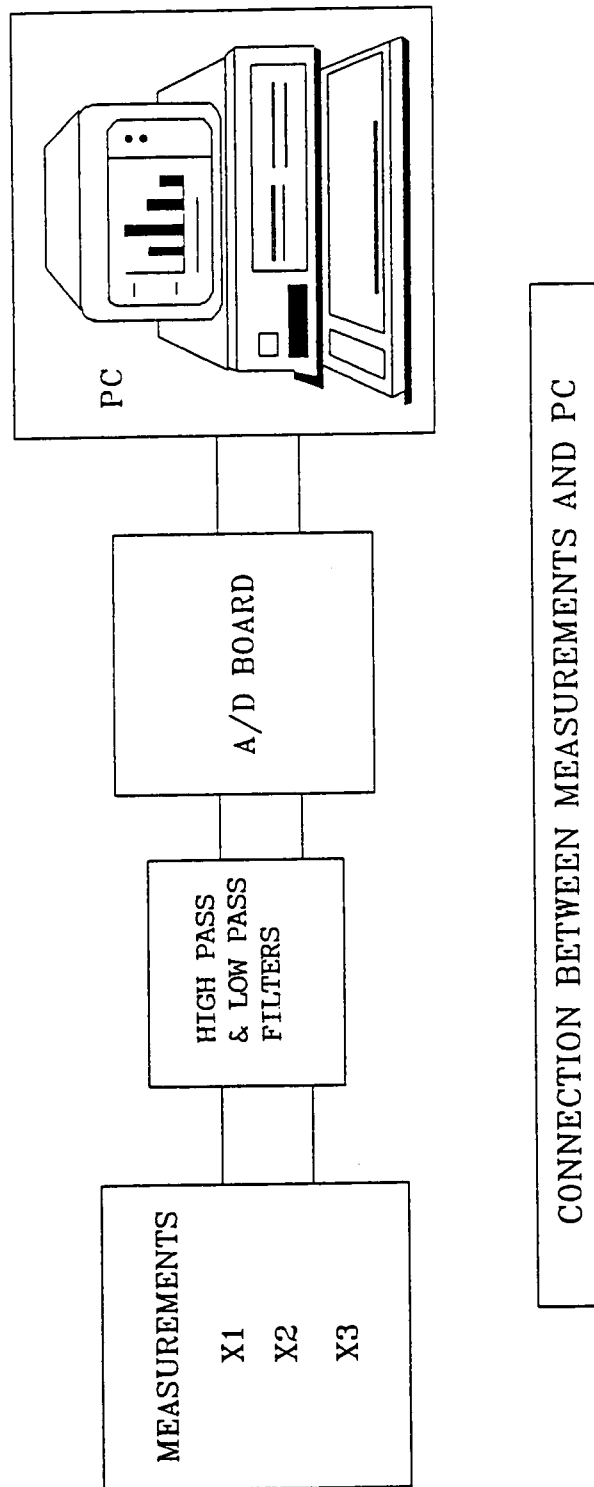


Figure 4.1. Interface between the measurements (sensors) and the data acquisition PC

4.3.1 Time-Domain Signatures

The sensors used in this study are air line pressure gauge, linear variable differential transformer, and valve differential pressure gauge. The sampled signals may be plotted for visual inspection. The amplitude probability density (APD) function [23] of the measurements gives an indication of the probability distribution of the signals. Some statistical parameters such as mean value, variance, skewness, and flatness were also calculated and displayed.

The amplitude probability density (APD) function of a random signal is estimated using discrete samples of the signal. For a true Gaussian process the APD function has the form

$$APD(X_i) = \frac{1}{\sqrt{2\pi C_x(0)}} e^{-\frac{(X_i - X_{mean})^2}{2C_x(0)}} \quad (4.1)$$

where X_{mean} is the mean value of the random process and $C_x(0)$ is the variance of the random process [24]. These two parameters determine the distribution uniquely. The autocorrelation of a stationary signal X is computed from

$$C_x(k) = \frac{1}{N} \sum_{i=1}^{N-k} [X(i+k) - X_{mean}] \times [X(i) - X_{mean}] \quad (4.2)$$

where the correlation lag $k=0, 1, 2, \dots$, and N is the number of samples. The mean value X_{mean} and the variance of a signal are calculated from

$$X_{mean} = \frac{1}{N} \sum_{i=1}^N X(i) \quad (4.3)$$

$$\sigma_x^2 = \frac{1}{N} \sum_{i=1}^N (X(i) - X_{mean})^2 \quad (4.4)$$

The skewness (G_{skew}) and flatness (G_{flat}), which are higher order moments of the random process, are also computed. These are defined as follows

$$G_{skew} = \frac{1}{N\sigma_x^3} \sum_{i=1}^N (X(i) - X_{mean})^3 \quad (4.5)$$

$$G_{flat} = \frac{1}{3N\sigma_x^4} \sum_{i=1}^N (X(i) - X_{mean})^4 \quad (4.6)$$

where X_{mean} is the mean value of signal X (for a noise signal it is close to zero). The standard deviation σ_x indicates the deviation or spread of the random process with respect to its mean value. The parameters G_{skew} and G_{flat} represent the quantitative measures of the extent of deviation of the actual APD from a Gaussian distribution. For an ideal Gaussian distribution $G_{skew}=0$ and $G_{flat}=1$.

4.3.2 Correlation Function

The random signals are analyzed using signal correlation analysis. More attention is given to autocorrelation functions of random signals.

The autocorrelation function of a stationary signal at a lag k is defined as follows.

$$C_{xx}(k) = \frac{1}{N} \sum_{i=1}^{N-k} [X(k+i) - X_{mean}] \times [X(i) - X_{mean}] \quad (4.7)$$

When the number of data points (N) is very large, it is necessary to calculate the average by dividing the entire sample into several blocks of data points.

4.3.3 Power Spectral Density Function

The auto power spectral density (APSD) functions of the measured time series are estimated by transforming and averaging over discrete time samples. In principle, these functions are the discrete Fourier transforms [23] of the corresponding auto and cross correlation functions defined as:

$$S(k) = \Delta t \sum_{i=0}^{N-1} C(i) e^{-j2\pi i k / N}, \quad (4.8)$$

where Δt is the sampling time interval. The APSD function is estimated in a simple manner. It is calculated from the fast Fourier transforms (FFT) of the corresponding digital time series, averaged over N_B number of data blocks. The APSD is given by

$$S_{xx}(k) = \frac{\Delta t}{MN_B} \sum_{i=1}^{N_B} \left\{ \sum_{m=1}^M X_m^{(i)} e^{-j2\pi m k / M} \sum_{n=1}^M X_n^{(i)} e^{-j2\pi n k / M} \right\} \quad (4.9)$$

where k is the discrete frequency value. The number of data points per block is denoted by M , and it must be a power of two. The number of independent spectral values is $M/2$, and the spectral values are calculated for $k=0, 1, \dots, M/2$.

4.3.4 Coherence Function

The ordinary coherence function between signal i and signal j is defined as follows:

$$COH_{ij}(f) = \frac{|S_{ij}(f)|^2}{S_{ii}(f)S_{jj}(f)} . \quad (4.10)$$

The $COH_{ij}(f)$ function has values between zero and unity, and indicates the commonality of the two signals as a function of frequency. $S_{ii}(f)$ is the auto power spectral density of signal i and $S_{ij}(f)$ is the cross power spectral density between signals i and j .

4.3.5 Autoregression Modeling of a Random Signal

The time series modeling of random signals is discussed in this section. The model can be used to forecast future values of a time series from current and past values, to determine the transfer function of a system, and its response to a given input. The power spectrum of the random signal can also be estimated using the autoregression model.

If $x(t)$ is a sequence of measurements of a random process and if $v(t)$ is a noise sequence causing the $x(t)$, then the finite order autoregression (AR) process is represented by the linear transformation

$$x(t) = \sum_{i=1}^n a_i x(t-i\Delta t) + v(t) \quad (4.11)$$

The power spectrum of the random process is given by [26]

$$S_x(f) = |H(f)|^2 S_w(f) \quad (4.12)$$

where: $S_x(f)$ = power spectrum of $x(t)$

$S_w(f)$ = power spectrum of $v(t)$

$H(f)$ = linear transformation from the AR model.

Thus the estimated power spectrum of a random process is given by

$$\hat{S}_x(f) = \frac{\hat{\sigma}_v^2 \Delta t}{\left| 1 - \sum_{k=1}^n \hat{a}_k e^{-j2\pi f k \Delta t} \right|^2}, \quad |f| \leq \frac{1}{2\Delta t} \quad (4.13)$$

where $S_w(f) = \sigma_v^2 \Delta t$, f is frequency in Hz, and Δt is sampling time in second. The noise variance is estimated as

$$\sigma_v^2 = \frac{1}{N} \sum_{k=1}^N v_k^2 \quad (4.14)$$

where v_k is the residual sequence of the random process.

Once the AR model of a random process is developed, the impulse response may be computed. From this, both step and ramp responses of the modeled system can be estimated by proper integration.

4.4 Results of Random Signal Analysis

The random signal analysis was performed for five positions of the test valve: fully open, 84% open, 64% open, 20% open, and fully closed. Appropriate signal conditioning was performed to filter unwanted frequencies from the signals. The cut-off frequency of high-pass filter was set at 0.1 Hz, and of low-pass filter at 40 Hz. The signals were sampled at a frequency of 200 Hz, and each signal contained 10,000 points. The measurements were air line pressure, valve stem position, and differential pressure across the test valve. The following functions and parameters were computed: variances, correlation functions, and power spectral density functions.

The above computation was also carried out for the case of valve anomalies at different valve positions. The introduced anomalies were diaphragm and air line leakage and stem friction changes. The problems, such as broken air line, lost bolt on the diaphragm support, and pressure variations were included in the air leakage anomaly. Bent stem and binding of the stem and its packing were simulated as a change in the friction force. The problem of valve seating was not simulated due to the difficulty of introducing this anomaly.

4.4.1 Time Series Plots

Time series plots represent the amplitude of the measurements in voltage. They show the signal amplitudes as a function of time. Typical sampled measurements of air line pressure, LVDT, and valve differential pressure are plotted

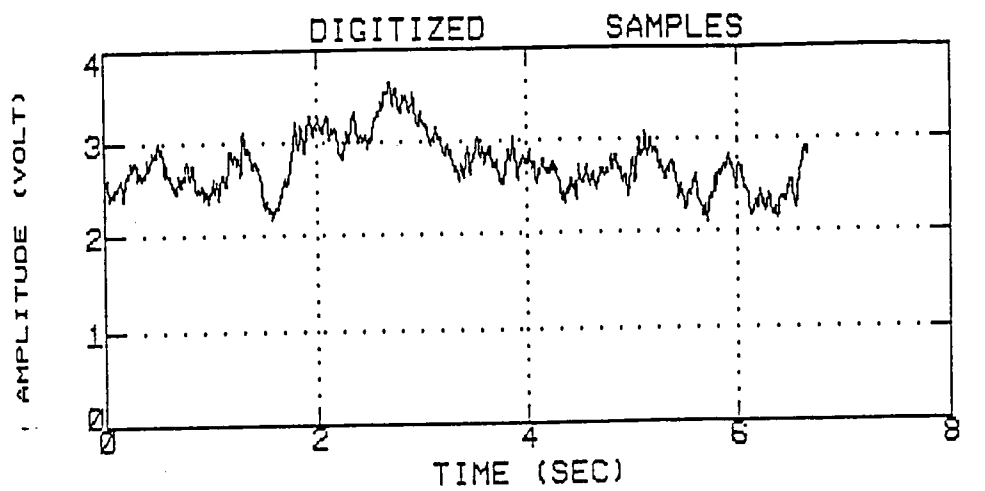
at valve normal operations in Figures 4.2 through 4.4. For each signal, there are six plots (a-f), corresponding to different valve positions. Figure 4.2 shows that the air line pressure signal is directly related to the valve stem motion. The fluctuations of the air line pressure signal are the strongest at the 100% open position, and decrease gradually as the valve approaches the closed position.

Figure 4.3 shows that the stem moves faster downwards than upwards. This is because the spring forces dominate the valve dynamics when no pressure is acting on the diaphragm. On the other hand, when a pressure is applied to the valve actuator, the pressure force moves the stem slowly. Skewness of LVDT signal is much greater than that of the air line pressure. This is because the fluctuations of the valve stem motion have a direct effect on the LVDT signal and indirectly influence the air line pressure signal. The largest fluctuation of LVDT signal is seen at valve 84 % open position. As the stem moves toward its fully closed position, the signal fluctuation increases again, but the magnitude of the signal is much lower than that at 84 % open position.

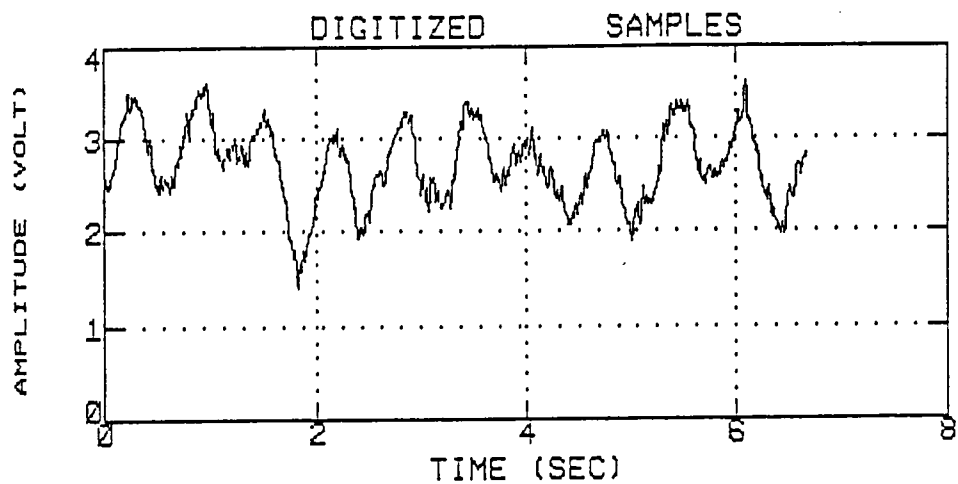
Figure 4.4 shows that the differential pressure signal is not affected by the valve stem dynamics when the stem motion is limited. The signal is very symmetric around its mean value. As the valve is gradually closed, the fluctuation of the signal steadily increases.

4.4.2 Variances of Measurements

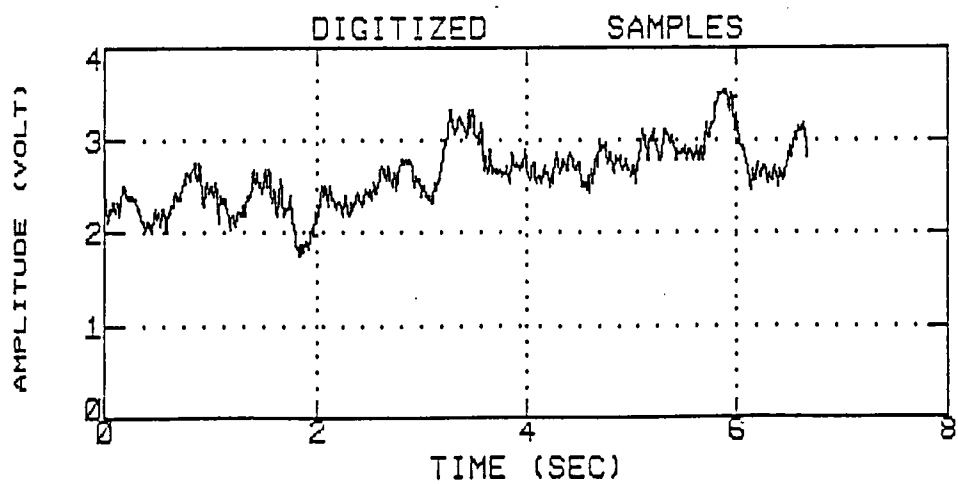
The variances of the measurements show signal fluctuations at different



(a). at 100% open

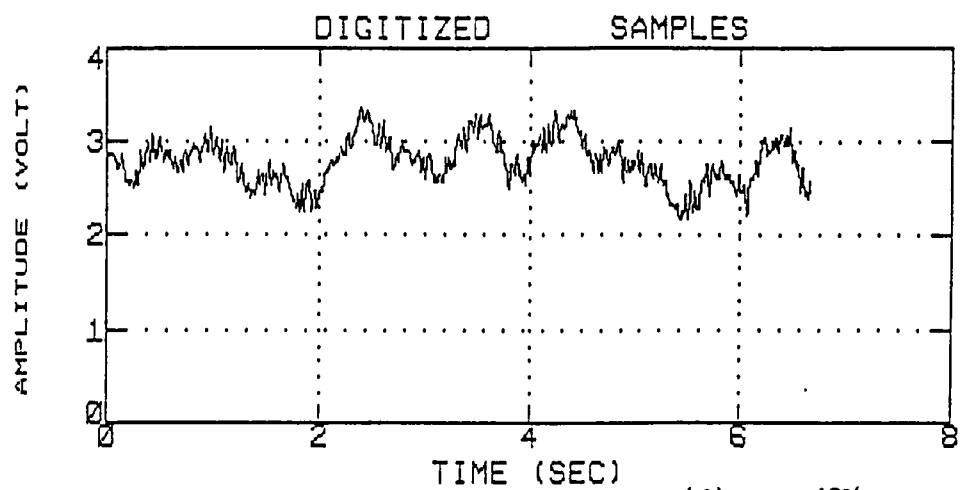


(b). at 84% open

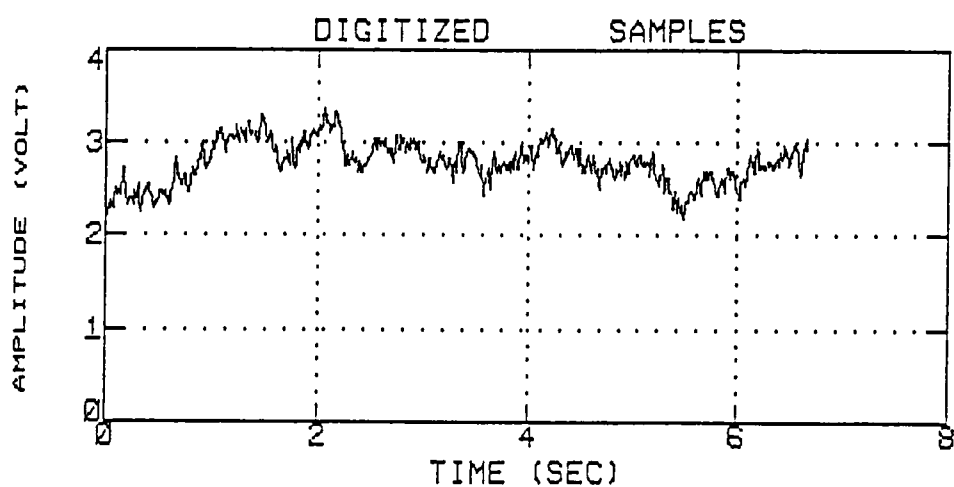


(c). at 62% open

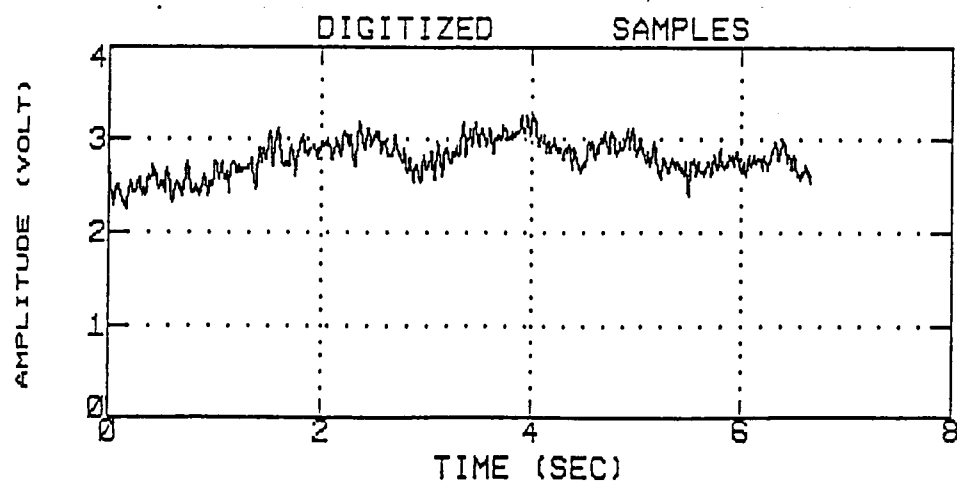
Figure 4.2. Time series plot of air line pressure signal at normal operation.



(d). at 40% open

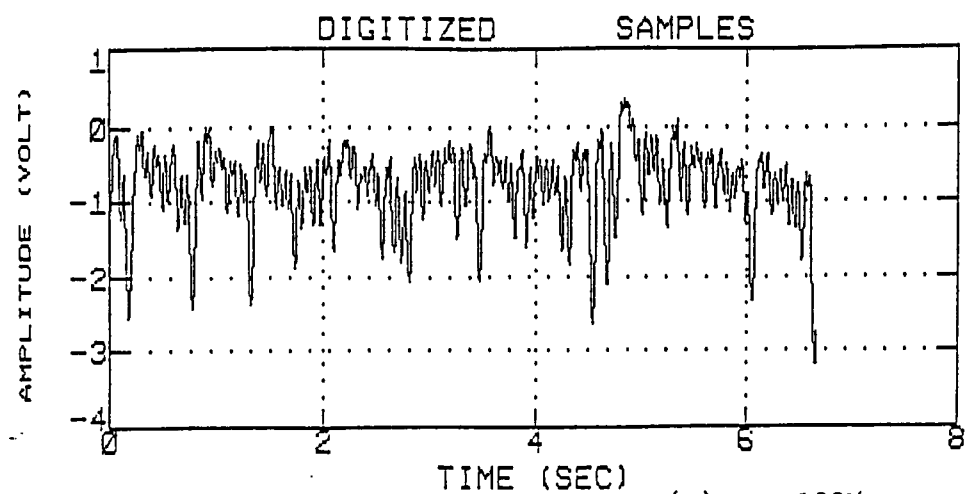


(e). at 18% open

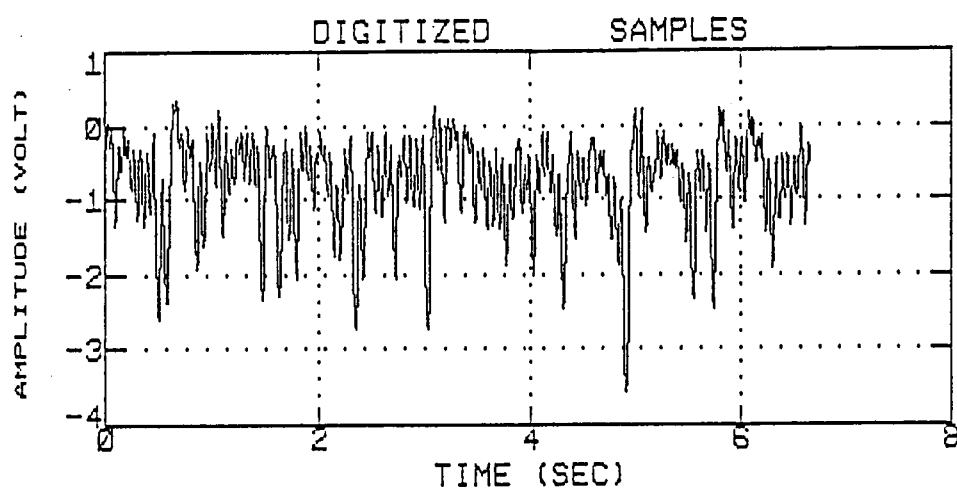


(f). at fully closed

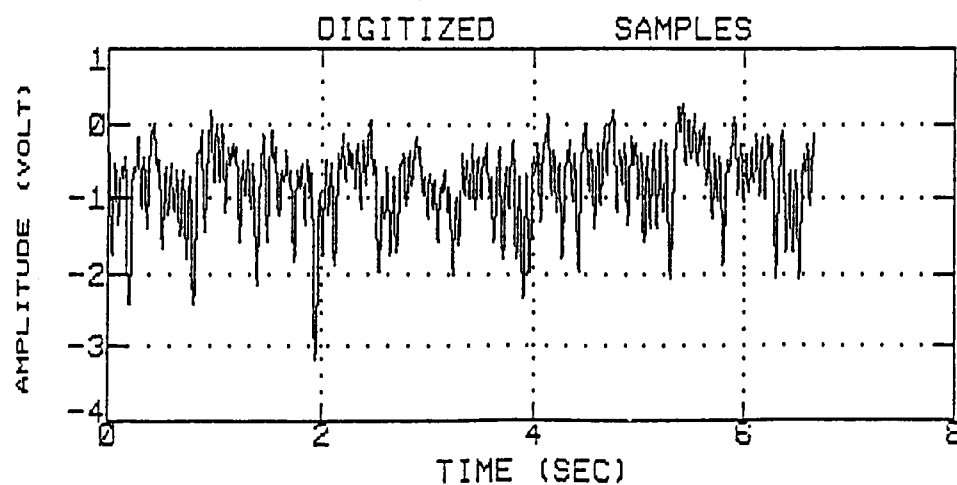
Figure 4.2. (Continued)



(a). at 100% open

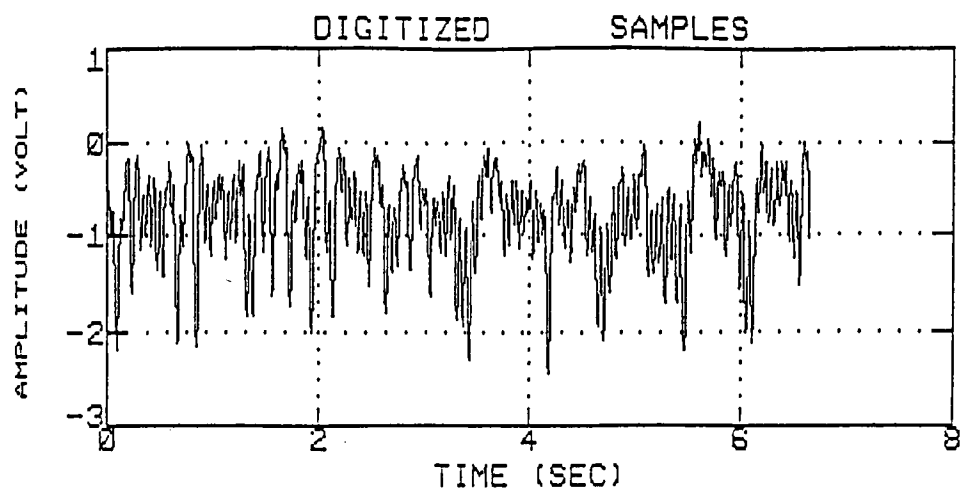


(b). at 84% open

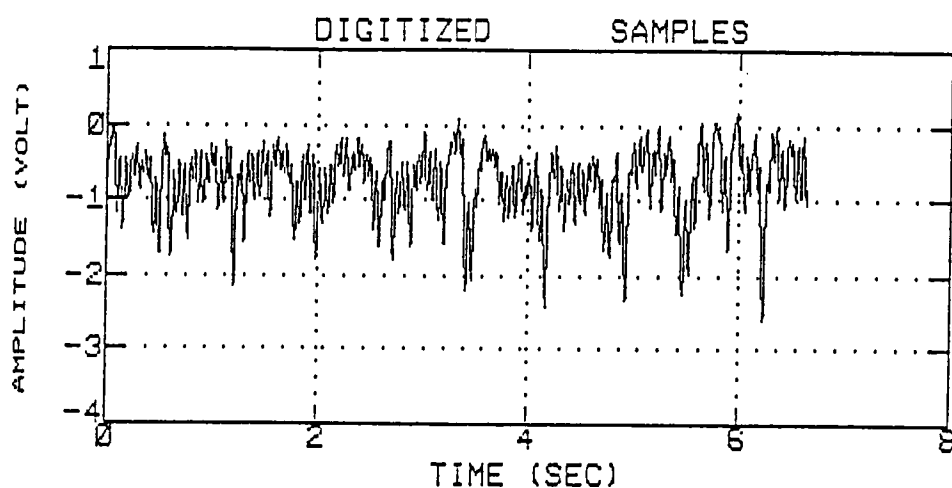


(c). at 62% open

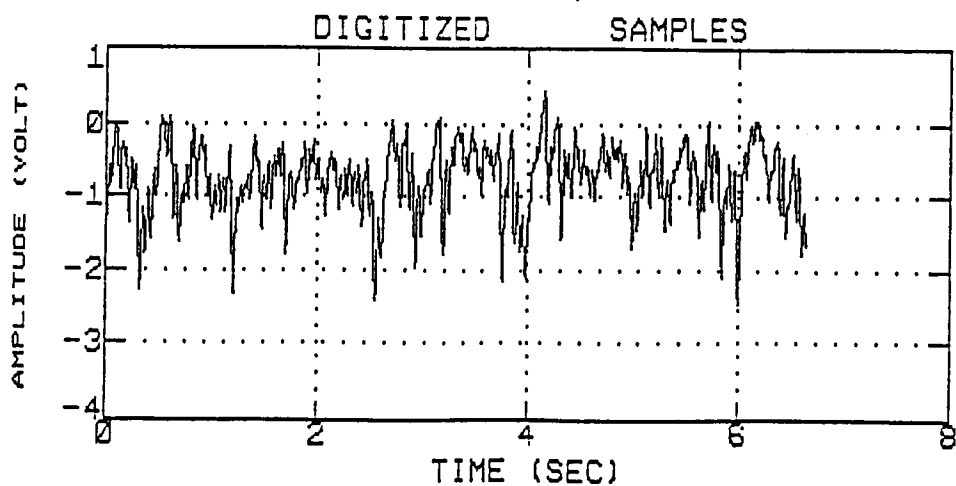
Figure 4.3. Time series plot of stem position signal (LVDT) at normal operation.



(d). at 40% open

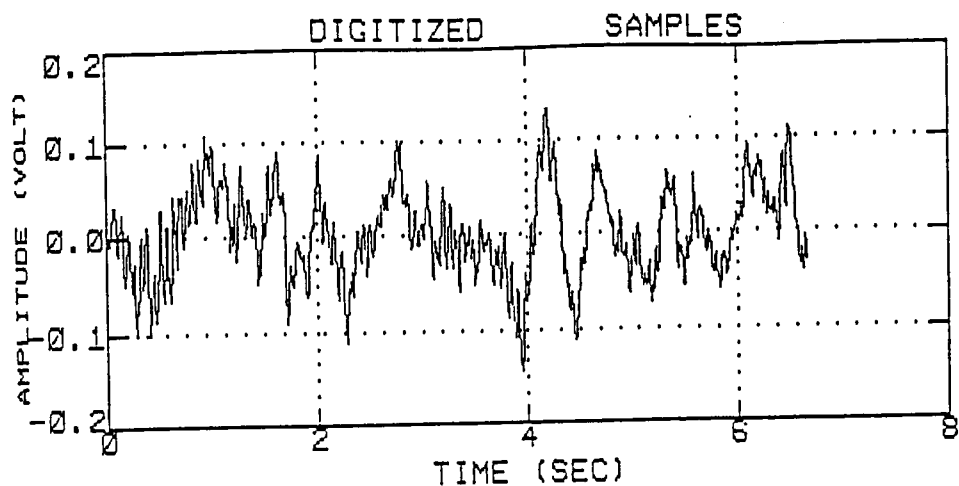


(e). at 18% open

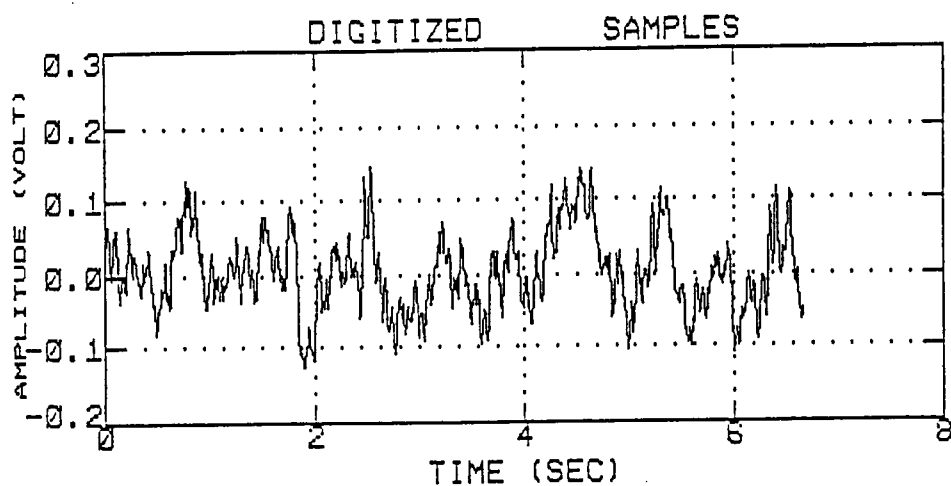


(f). at fuully closed

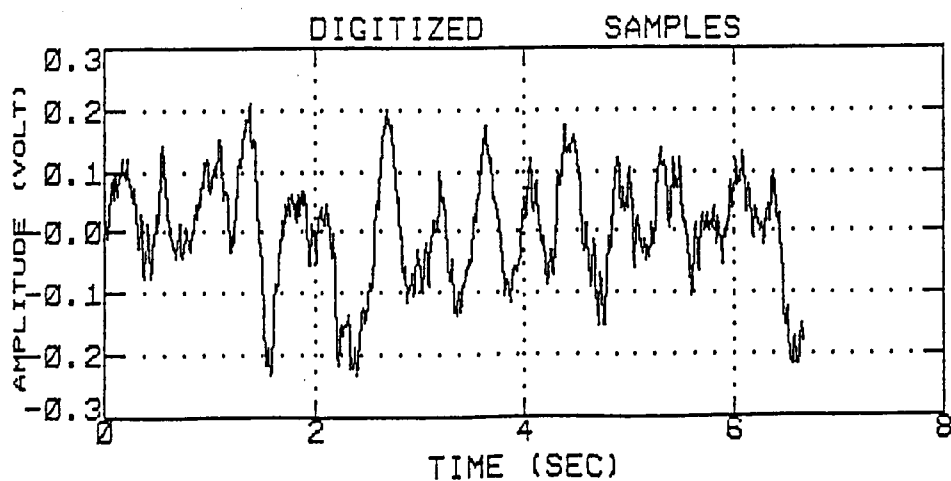
Figure 4.3. (Continued)



(a). at 100% open

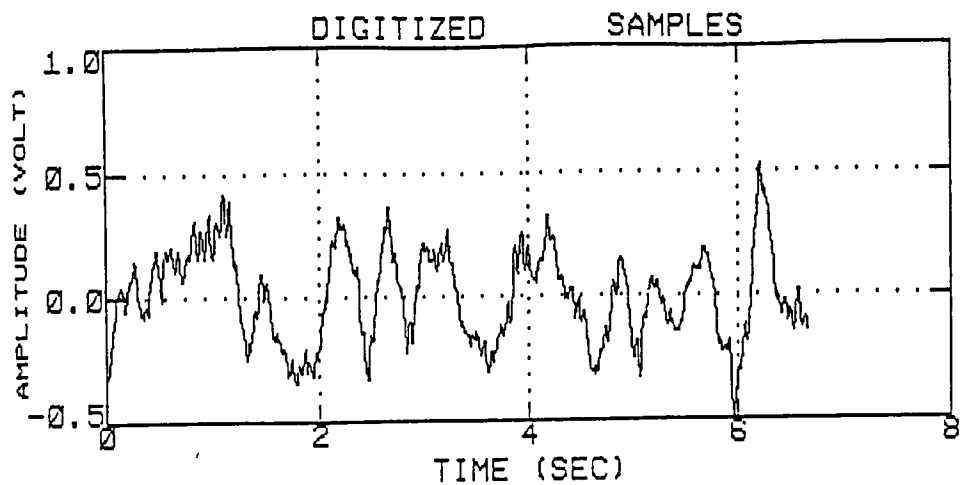


(b). at 84% open

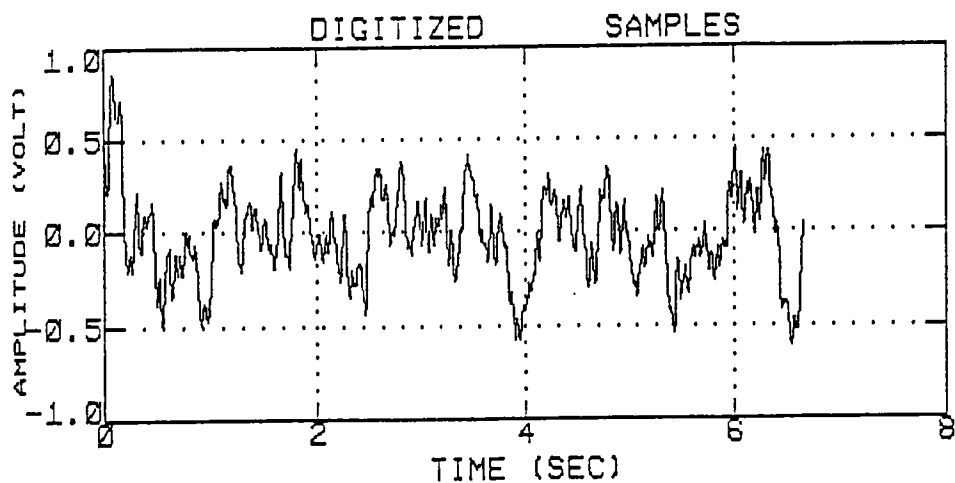


(c). at 62% open

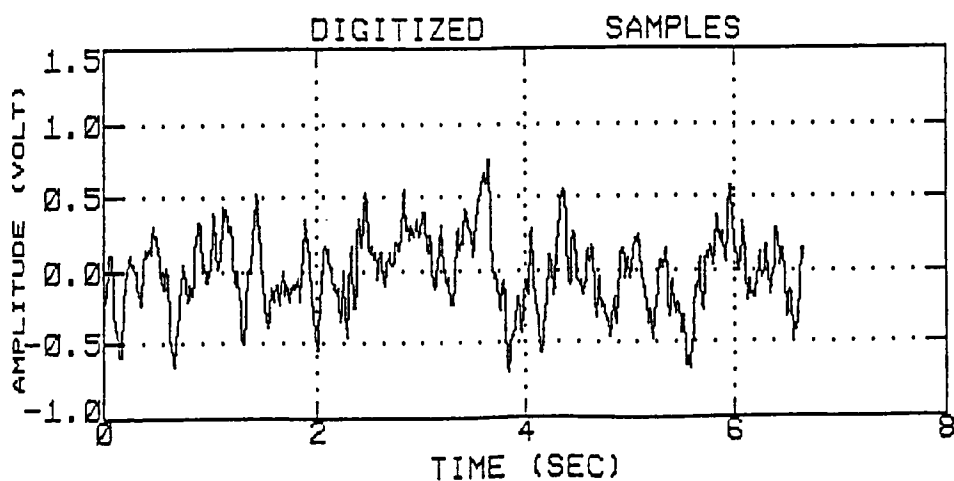
Figure 4.4. Time series plot of valve differential pressure signal at normal operation.



(d). at 40% open



(e). at 18% open



(f). at fuully closed position

Figure 4.4. (Continued)

operations. Both normal operation and anomaly conditions are studied. The variances of the measurements are plotted for different stem positions.

4.4.2.1 Variance of Measurements at Normal Operation

The relationship between the stem travel and the variances of random signals at normal operation is shown in Figure 4.5.

The variance of the air line pressure signal decreases gradually as the valve position changes from fully closed to fully open. At the valve fully closed position, the stem vibrations are maximum causing the largest variation of the air line pressure signal. However, as the valve is opened (more than 40 %), the effect due to the stem position decreases.

The variance of LVDT signal increases as the valve is gradually closed. As the valve is closed, the flow impact force increases, so the variance of the signal increases. It reaches its largest at the valve 18 % open position. As the valve is further closed, the flow impact force continues to increase, but the valve plug has started to seat, so the vibration effect is reduced. The variance of the stem travel signal is minimum at the valve fully closed position.

The variance of cross valve differential pressure signal increases as the valve is gradually closed. The variance of the signal is maximum at the valve fully closed position.

Variances of Measurements

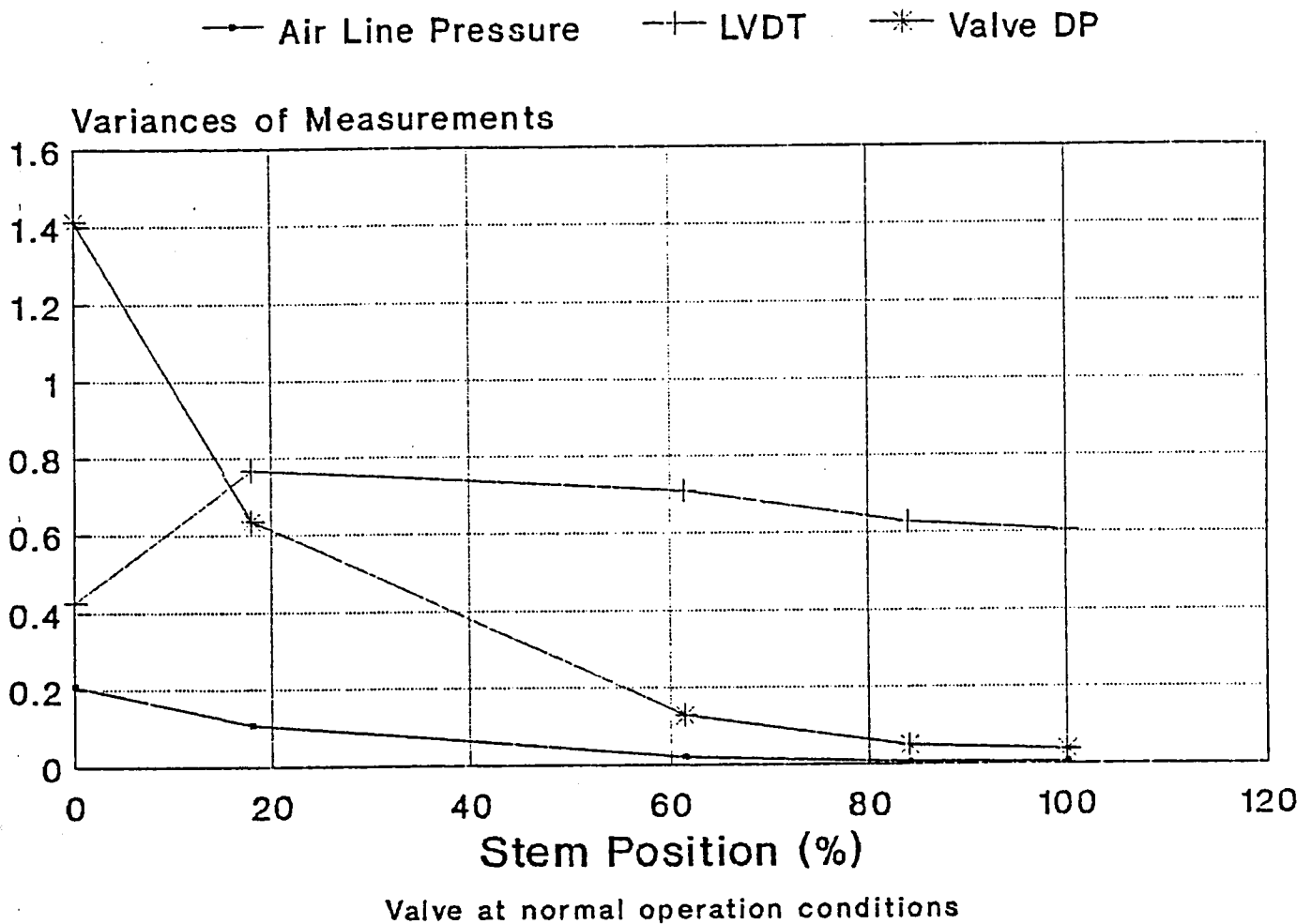


Figure 4.5. Relationship between stem travel and variance of random signals at normal operation.

4.4.2.2 Variance of Measurements With Air Leakage

The relationship between the stem travel and the variance of random signals with air leakage is shown in Figure 4.6.

The air line signal variance with air leakage decreases gradually as the valve position changes from fully closed to fully open. The magnitude of the variance with air leakage is smaller than that at normal operation.

The variance of LVDT signal is affected considerably by the air leakage anomaly. The maximum variance occurs at valve 62 % open position, and the magnitude of the variance increases. This is because, as the pressure forces acting on the diaphragm become small, the spring forces result in increased vibration of the valve stem. At the valve fully open and fully closed positions, the variances of the stem travel will not be affected by the air leakage.

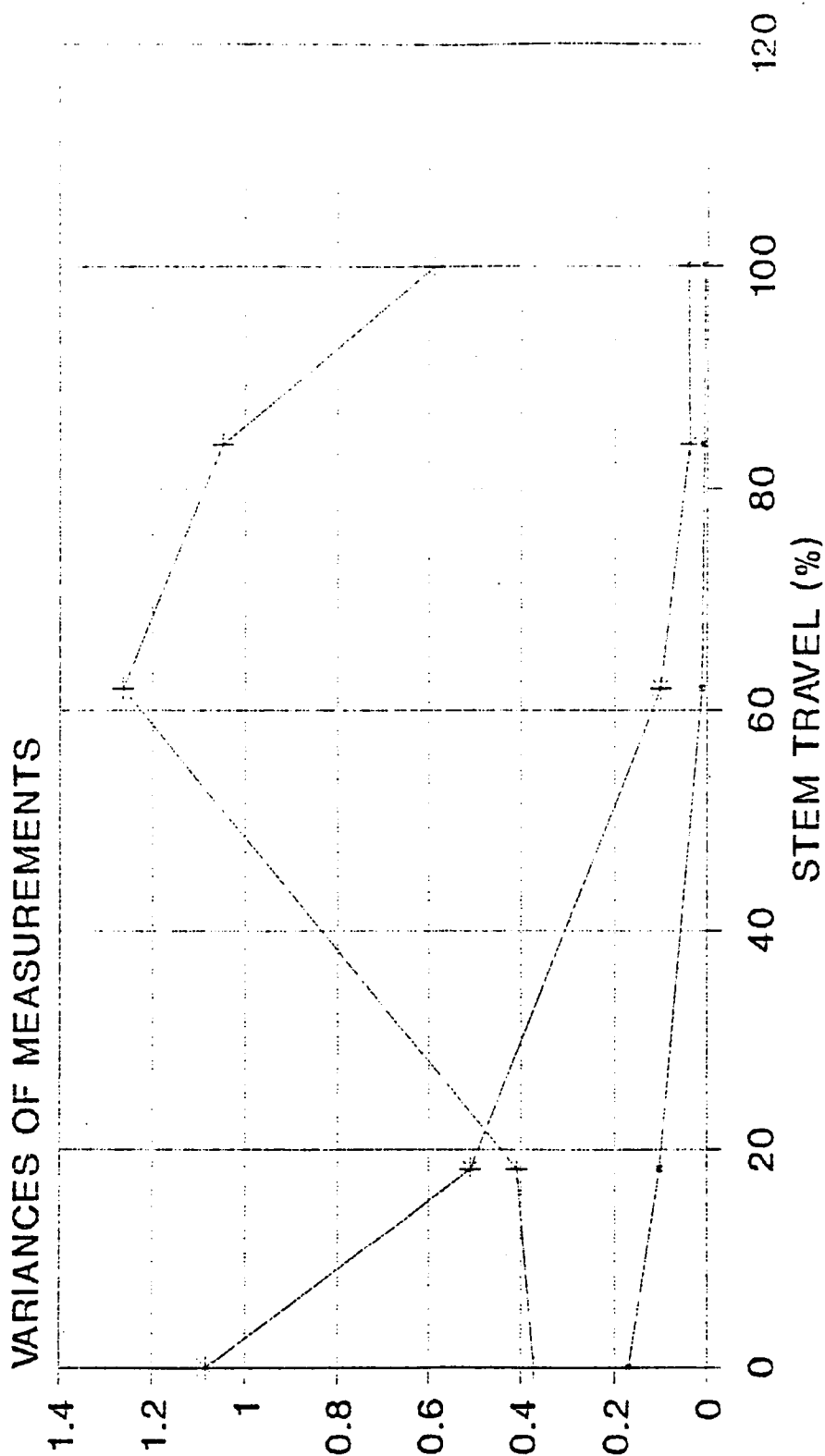
The variance of valve differential pressure signal with air leakage increases as the valve is gradually closed. The variance of the signal has its maximum at the fully closed position.

4.4.2.3 Variance of Measurements with Increased Stem Friction

The relationship between the stem travel and the variances of random signals with friction variations is shown in Figure 4.7.

The air line pressure signal variance with friction variation decreases gradually as the valve travels from fully closed to fully open position. The magnitude of the

--- AIR LINE PRESSURE - - - LDVT - - - VALVE DP

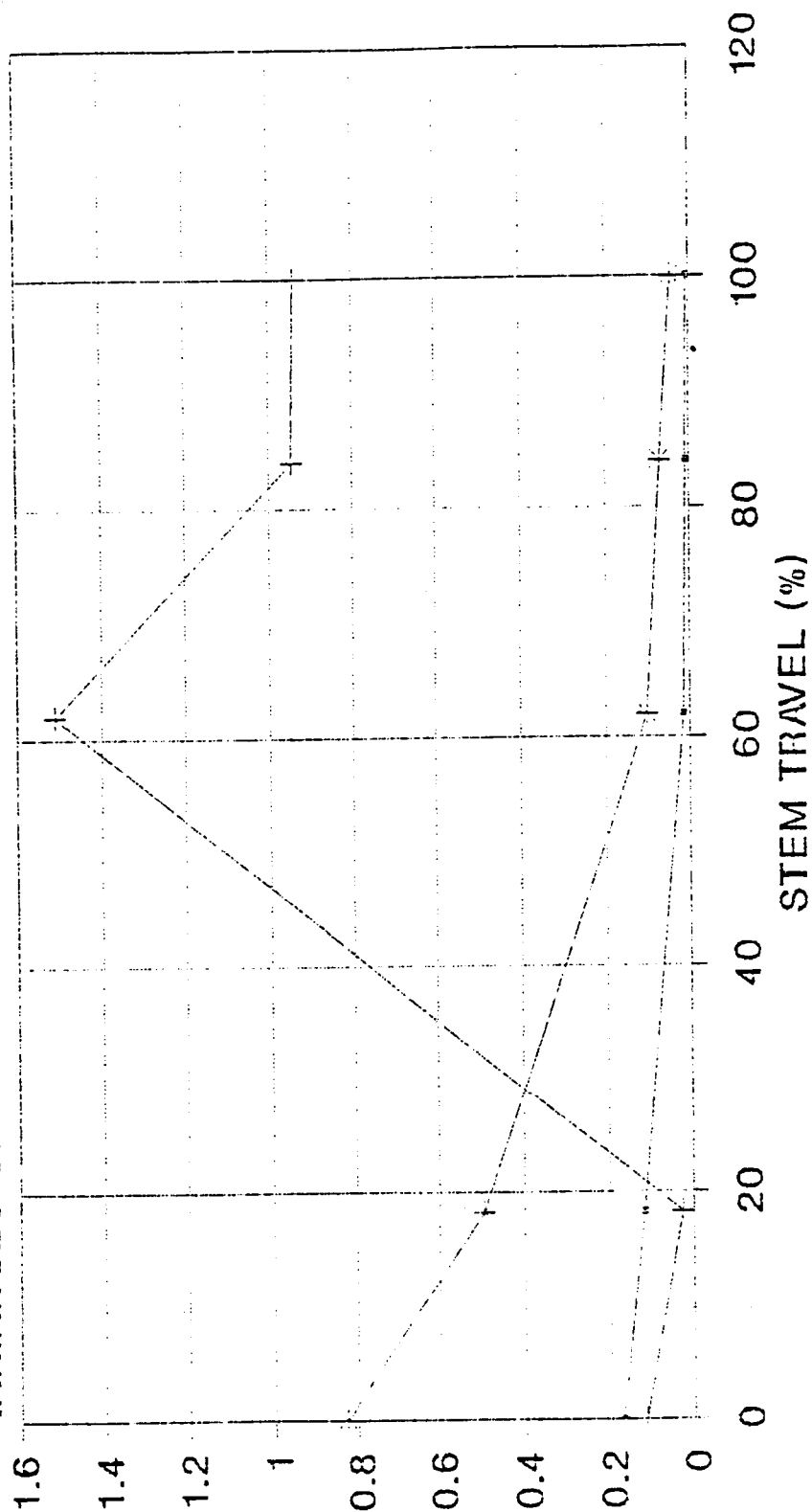


STEM TRAVEL=1.6 CM & FULL FLOW=5.614 G/M

Figure 4.6. Variance of measurements with air leakage

--- AIR LINE PRESSURE -|- LDVT -|- VALVE DP

VARIANCES OF MEASUREMENTS



STEM TRAVEL=1.6 CM & FULL FLOW=5.614 G/M

Figure 4.7. Variance of measurements with increased stem friction

variance of the air line pressure signal is almost the same as those for normal operation. The reason for this is that the air line pressure is the input to the system, and hence it will not be affected by increased stem friction.

The variance of the LVDT signal is affected considerably by this anomaly. At valve 62 % open position, the magnitude of the variance is greater than that for both normal operation and for the case of air leakage. The shape of the variance of the LVDT signal is also changed due to the friction effect.

The variance of valve differential pressure signal with friction variation increases as the valve is gradually closed. The variance of the signal has its maximum at valve fully closed position. The shape matches with the normal operation condition and with the case of air leakage, but the magnitude of the variance is less than that for the case of air leakage. The increased friction on the stem restricts its motion and hence the variance of the cross valve pressure signal decreases.

4.4.3 Correlation Functions of Fluctuating Signals

The autocorrelation function shows how the signal is related at different time lags. Typical autocorrelation functions of the measurements at valve half open position are shown in Figures 4.8 through 4.10.

Figure 4.8 shows the normalized auto correlation function of air line pressure signal at valve half open position. At valve fully open and fully closed positions, the air line pressure signal has a very small correlation after a lag of 0.4 second. A damped oscillation behavior is seen at other valve positions.

SIGNAL NAME 1 air line
CORREL. FUNCT. FILE: ran3.oor
CORREL. AT ZERO LAG: 2.203×10^{-2}

CHANNEL NO.: 1

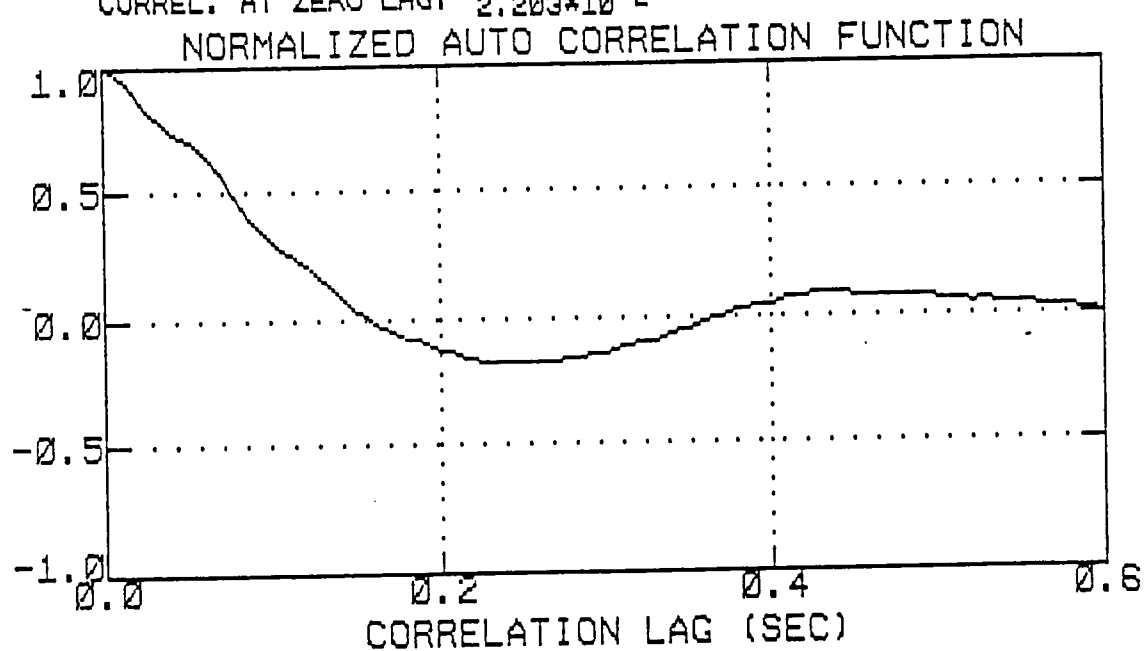


Figure 4.8. Correlation function of air line pressure signal. (at 64% open position)

SIGNAL NAME 1 lvdv

CHANNEL NO.: 2

CORREL. FUNCT. FILE: ran3.cor

CORREL. AT ZERO LAG: 7.495×10^{-1}

NORMALIZED AUTO CORRELATION FUNCTION

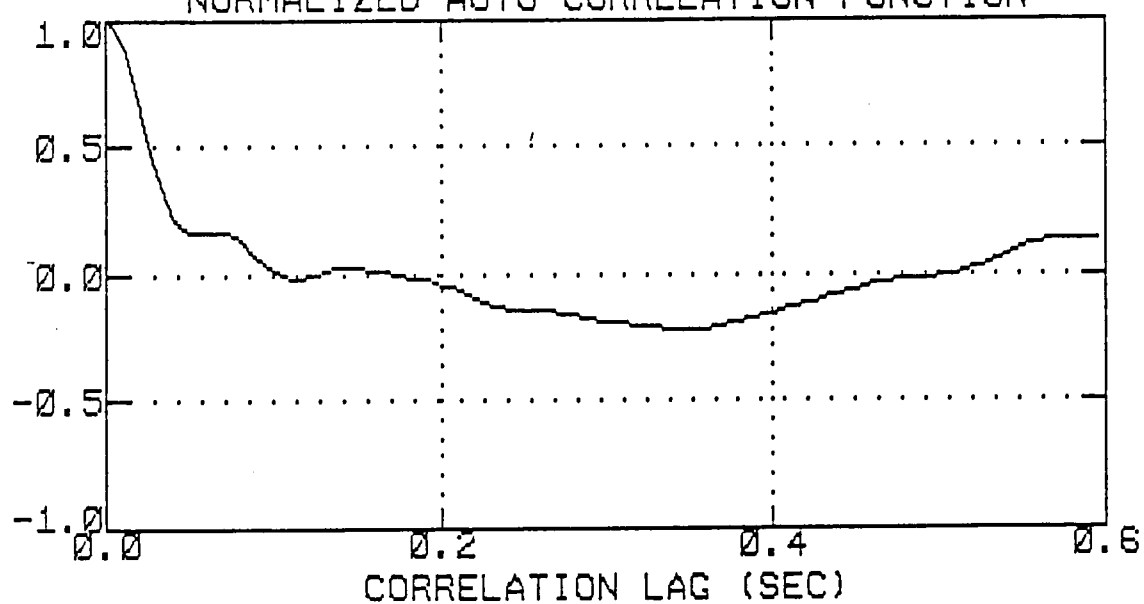


Figure 4.9. Correlation of stem motion signal (at 64% open position)

SIGNAL NAME 1 valve dp
CORREL. FUNCT. FILE: ran3.oor
CORREL. AT ZERO LAG: 1.347×10^{-1}

CHANNEL NO.: 3

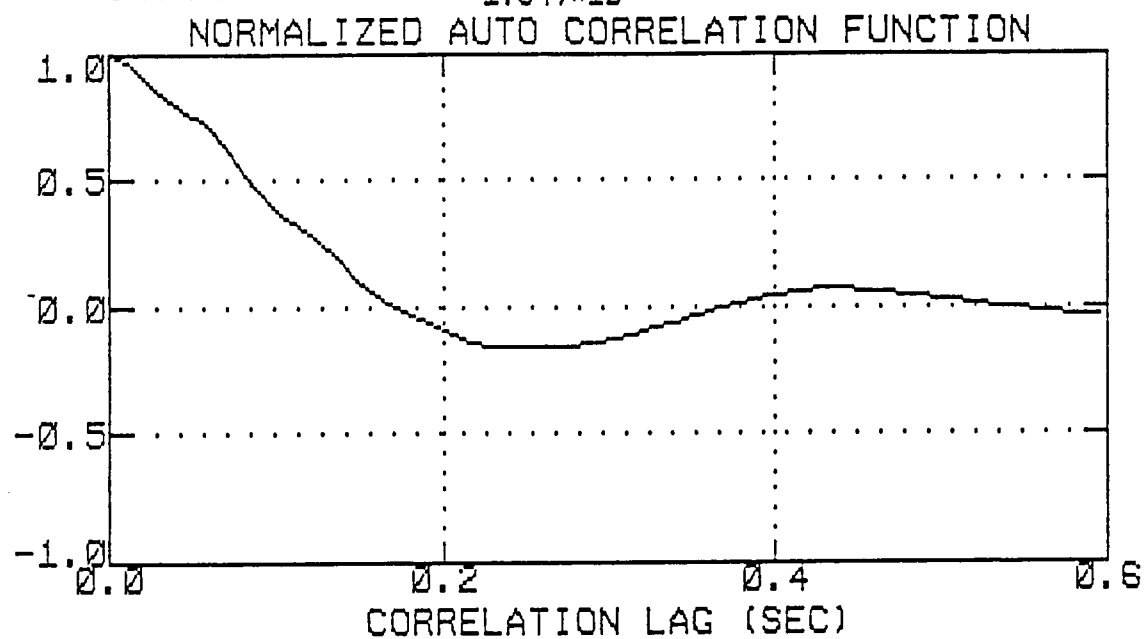


Figure 4.10. Autocorrelation of valve differential pressure signal (at 64% open position).

Similar plots of normalized autocorrelation functions are shown in Figures 4.9 and 4.10 for the LVDT and valve differential pressure signals, respectively.

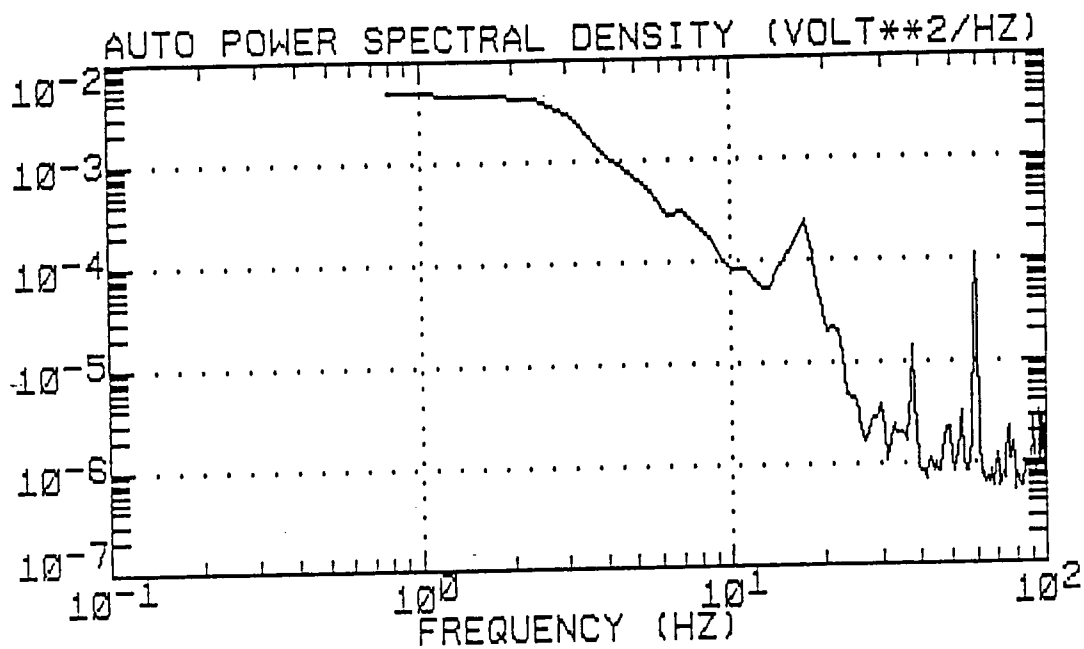
The similarity between the air line pressure and cross valve pressure drop is due to the use of similar pressure transmitters.

4.4.4 Power Spectral Density Functions of Fluctuating Signals

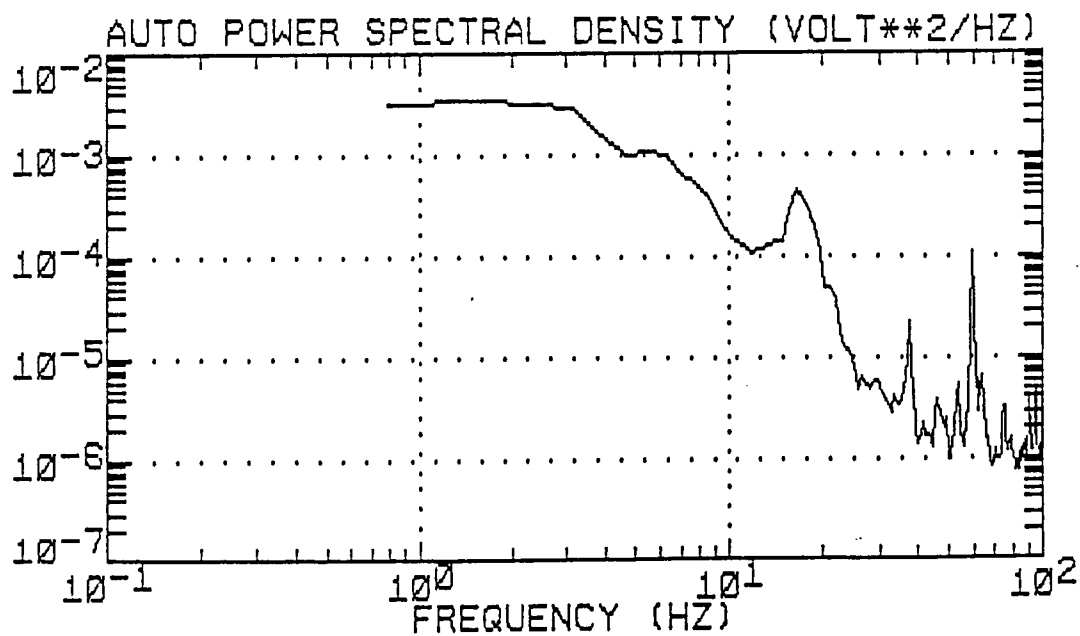
The power spectral density functions show the frequency components (or bandwidth) of the signals. In this section, auto power spectral density functions (APSD) are presented. Typical APSDs of the measurements at valve normal operation are plotted in Figure 4.11 through 4.13.

Figure 4.11 shows the APSD of air line pressure signal at different valve positions. Three peaks are noticed in the APSDs of the air line pressure signal. The frequencies of these peaks are approximately 18, 38, and 60 Hz. The 60 Hz frequency is due to electrical pick up and pump rotation. The system pump speed is 3600 rpm. As the valve approaches the closed position, the magnitudes of low frequency and 18 Hz peaks increase. This indicates that the air line pressure signal fluctuates more at both low frequency and at about 18 Hz when the valve is more closed. However, the magnitudes of other two peaks do not change as the valve position changes.

Figure 4.12 shows that the LVDT signal is a wide band signal compared to the air line pressure signal. The spectrum has a higher break frequency. The 60 Hz frequency exists due to the electrical pick up and pump rotation. The APSD shape

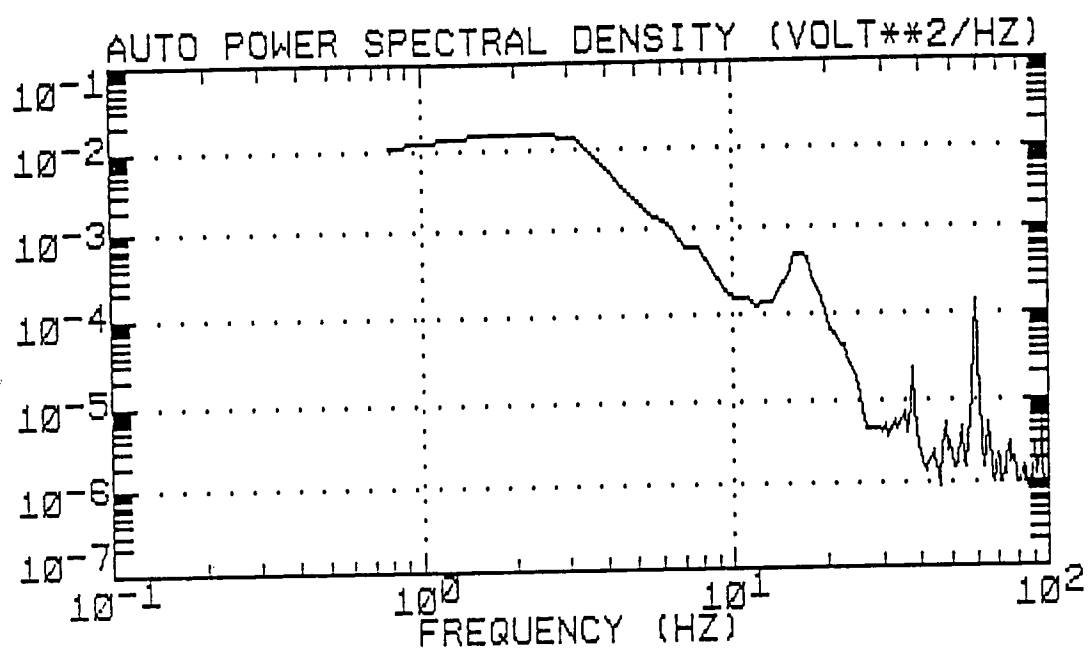


(a). at 100% open

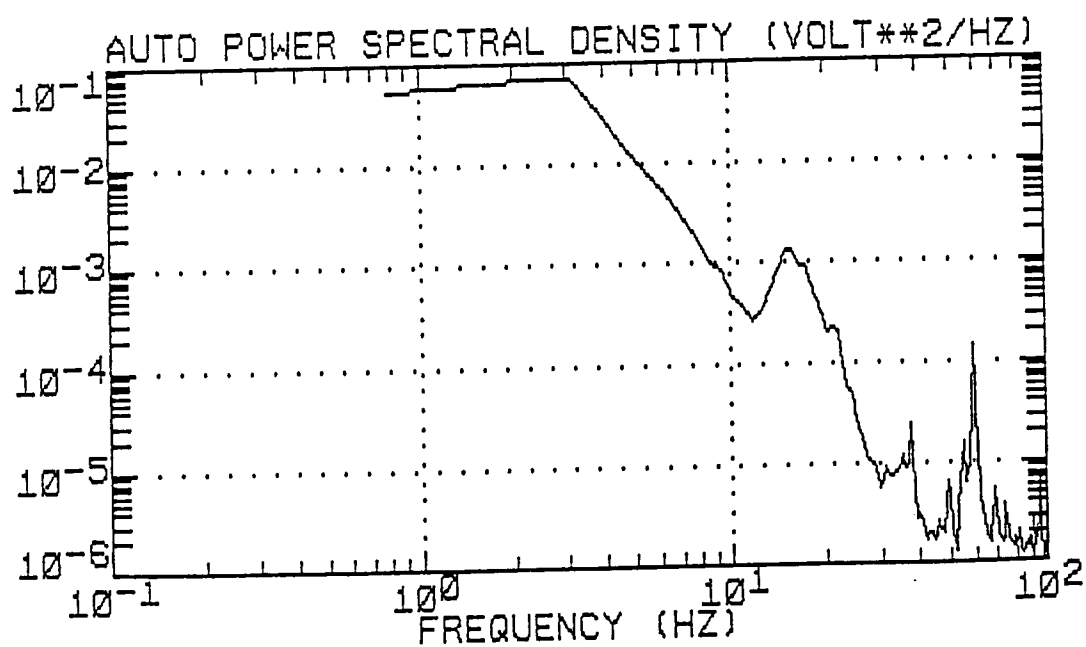


(b). at 64% open

Figure 4.11. Power spectral density functions of air line pressure signal at different valve positions.

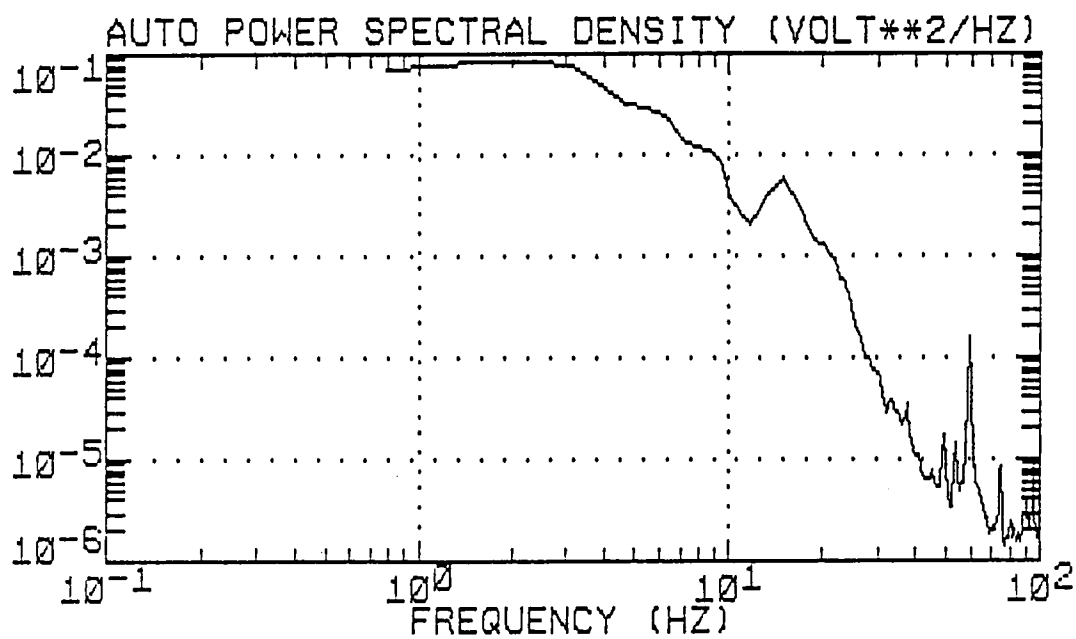


(c). at 40% open



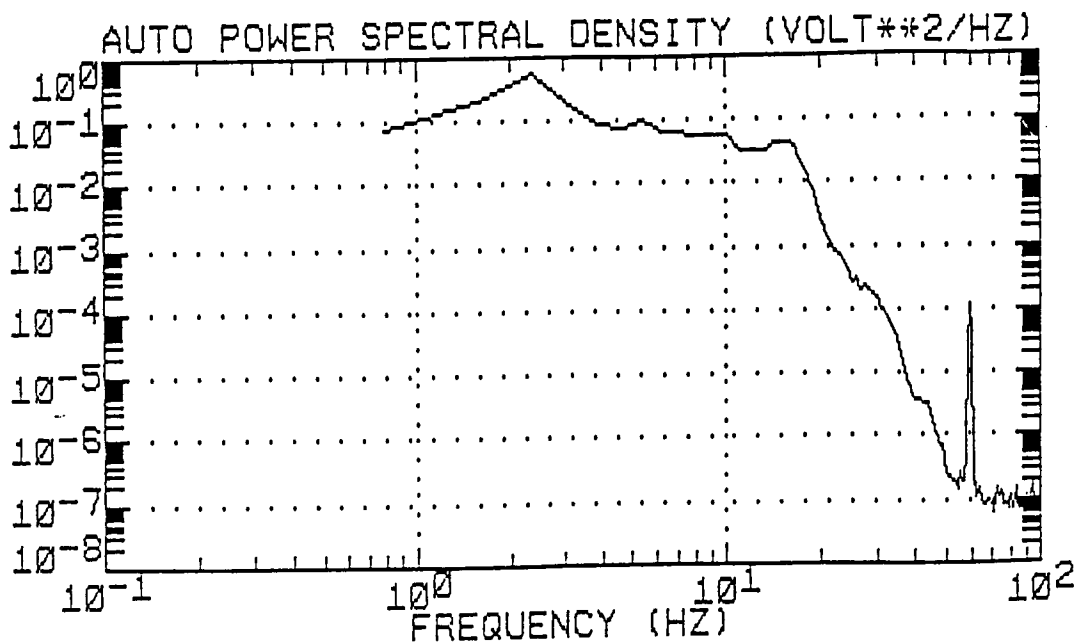
(d). at 18% open

Figure 4.11. (Continued)

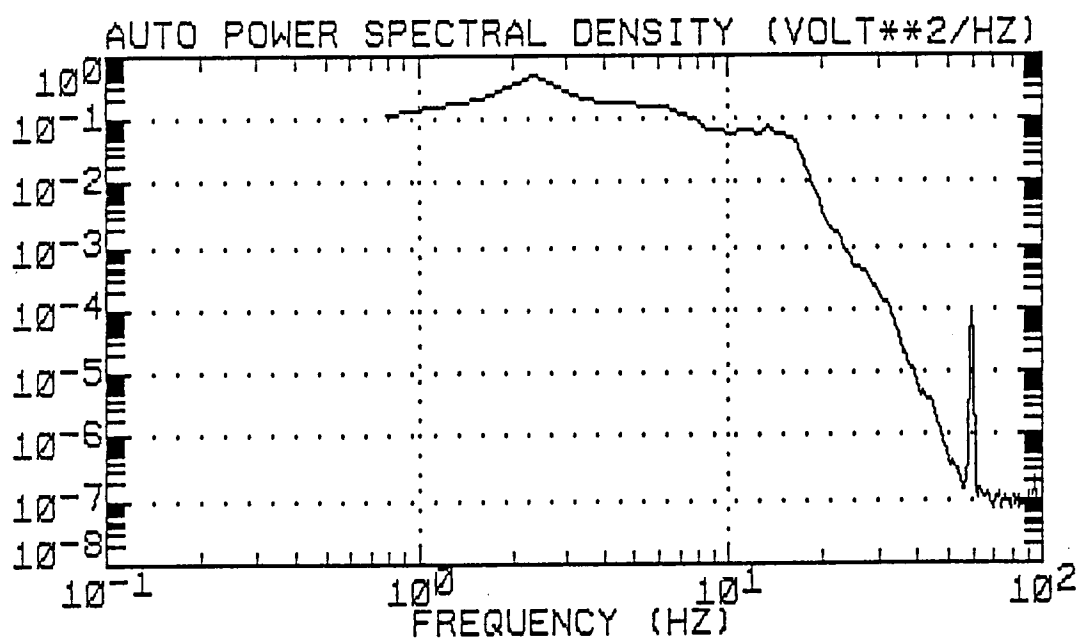


(e). at fuuly closed position

Figure 4.11. (Continued)

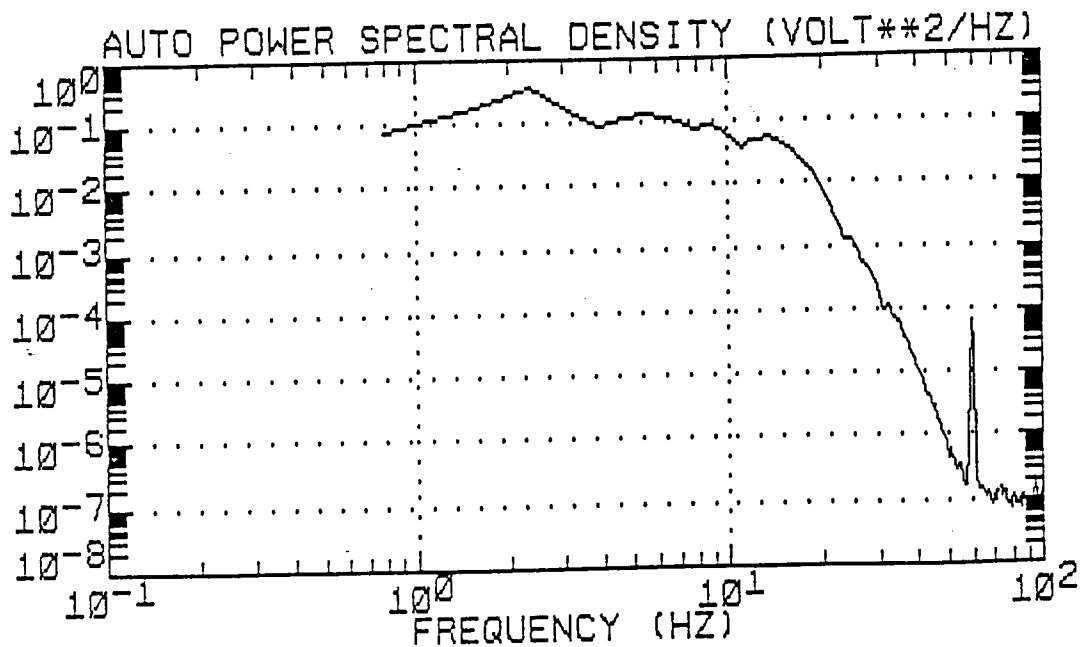


(a). at 100% open

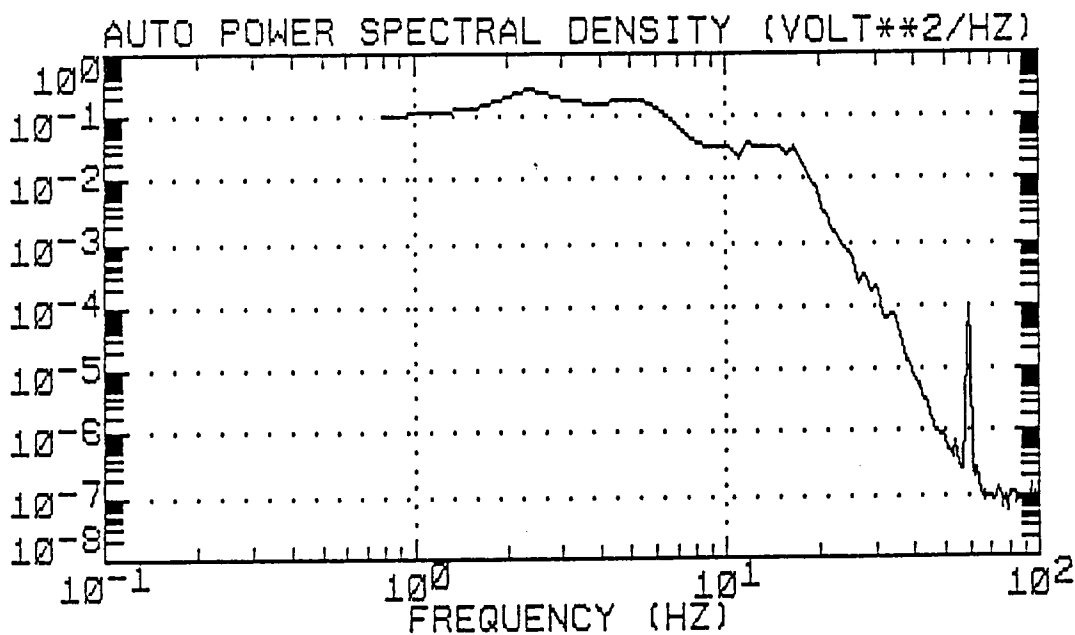


(b). at 64% open

Figure 4.12. Power spectral density function of LVDT signal at different valve positions

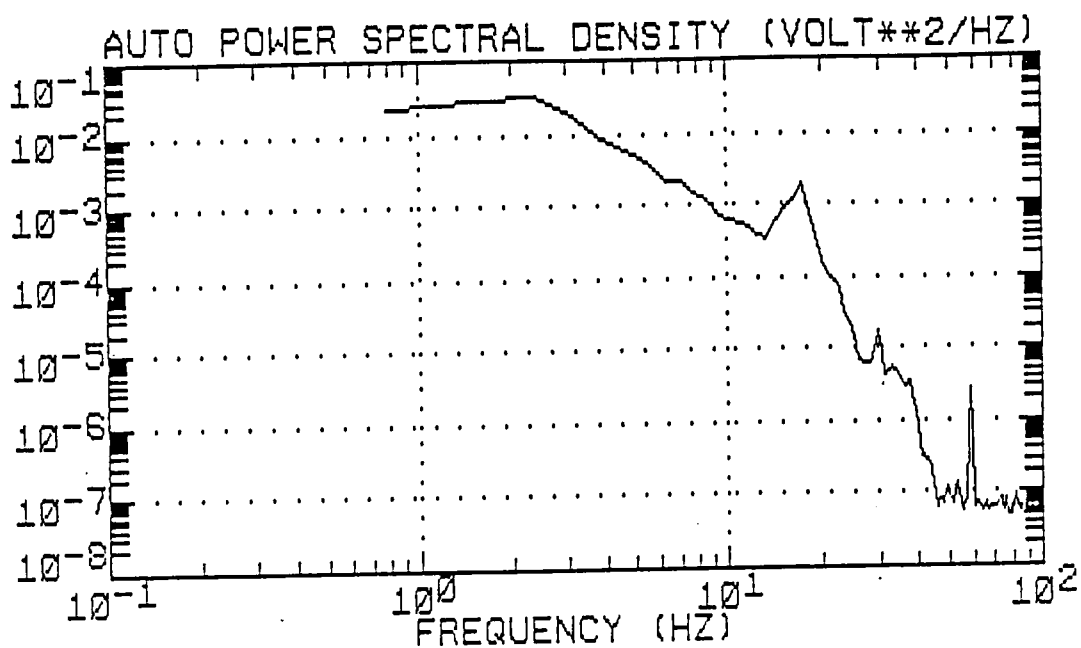


(c). at 40% open

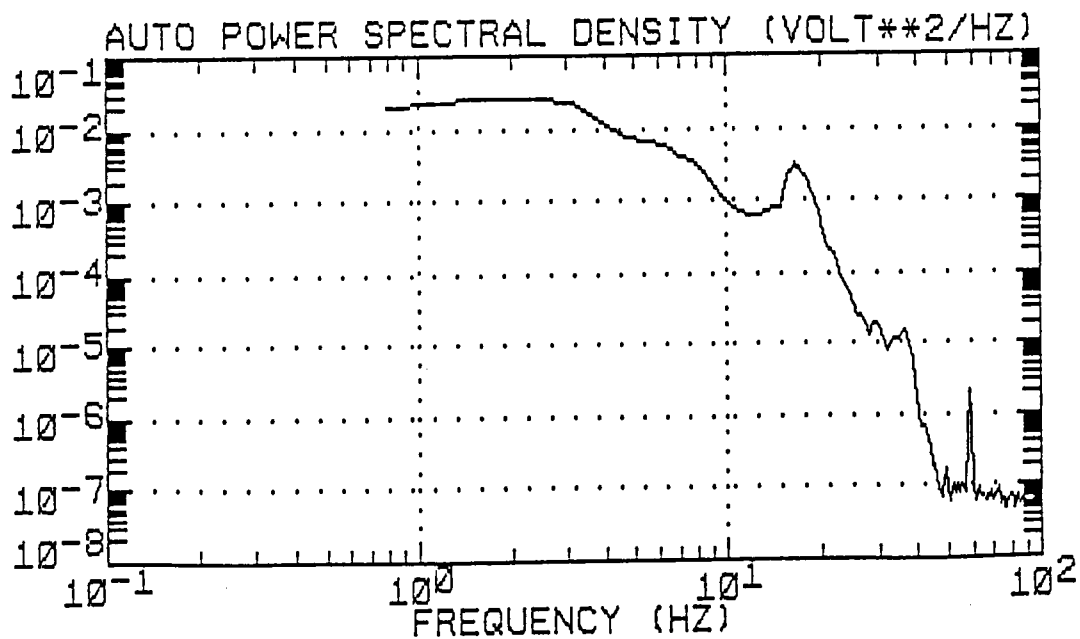


(d). at fuully closed position

Figure 4.12. (Continued)

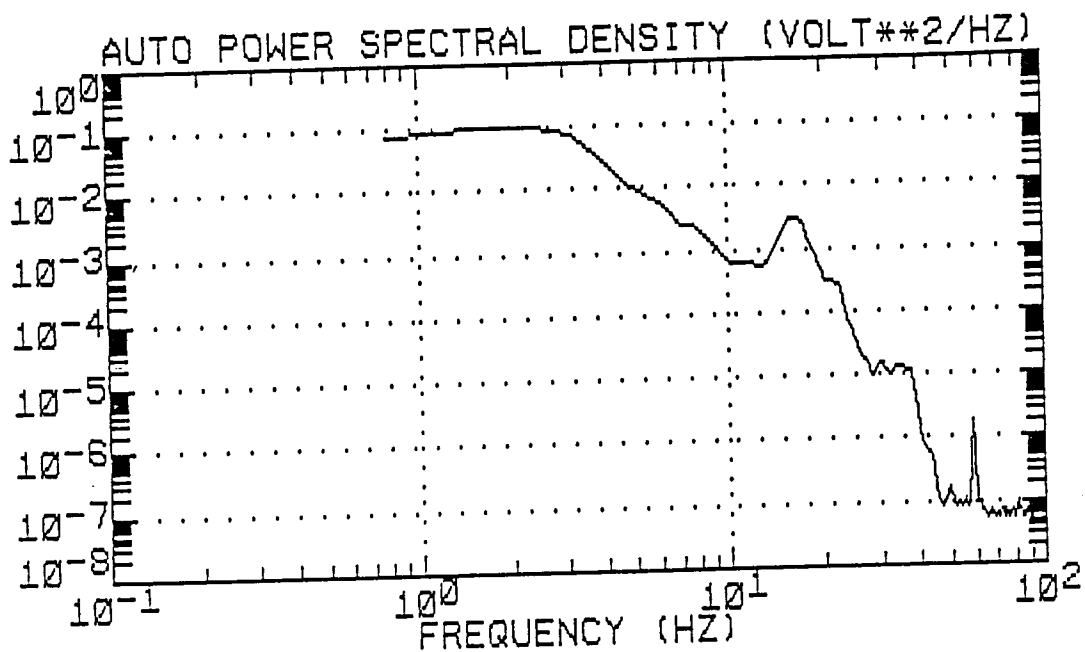


(a). at 100% open

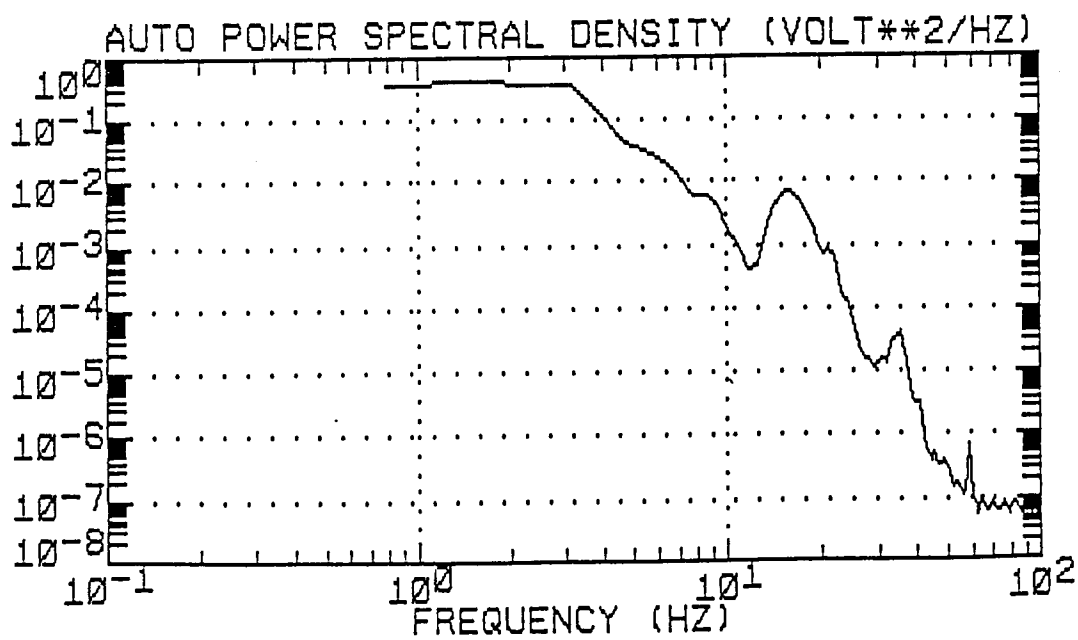


(b). at 64% open

Figure 4.13. Power spectral density function of valve differential pressure signal at different stem positions.

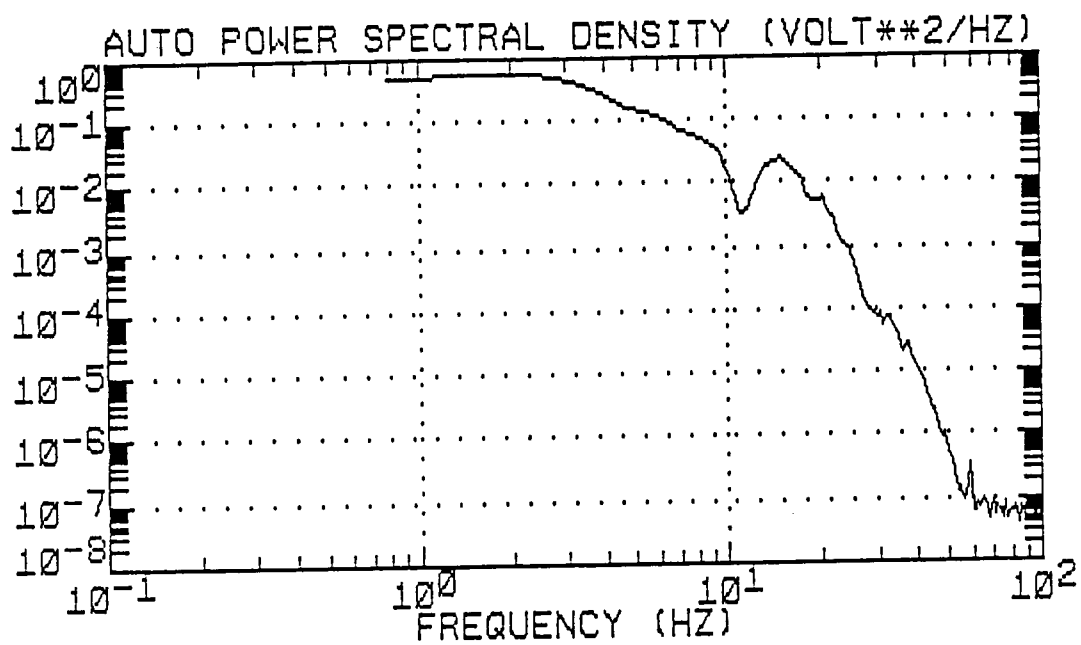


(c). at 40% open



(d). at 18% open

Figure 4.13. (Continued)



(e). at fuully closed position

Figure 4.13. (Continued)

changes as a function of valve position, but the overall variance remains approximately the same.

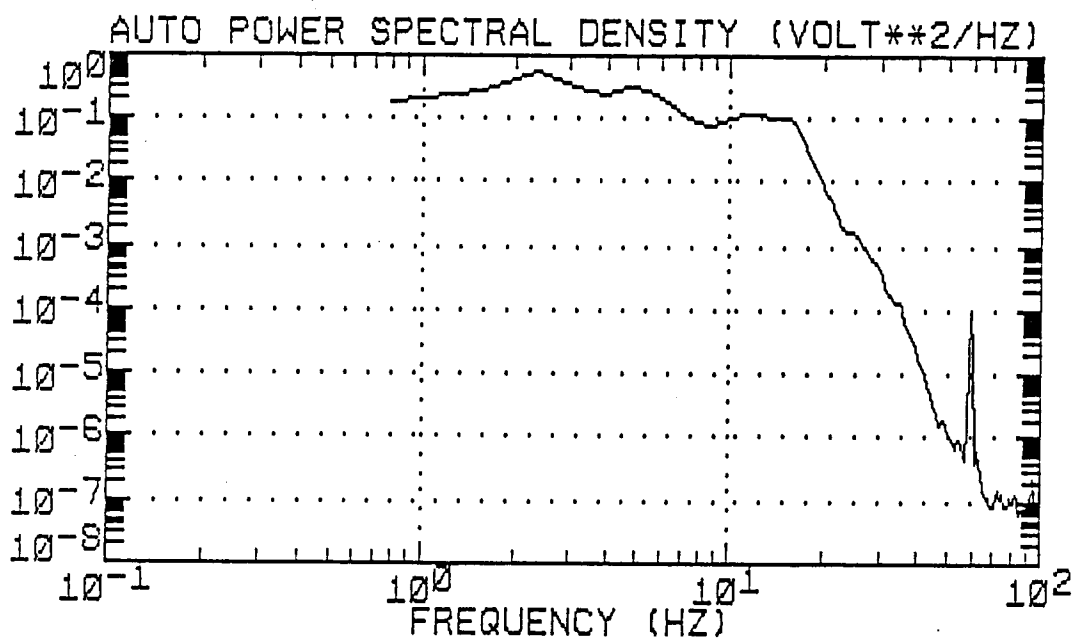
The APSD of valve differential pressure signal are plotted in Figure 4.13. The frequency peaks of 18, 30, and 60 Hz are also noticed. As the valve is gradually closed, the magnitude of the APSD of the differential pressure signal increases. The overall shape of the APSD also changes.

The APSD of the LVDT signal at valve 64% open position with air leakage is shown in Figure 4.14. The behavior of the LVDT signal with air leakage is similar to the results for normal operation. The higher break frequency is at about 18 Hz. The 60 Hz frequency also appears in the APSD. The shapes of APSDs of LVDT change at different valve positions, but the magnitude of the signal changes very little.

The APSDs of LVDT signals at valve 64% open position for the normal case and with increased stem friction are shown in Figure 4.15. These clearly indicate the effect of anomaly in valve operation. Both shapes and magnitude of APSDs of LVDT signals change compared to the normal cases. This change may provide the valve monitoring information.

4.4.5 Autoregression (AR) Modeling of Fluctuating Signals

The autoregression modeling may be used to estimate both frequency- and time-domain signatures. For example, the step response of a system modeled by a signal can be easily computed. Figure 4.16 shows the power spectral density function



at 64% open position

Figure 4.14. Power spectral density function of LVDT signal with air leakage.

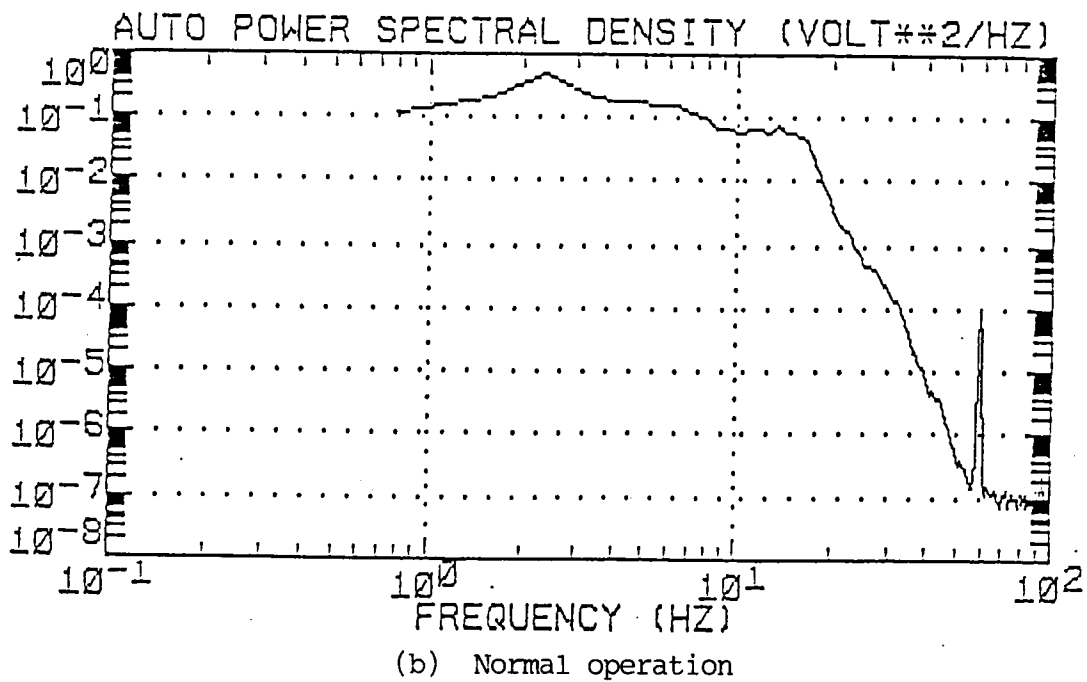
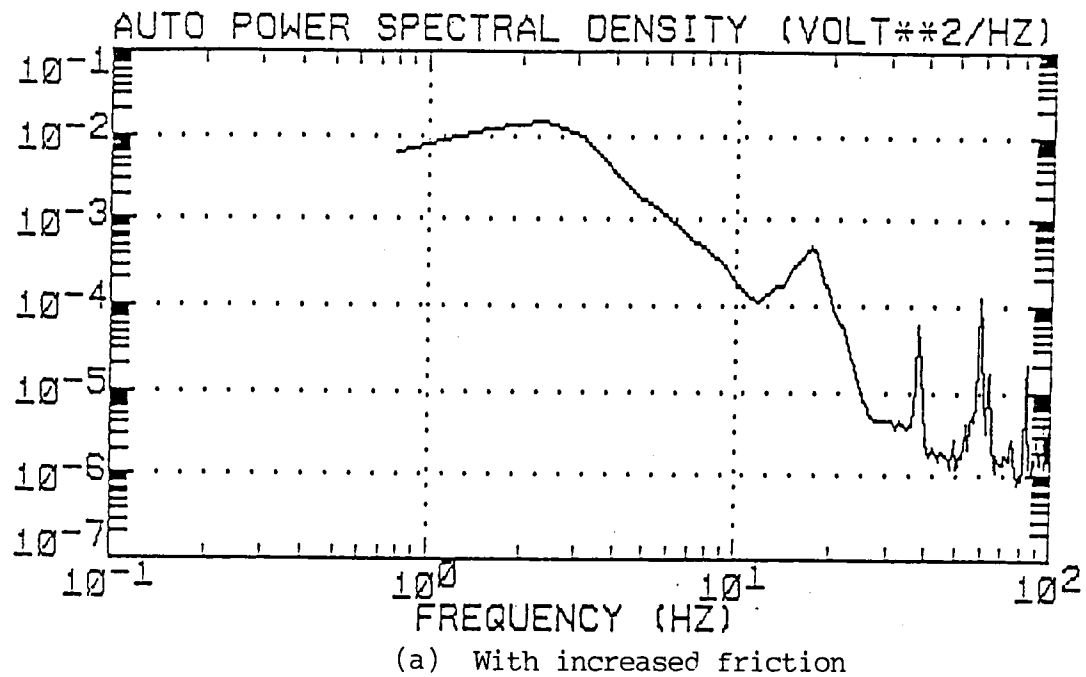


Figure 4.15. Power spectral density function of LVDT signal at 64 % open position

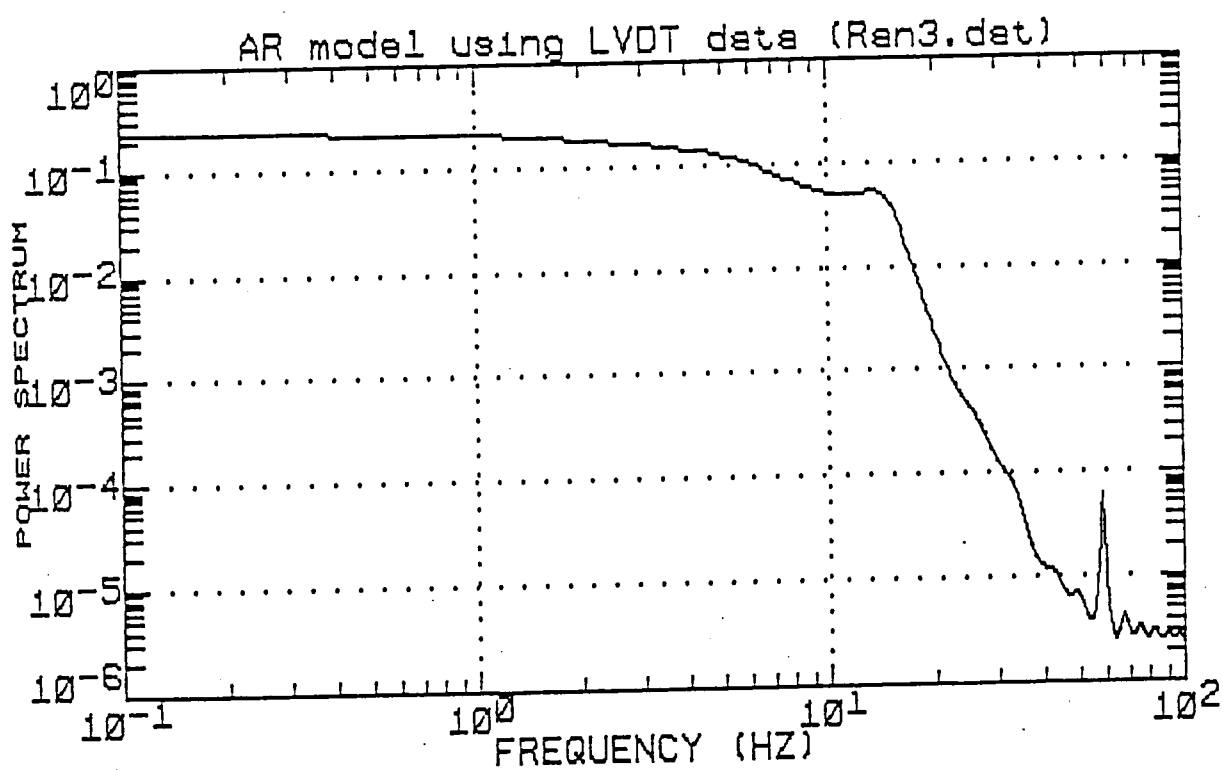


Figure 4.16. Power spectral density function of LVDT signal at valve 64 % open position using AR model approach, model order $n=30$.

and Figure 4.17 shows the step response of the system motion which is modeled using the LVDT signal. The AR model order is $n=30$. The estimated time constant is 0.044 second. The AR modeling approach is also useful in processing a limited amount of data. In practice, the above parameter may be computed on-line and its changes may be monitored to provide information about varying system behavior.

4.5 Conclusions from Random Signal Analysis

The study of random signal analysis of valve related signals has shown that some of the anomalies can be monitored using this approach. The following observations are made from this study.

1. During normal operation, the stem movement is faster in one direction than in the other direction.
2. Valve stem vibration reaches its highest value at valve 84% open.
3. The valve differential pressure fluctuation increases as the valve approaches its closed position.
4. Air line pressure signal has a low frequency component due to the air supply source.
5. As the air pressure on the valve diaphragm decreases, the air pressure signal variance decreases.
6. The valve stem has a natural frequency around 18 Hz.
7. The valve differential pressure signal has a frequency peak around 30 Hz, which may be due to flow impact.
8. The measurements have a peak at 60 Hz which is due to the electrical pick-up and the pump rotational frequency of 3600 rpm.
9. Air leakage can be seen in the APSDs of the measurements.

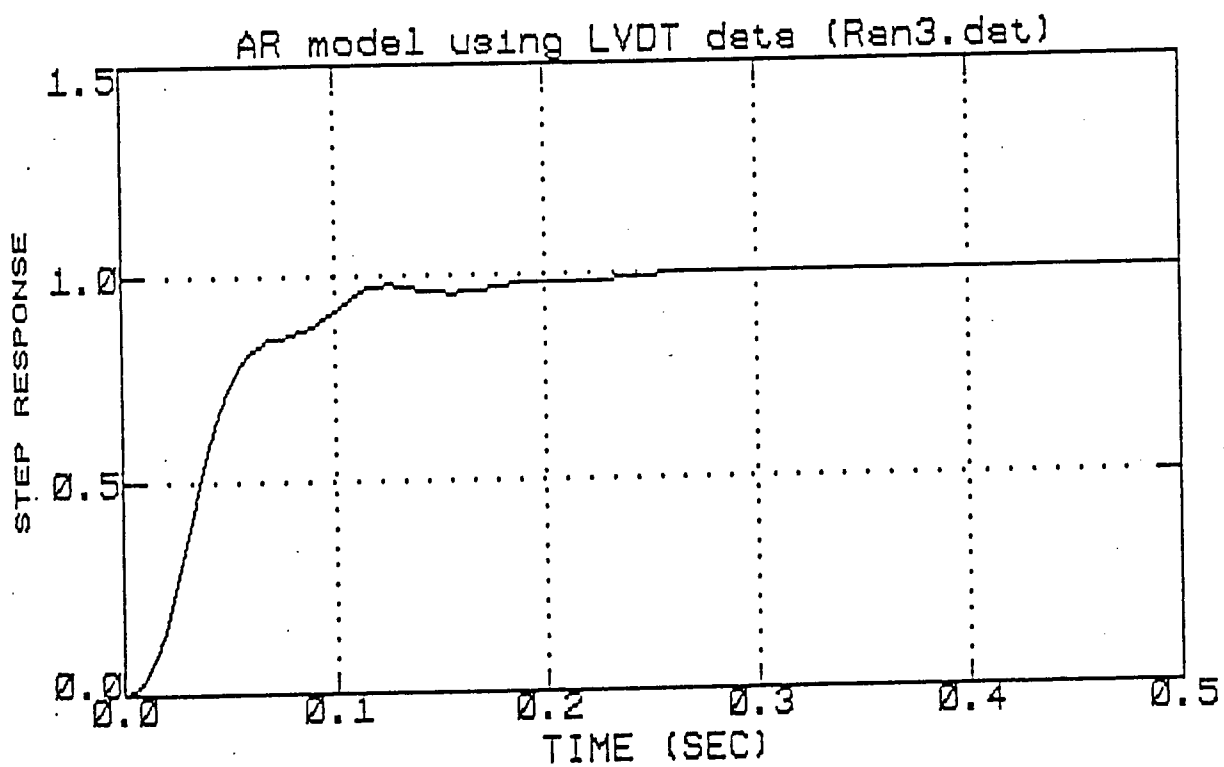


Figure 4.17. Step response of stem motion dynamics at 64 % valve open position, estimated from the AR model. The corresponding time constant is about 0.044 second.

10. Friction variation can be seen in the APSDs.
11. Friction increase will not affect the frequency components of the signals.
12. Time series plots also indicates the operational condition of the valve.
13. The variance of the signal clearly indicates the valve operational condition.
14. AR model provides another approach for random data analysis, especially when limited data are available.

4.6 Summary of Random Signal Analysis

Random signal analysis is a very useful method to diagnose valve anomalies and to monitor its operation. Changes in the valve dynamics are shown in both the time-domain and the frequency-domain signatures. The study shows the feasibility of using random signals to monitor AOV operation.

Further analysis is necessary to identify specific noise signatures that indicate incipient changes in the behavior of AOVs. Both time- and frequency-domain signatures may be quantified for application to industrial systems.

CHAPTER V

MODELING OF AIR-OPERATED VALVE DYNAMICS

5.1 Introduction

Detailed steady-state and dynamic modeling of the test valve are performed as part of the project objectives. The simulation analysis of the model is conducted as a means of verifying experimental results and to model the valve conditions. These studies are also useful in designing proper experiments to test the valve response. The model study includes the following tasks.

1. Model development.
2. Calculation of model parameters.
3. Model simulation.
4. Comparison of model results and the test results.
5. Analysis of anomalies using the model simulation.
6. Determination of the sensitivity of response to model parameters.

5.2 Model Development

The valve and its actuator are mainly composed of a seating plug (obturator), valve housing body, the stem and connection, actuator supporting yoke, diaphragm, spring, and upper and lower chambers. The dynamics of the AOV is simplified in

the form of a second order mass-spring system. Both steady-state and dynamic models have been developed.

5.2.1 Development of Steady-State Model

Assuming that the actuating pressure is applied gradually to the valve diaphragm, a steady-state model of the valve and the actuator may be derived. The stem position is established by a balance of force acting on the stem. These forces are due to the pressure on the diaphragm, spring forces, friction force on the stem, and the flow forces on the valve plug. All the forces are shown in Figure 5.1.

5.2.1.1 Total Force Balance

The steady-state model is derived by a summation of forces on the valve stem, with the positive direction being downward.

$$PA_{eff} + P_{v2}A_{v2} = P_cA_c + nkx + P_{v1}A_{v1} + F_f + F_d \quad (5.1)$$

where: A_{eff} : the effective area of diaphragm (m^2);

A_v : effective inner valve area (m^2);

k : spring constant (N/m);

P : pressure on the diaphragm area (pa or N/ m^2);

P_{v2} : valve downstream pressure (pa);

A_{v2} : valve plug area acted by pressure P_{v2} (m^2);

P_c : pressure on the spring package chamber (pa);

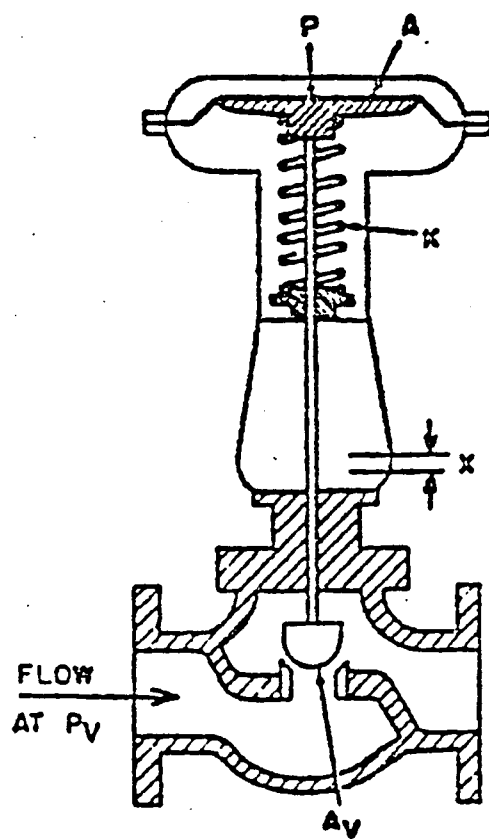


Figure 5.1. Forces acting on the obturator and actuator.

A_c : diaphragm area acted by pressure P_c (m^2);

n : number of springs used in the valve actuator;

P_{v1} : upstream pressure (pa);

A_{v1} : valve plug area acted by P_{v1} (m^2);

F_f : total friction forces on the valve stem (N); and

F_d : resistant force due to the deformation of the diaphragm.

The above equation applies to an air-to-open actuator and the valve combination with the flow passing under the plug. Another popular actuator configuration, commonly referred to as air-to-close, is one causing the stem to lower on an increase of air pressure. Using the same sign convention, the equation for this type of valve is given by

$$PA_{eff} + P_{v2}A_{v2} = P_cA_c - nkx + P_{v1}A_{v1} - F_f + F_d \quad (5.2)$$

The test valve is an air-to-open type actuator.

5.2.1.2 Relationship Between Total Forces (ΣF) and Stem Displacement (x)

To develop the AOV model, the diaphragm can be simplified as a center rigid plate with a diameter d and the outer edge fixed at diameter D (Figure 5.2). The forces on the diaphragm are the spring forces, pressure forces on both sides of the diaphragm, and resistance force due to the diaphragm deformation. At steady-state condition the summation of these forces is equal to zero.

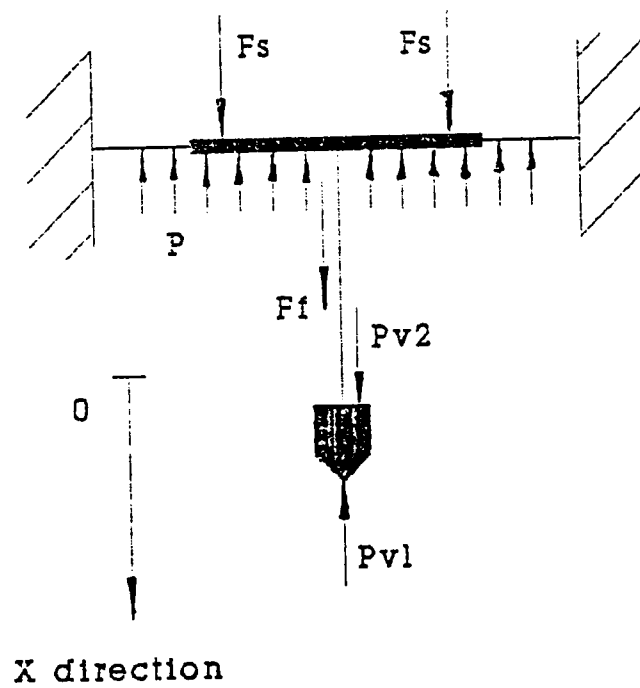


Figure 5.2. Forces on the diaphragm and stem for the test valve.

A. Deformation caused by the pressures on both sides of the diaphragm

The deformation of the diaphragm caused by pressures on it is given by [26]

$$x_p = \frac{3(P-P_c)(m^2-1)}{16m^2 E_d t_d^3} [D^4 + 3d^4 - 4D^2 d^2 (1 + \log \frac{D}{d}) + \frac{16D^2 d^4}{D^2 - d^2} \log^2 \frac{D}{d}] \quad (5.3)$$

where m: the reciprocal of ν , Poisson's ratio;

t_d : thickness of diaphragm (m);

E_d : modulus of elasticity (pa).

B. Deformation caused by all forces on the diaphragm

All the forces acting on the diaphragm are simplified as a linear distribution force along the diaphragm inner ridge plate, which is referred to as w , the ratio of the total forces acted on the diaphragm alone and the diaphragm inner circumference, and is given by

$$w = \frac{\Sigma F}{\pi d} = \frac{F_f - P_{v1} A_{v1} + P_{v2} A_{v2}}{\pi d} \quad (5.4)$$

The deformation resulting from all the forces on the valve stem and the diaphragm is given by the equation:

$$x_w = \frac{3w(m^2-1)}{4\pi m^2 E_d t_d^3} [D^2 - d^2 - \frac{4D^2 d^2}{D^2 - d^2} \log^2 (\frac{D}{d})] \quad (5.5)$$

where X_w is the deformation caused by all the forces on the diaphragm.

C. Relationship between forces and the diaphragm position

The relationship between the deformation of the diaphragm caused by both pressures and forces on the valve stem and the diaphragm is given by

$$x_p - x_w = x \quad (5.6)$$

where x is defined as the stem position at time t .

Therefore, the steady-state model of the valve is a function of the pressure acting on the valve diaphragm, the position of the valve stem, and the spring and packing properties. The steady-state model is given by the following equation:

$$x = \frac{3(P-P_c)(m^2-1)}{16m^2Et^3} \left[D^4 + 3d^4 - 4D^2d^2 \left(1 + \log \frac{D}{d} \right) + \frac{16D^2d^4}{D^2-d^2} \log^2 \frac{D}{d} \right] - \frac{3w(m^2-1)}{4\pi m^2Et^3} \left[D^2 - d^2 - \frac{4D^2d^2}{D^2-d^2} \log^2 \frac{D}{d} \right] \quad (5.7)$$

Friction effects occur in the valve stem packing, in the actuator stem guide, and in the valve plug guide or guides. Usually, for static valve actuator sizing problems, negligible error is introduced by ignoring the friction terms. These equations can be simplified if the friction and inertia forces are not considered. If the equations are plotted as pressure versus stem travel, the relationship in Figure 5.3 is obtained.

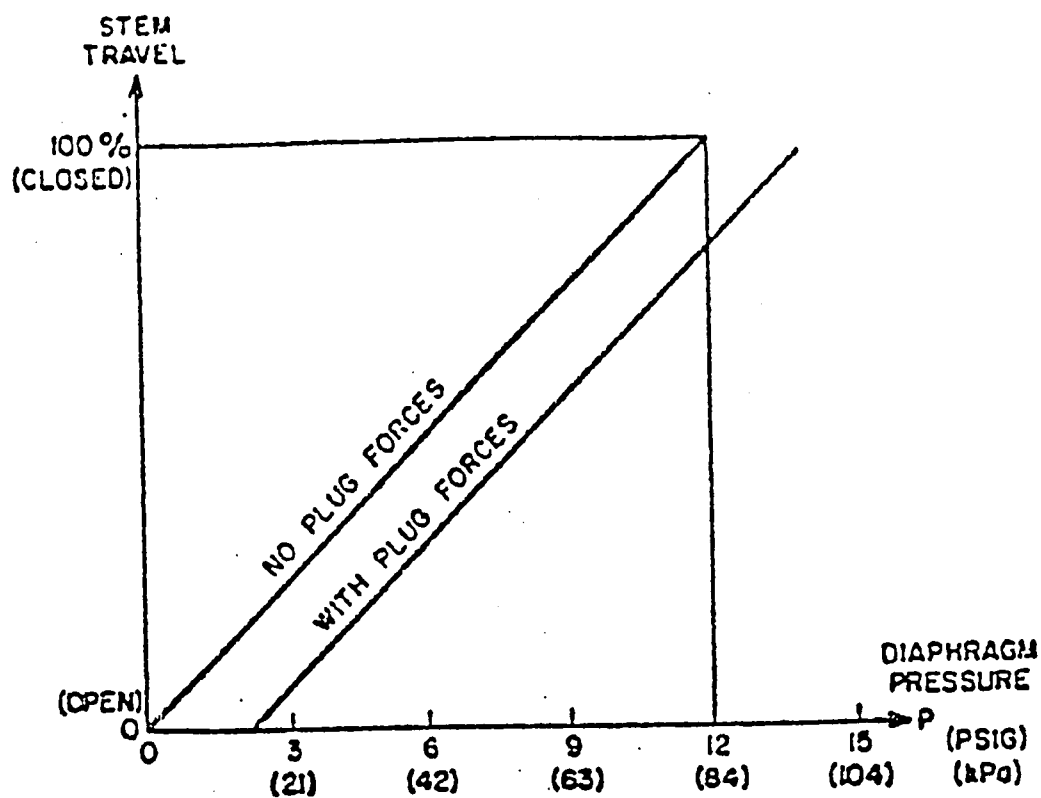


Figure 5.3. Relationship between stem travel versus pressure on diaphragm [10].

5.2.2 Development of Dynamic Model of AOVs

The dynamic performance of the valve and its actuator is given by simplifying the valve and actuator as a spring mass system with friction term in the equation. The derivation of the model is completed by the force balance equation.

The spring mass system can be written as a second order system with the friction terms in the system when the pressure P acts on the valve actuator diaphragm [10-15]

$$M \frac{d^2x}{dt^2} = -p_c A_c - p_{v2} A_{v2} - F_f + P A_{eff} - k_n x - F_d + P_{v1} A_{v1} \quad (5.8)$$

where M is the mass of the valve stem (kg), and F_f and F_d are the friction force and diaphragm resistance force respectively. The F_f and F_d are given by the following equations

$$F_f = c \frac{dx}{dt} \quad (5.9)$$

$$F_d = \frac{16m^2 E t^3}{3(m^2 - 1)} \left[D^4 + 3d^4 - 4D^2 d^2 \left(1 + \log \frac{D}{d} \right) + \frac{16D^2 d^4 \log^2 \frac{D}{d}}{(D^2 - d^2)} \right]^{-1} x$$

$$= bx \quad (5.10)$$

where c is the viscosity of the valve lubrication material, and b is a constant determined by the diaphragm properties and the parameters. Thus, a valve model of the stem position versus the actuator pressure is derived as

$$M \frac{d^2x}{dt^2} + c \frac{dx}{dt} + (K+b)x = PA_{eff} + Other\ Terms \quad (5.11)$$

where the other terms are given by

$$Other\ Terms = -P_c A_c - P_{v2} A_{v2} + P_{v1} A_{v1} \quad (5.12)$$

where c = the viscosity of the valve lubrication material;

b = a constant determined by the diaphragm properties.

The spring-mass system will be written as follows when the actuator acting pressure P is released:

$$M \frac{d^2x}{dt^2} = P_c A_c + P_{v2} A_{v2} - F_f - Kx + F_d - P_{v1} A_{v1} \quad (5.13)$$

where F_f and F_d are resistant forces of friction and diaphragm. The equation is rearranged in the form:

$$M \frac{d^2x}{dt^2} + c \frac{dx}{dt} + (K+b)x = P_c A_c + P_{v2} A_{v2} - P_{v1} A_{v1} \quad (5.14)$$

The spring mass system may be linearized to a linear second order system if the nonlinear elements and smaller force terms are neglected. During normal operation of the test valve, $P_c A_c$, $P_{v2} A_{v2}$ and $P_{v1} A_{v1}$ are not dominant terms in the equation, so the linear model for the stem position can be simplified as

$$M \frac{d^2x}{dt^2} + c \frac{dx}{dt} + (K+b)x = PA_{eff} \quad (5.15)$$

and

$$M \frac{d^2x}{dt^2} + c \frac{dx}{dt} + (K+b)x = 0 \quad (5.16)$$

5.3 Calculation of Model Parameters

The parameters in both the dynamic and steady-state models are calculated based on their physical meaning. The parameters needed for valve modeling are the spring constant, effective area of the diaphragm, pressure on the diaphragm spring chamber if it is sealed, and the valve packing material friction coefficient.

5.3.1 Spring Constant (K)

The spring constant for each spring is determined from the equation [27]

$$k = \frac{Gd_s^4}{8D_s^3N} \quad (5.17)$$

where G : the torsion modulus of spring material (90 Gpa);

d_s : the diameter of the spring wire (0.00396 m);

D_s : the average diameter of the spring ring (0.032m); and

N : the nominal number of spring cycles (3).

The spring constant k has the value

$$k = \frac{90 \times 10^9 \times 0.00396^4}{8 \times 0.032^3 \times 3} = 2.814 \times 10^4 (N/m).$$

There are six springs used in the valve actuator. So, the total spring constant K is 168840 N/m.

5.3.2 Effective Diaphragm Area (A_{eff})

The effective diaphragm area A_{eff} [28] is given by the equation

$$A_{eff} = \frac{\pi}{12}(D^2 + d^2 + Dd) \quad (5.18)$$

where D is the outside diameter of the diaphragm (0.208 m) and d is the inner diameter of the diaphragm (0.155 m). So the effective diaphragm area A is 0.0261 m².

5.3.3 Pressure in the Spring Chamber

When the reverse side of the diaphragm pressure chamber is sealed, the pressure P_c in the chamber must be calculated due to temperature and volume changes. The relationship is given by

$$\frac{P_c V_c}{T_c} = \frac{P_{cl} V_{cl}}{T_{cl}}, \quad (5.19)$$

where P_{cl} and P_c are the pressure at actuator installation condition and the pressure after a deformation of the diaphragm and the change of temperature.

V_{cl} and V_c : the volumes of the spring package before and after the diaphragm deformation and any temperature change;

T_{cl} and T_c : the temperatures at the installation condition and the operation ambient condition.

The volume change of the spring package due to the diaphragm deformation x is given by

$$V_c = V_{cl} - \frac{\pi}{12}(D^2 + Dd + d^2)x \quad (5.20)$$

From Equations (5.19) and (5.20)

$$P_c = \frac{P_{cl} V_{cl} T_c}{T_{cl} [V_{cl} - \frac{\pi}{12}(D^2 + Dd + d^2)x]} \quad (5.21)$$

Assuming: $P_{cl} = 101.325 \text{ kpa (1 atm)}$;

$T_{cl} = T_c$ and

$$\begin{aligned} P_c &= \frac{1.01325 \times 10^5}{1 - \frac{\pi(D^2 + Dd + d^2)x}{12V_{cl}}} \\ &= \frac{1.01325 \times 10^5}{1 - ex} \end{aligned} \quad (5.22)$$

where e is a constant determined by

$$e = \frac{\pi(D^2 + Dd + d^2)}{12V_{cl}} = \frac{A_{eff}}{V_{cl}} \quad (5.23)$$

and V_{cl} is $9.832 \times 10^{-4} \text{ m}^3$. So, the constant e is equal to 26.44 m^{-1} .

When the P_c is expanded in a binomial form we have

$$P_c = 1.01325 \times 10^5 (1 + ex + e^2 x^2 + e^3 x^3 + \dots). \quad (5.24)$$

Neglecting higher order terms we get

$$P_c = 1.01325 \times 10^5 (1 + 26.44x) \quad (5.25)$$

As stated previously, this pressure is considered only if the diaphragm spring chamber is sealed.

5.3.4 Diaphragm Resistant Coefficient (b)

The constant b is given by [28]

$$b = \frac{16m^2 E_d t_d^3}{3(m^2 - 1)} \left[D^4 + 3d^4 - 4D^2 d^2 \left(1 + \log \frac{D}{d} \right) + \frac{16D^2 d^4}{(D^2 - d^2)} \log^2 \frac{D}{d} \right]^{-1} \quad (5.26)$$

where $m = 2.5$, $E_d = 2.07$ Gpa, $t_d = 0.002$ m, $D = 0.208$ m, and $d = 0.155$ m. Using these values b is equal to 1.0×10^{-7} N/m.

5.3.5 Determination of Mass (M) and Friction Coefficient (c)

The mass term in the model is measured by weighing the moving parts of the test valve. These include the valve stem and plug, diaphragm holding plates, connection nuts, and half weight of the spring. The total mass is 0.85 kg.

The friction coefficient c is 0.36 N-s/mm, according to Ref. [29]. This is the friction coefficient of leather on a cast iron polished surface.

5.4 Simulation Studies Using the Valve Model

For the simplified valve model, a second order spring-mass system is given by Equation (5.15). The transfer function of this model is obtained by taking the Laplace transform of the second order model.

$$\frac{X(s)}{P(s)} = \frac{\frac{A_{eff}}{K+b}}{\frac{M}{K+b}s^2 + \frac{c}{K+b}s + 1} \quad (5.27)$$

where s is the Laplace operator. Substituting A_{em} , M , K , b , and c into the transfer function, we have

$$\begin{aligned} \frac{X(s)}{P(s)} &= \frac{1.55 \times 10^{-7}}{5 \times 10^{-6}s^2 + 2.13 \times 10^{-2}s + 1} \\ &= \frac{1}{32.567 \times s^2 + 137930 \times s + 6468940} \end{aligned} \quad (5.28)$$

The natural frequency of this spring-mass system varies according to the valve size. The model provides two break frequencies, one at $f_1 = 7.4$ Hz and the other at $f_2 = 682$ Hz. The dominant response corresponds to the lower frequency and is also seen in the power spectral density function of the LVDT signal.

5.4.1 Bode Plot from the Transfer Function

Bode plots are used to analyze the system properties. In the present case

system bandwidth and response may be identified. Bode plot of magnitude of the transfer function as a function of frequency is shown in Figure 5.4. From the Bode plot the approximate break frequency is 46.5 rad/sec or 7.4 Hz.

5.4.2 Response of the Model to a Step Input

The response of the model to a step input is calculated using PC-MATLAB [25]. The test valve model is given by Equation (5.15).

$$\frac{d^2x}{dt^2} + \frac{c}{M} \frac{dx}{dt} + \frac{K+b}{M}x = \frac{A_{eff}}{M}P$$

where $M = 0.85$ kg, $c = 0.36$ N·S/mm², $K = 168840$ N/m, and $A_{eff} = 0.026$ m², and $b = -1 \times 10^7$ N/m. The step input to the model is the pressure acting on the diaphragm. It is introduced by letting $P = 84.74$ kpa (12 psig), which is equal to the experimental step pressure input. The response of the stem travel to this step input is plotted in Figure 5.5. The approximate time constant from the model step response is 0.022 second. This is very close to the experimental value. The random signal analysis gives a time constant of 0.04 second. This estimate is always conservative.

5.4.3 Response of the Model to a Ramp Input

The response of the model to a ramp input is shown in Figure 5.6. The ramp input ranges from 3 psig to 12 psig in 5.4 seconds. The response of the model is

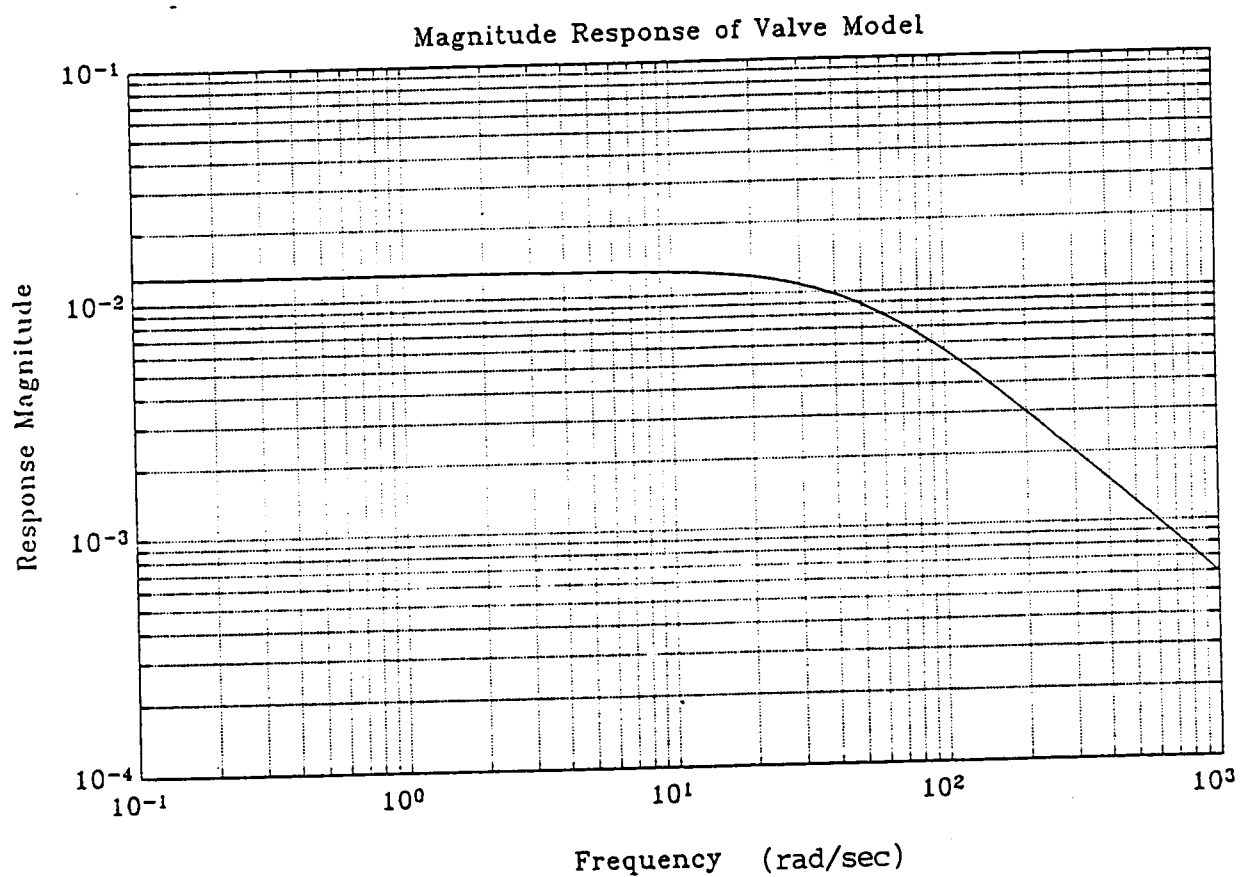


Figure 5.4. Bode plot of the magnitude of the transfer function vs. frequency. The break frequency $f_b = 46.5$ rad/sec or 7.4 Hz.

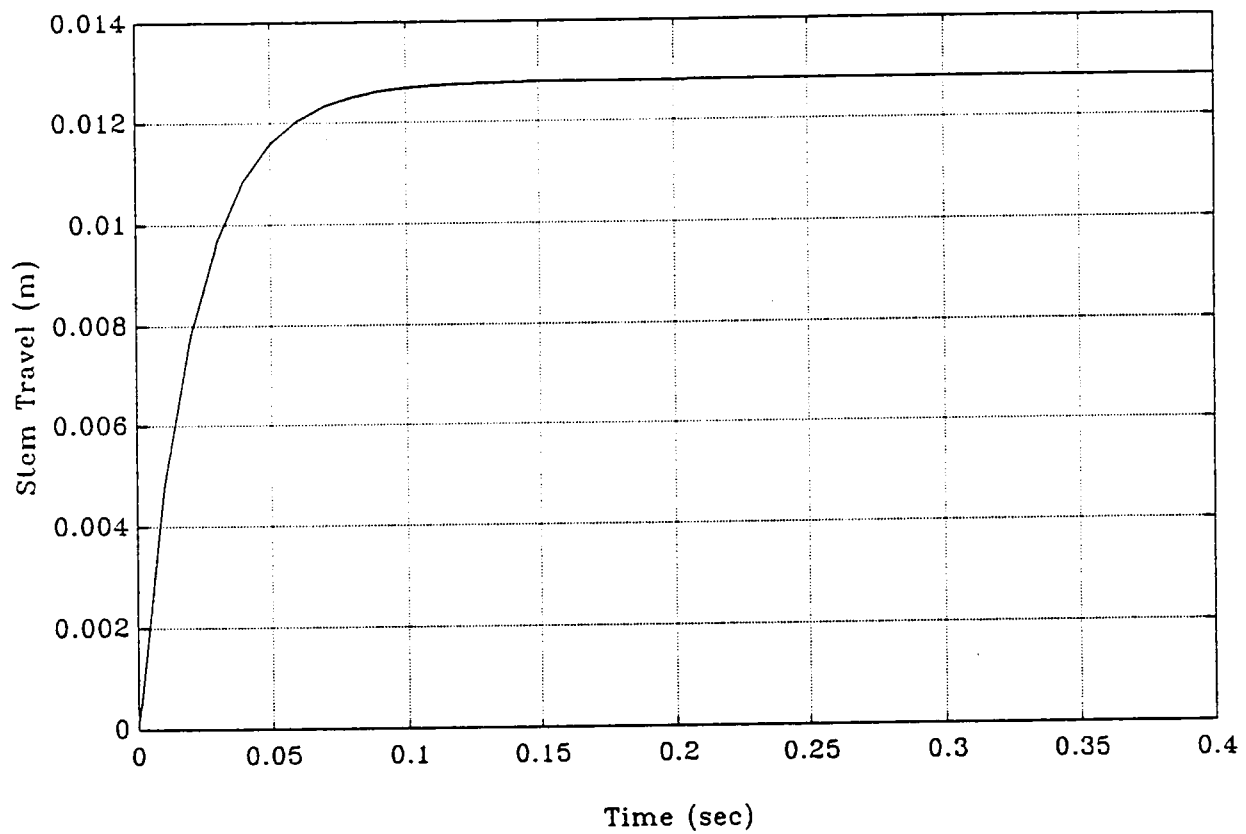


Figure 5.5. Response of the model to a step pressure input.

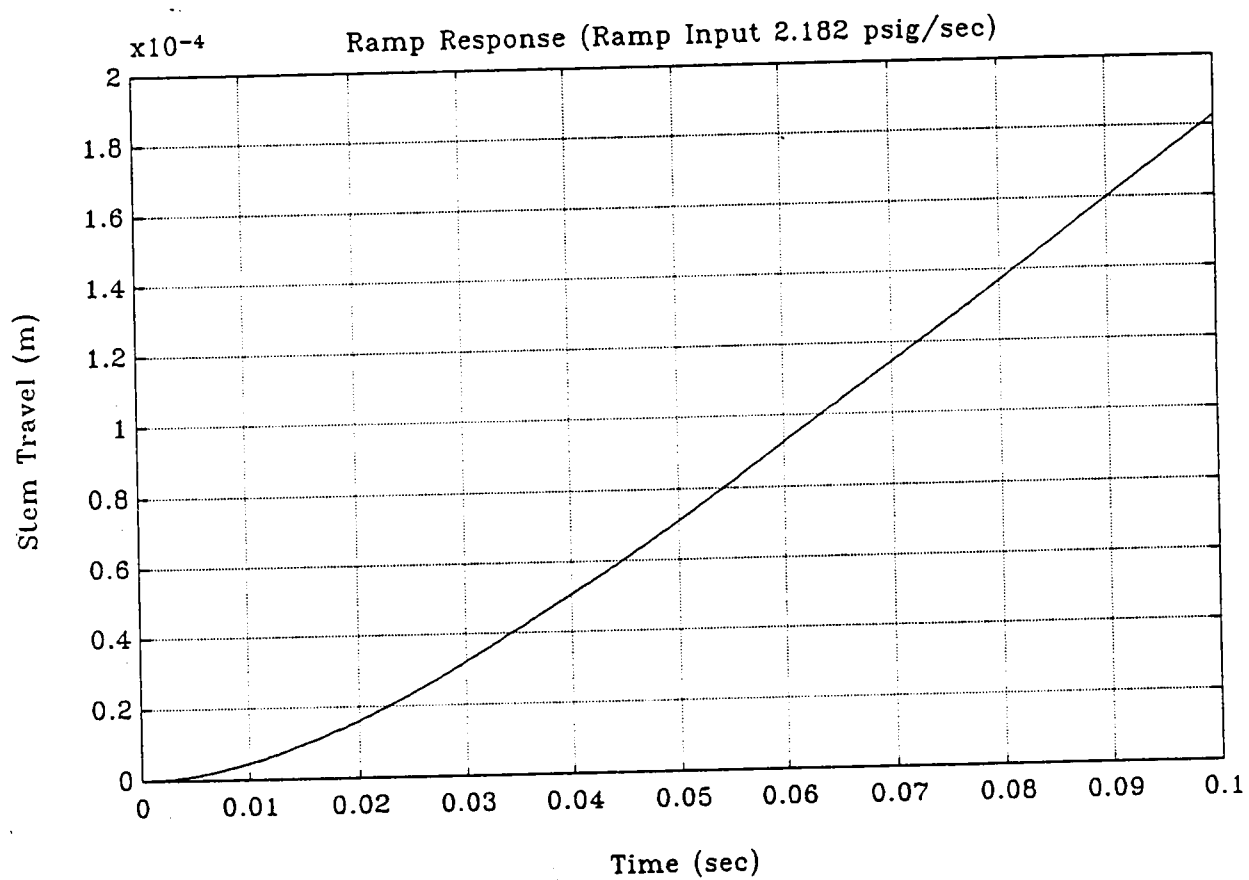


Figure 5.6. Response of the model to a ramp pressure input.

similar to the response shown in the experimental results.

5.5 Model Simulation with Anomalies

Model simulations are also performed with the introduced anomalies. These anomalies are the air line or diaphragm leakage and the friction variation. The air leakage is simulated by changing the pressure input magnitude. The friction variation is simulated using the friction coefficient in the model. The simulation, strictly speaking, is not very accurate for friction increases. However, a similar result is given. With these introduced anomalies to the model, the step responses and ramp responses are studied.

5.5.1 Response of the Model with Air Leakage

The air leakage was simulated by gradually reducing the pressure input to the model by 10 % of its original value.

The responses of the model to the step input with air leakage anomaly is given in Figure 5.7. The pressure was reduced by 5 % of its original value. As the pressure was reduced, it was obvious that the stem travel will also decrease, and was verified by the simulation results. For a pressure of 12 psig, the stem travel is about 0.0128 m, which is very close to the actual stem travel. The time constants of the step response corresponding to different pressure levels do not change.

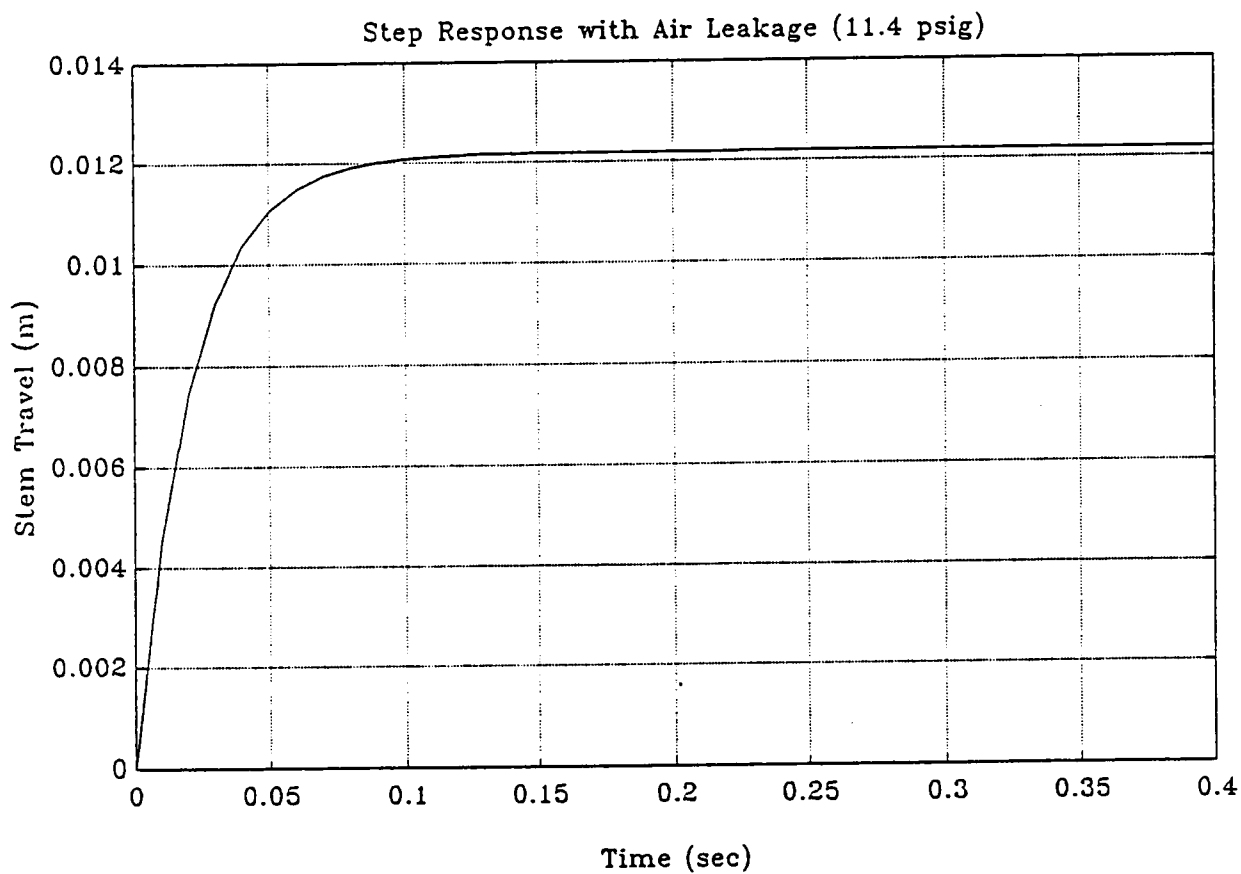
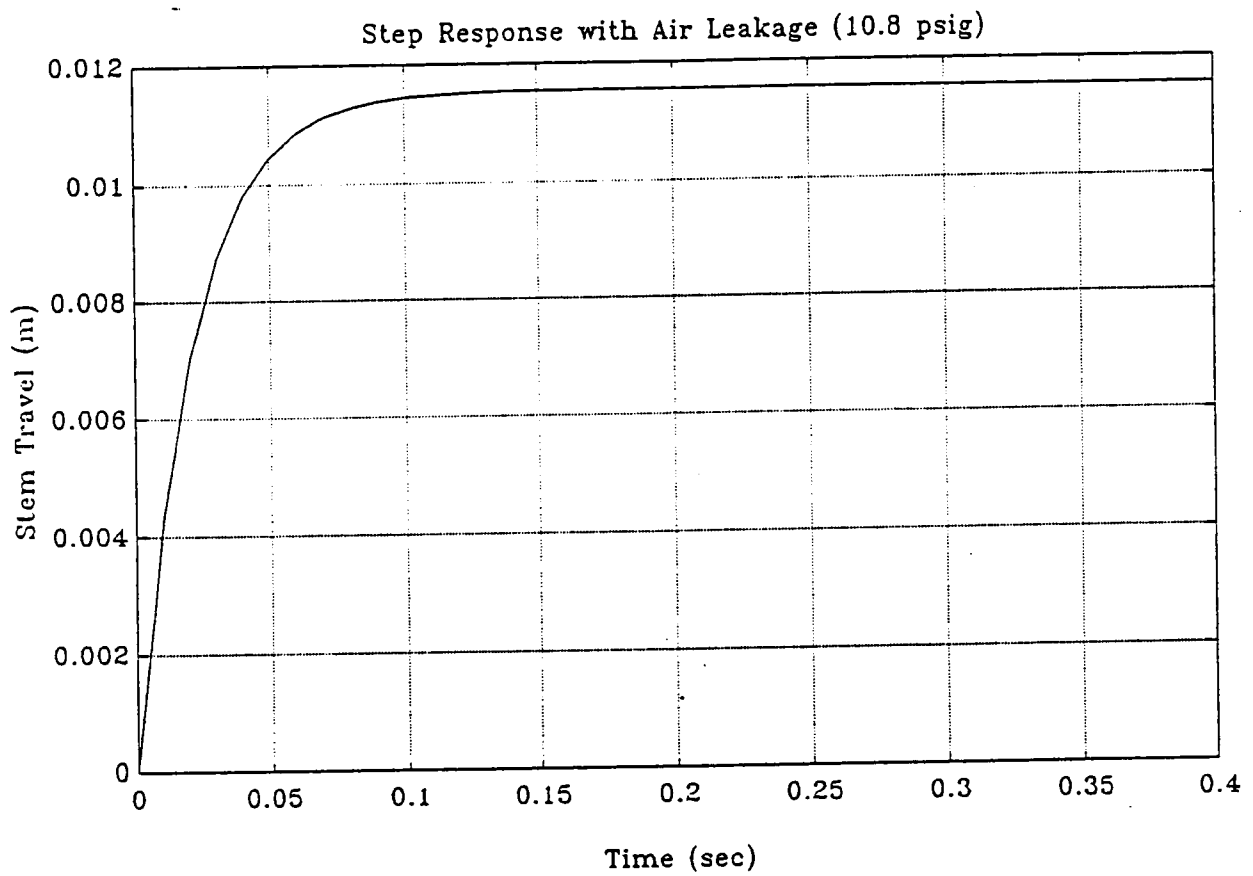
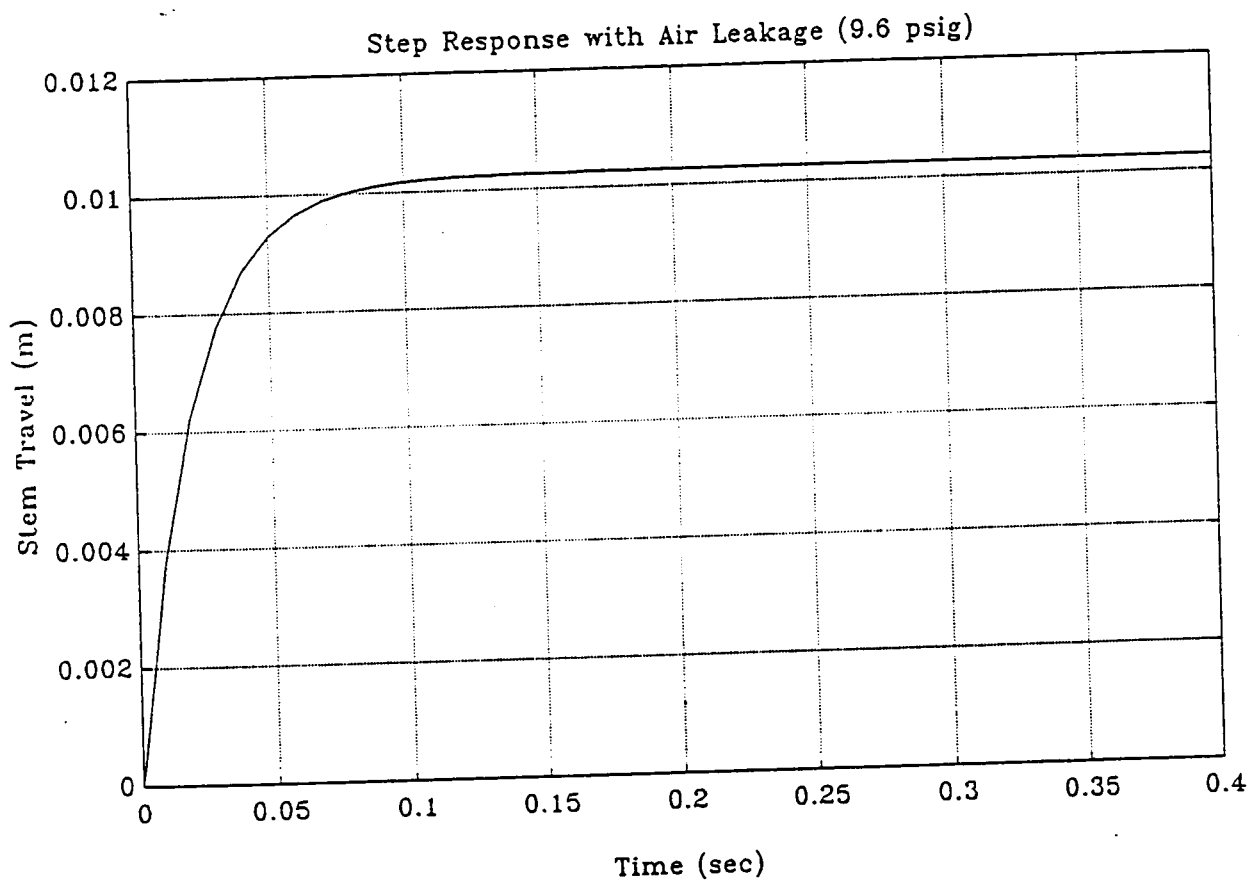


Figure 5.7. Response of the model simulation to the step input with air leakage.



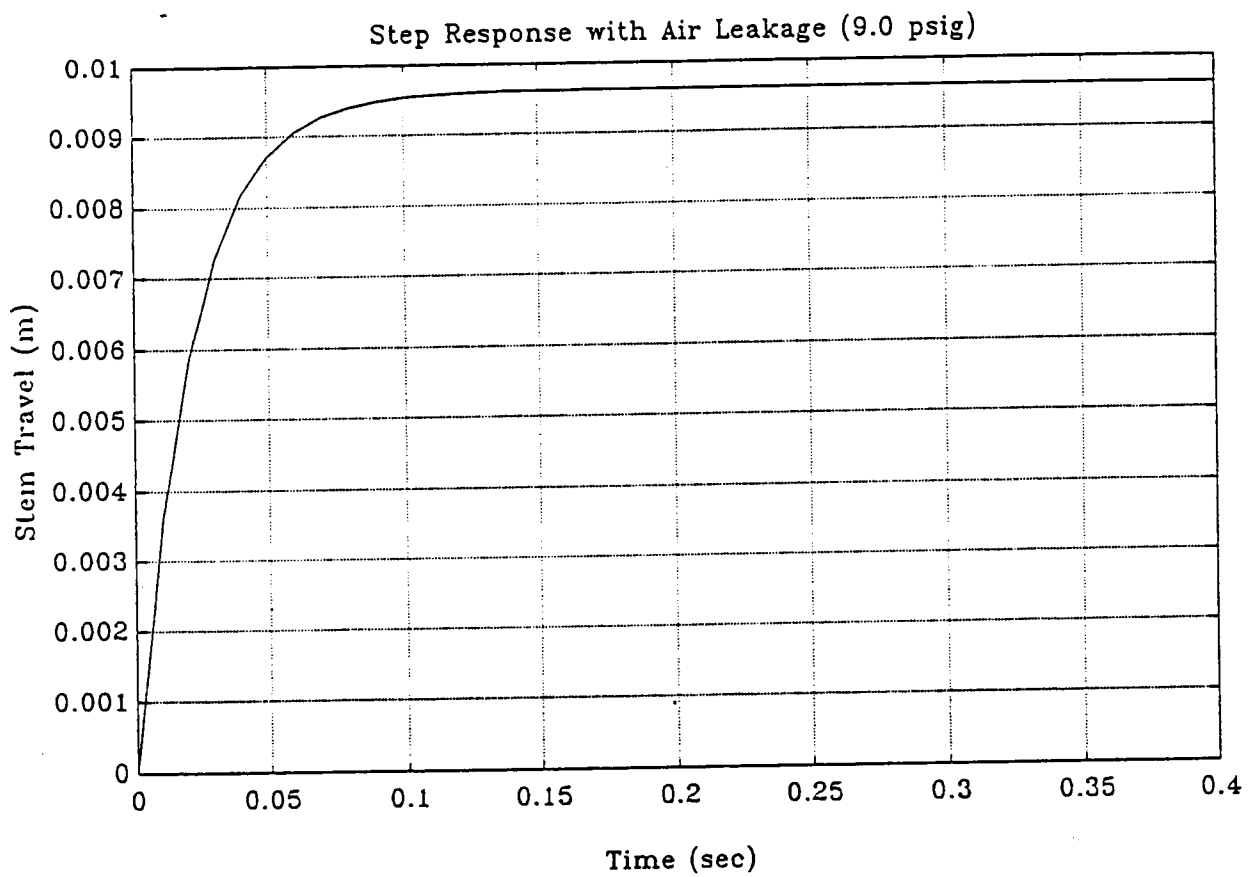
Actuating pressure at 10.8 psig

Figure 5.7. (Continued)



Actuating pressure at 9.6 psig

Figure 5.7. (Continued)



Actuating pressure at 9.0 psig

Figure 5.7. (Continued)

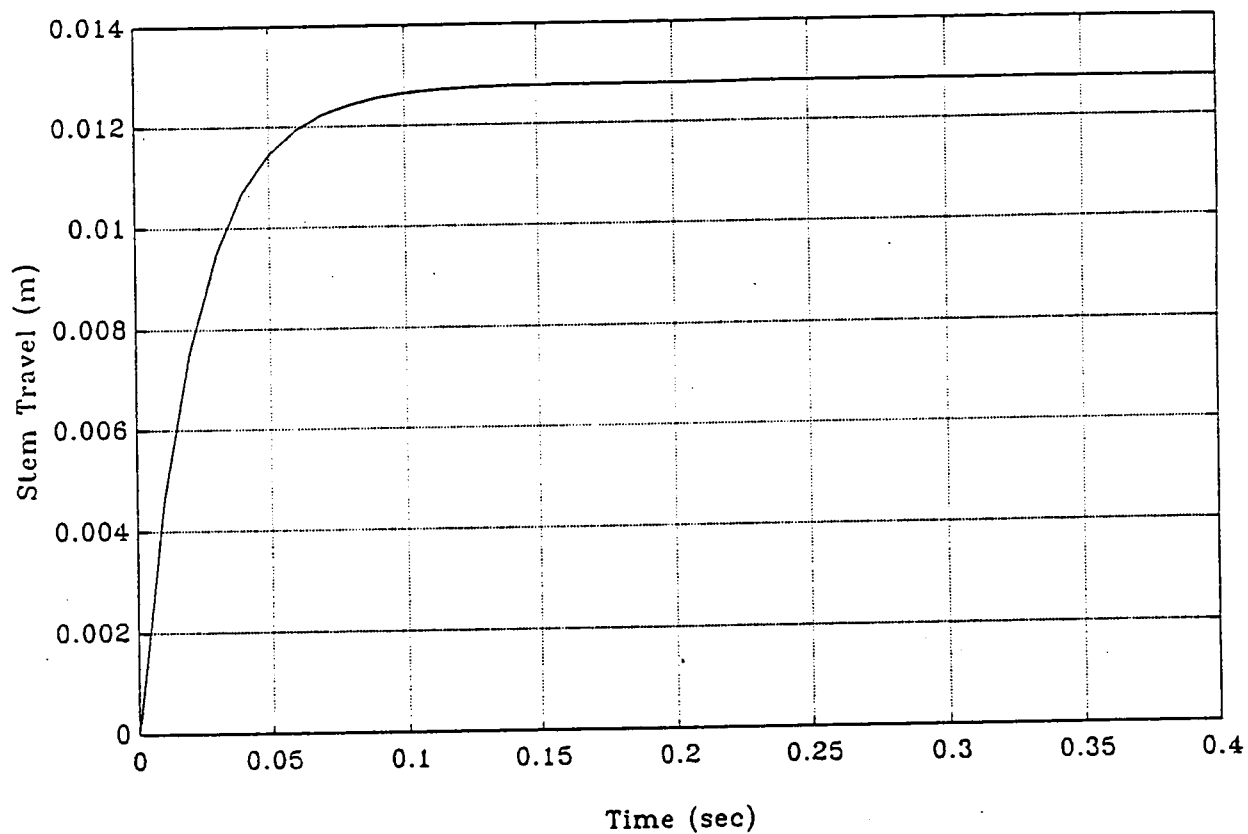
5.5.2 Response of the Model with Increased Stem Friction

The friction variation is simulated by increasing the friction coefficient (c) in steps of 5 % of its original value and computing the corresponding pressure step response functions.

The responses of the model to a step input with friction variation anomaly is given in Figure 5.8. The step pressure input is kept constant while the friction coefficients are changed during each calculation. As the friction increases, the final stem travel is the same, but the time constant changes. This indicates that friction changes result in delay in the stem travel for increased friction. Figure 5.9 shows the ramp response of the model with friction variations. As the friction increases, the stem travel will be delayed although the same ramp input is used. As the friction increases in steps of 5 %, the stem travel time is delayed by about 0.1 ms.

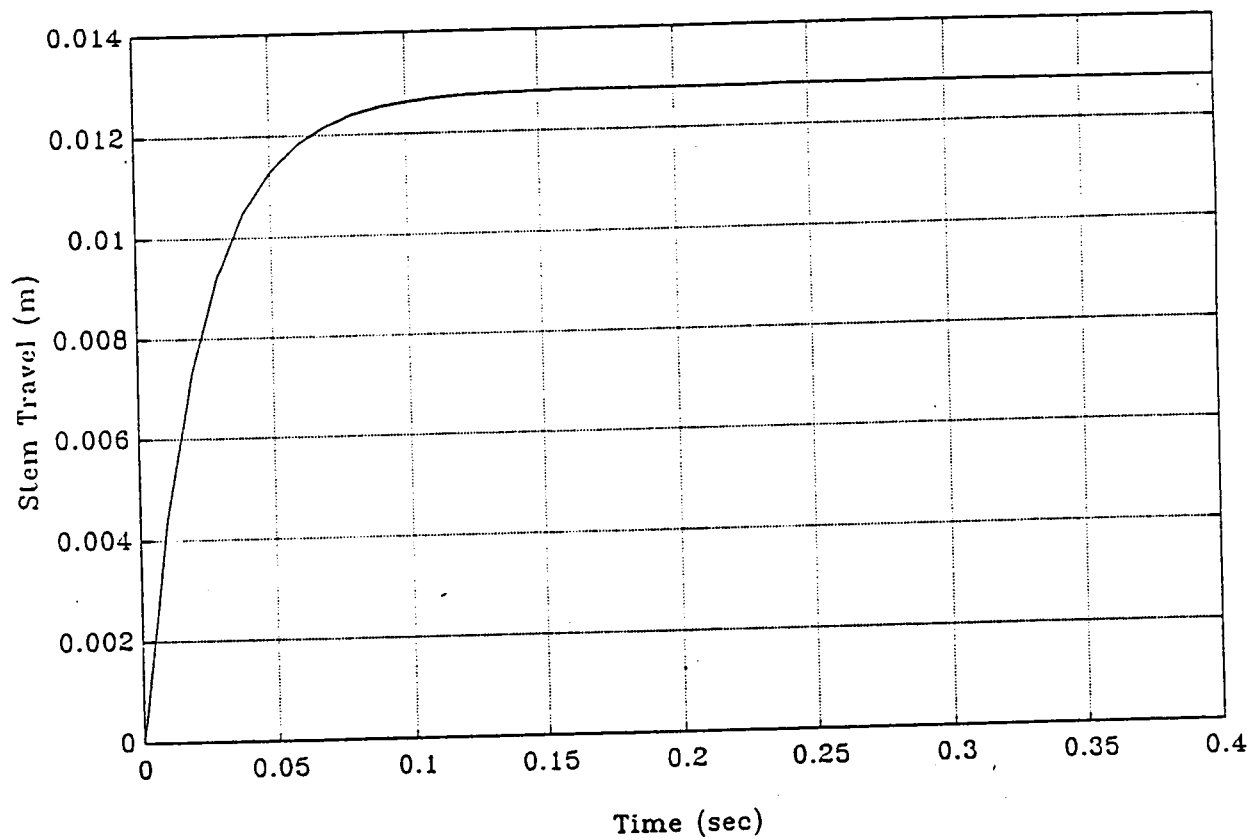
5.5.3 Response of the Model with Spring Constant Degradation

The response of the model to a step input with spring constant degradation is also studied. The simulation result is shown in Figure 5.10. The spring degradation is simulated by reducing the spring constant. As the spring constant is decreased, the stem travel increases even though the acting pressure is kept constant. If one of the six springs in the test valve is damaged, the stem travel will increase by about 4 mm. This corresponds to a pressure of 12 psig to complete a full stroke of the test valve.



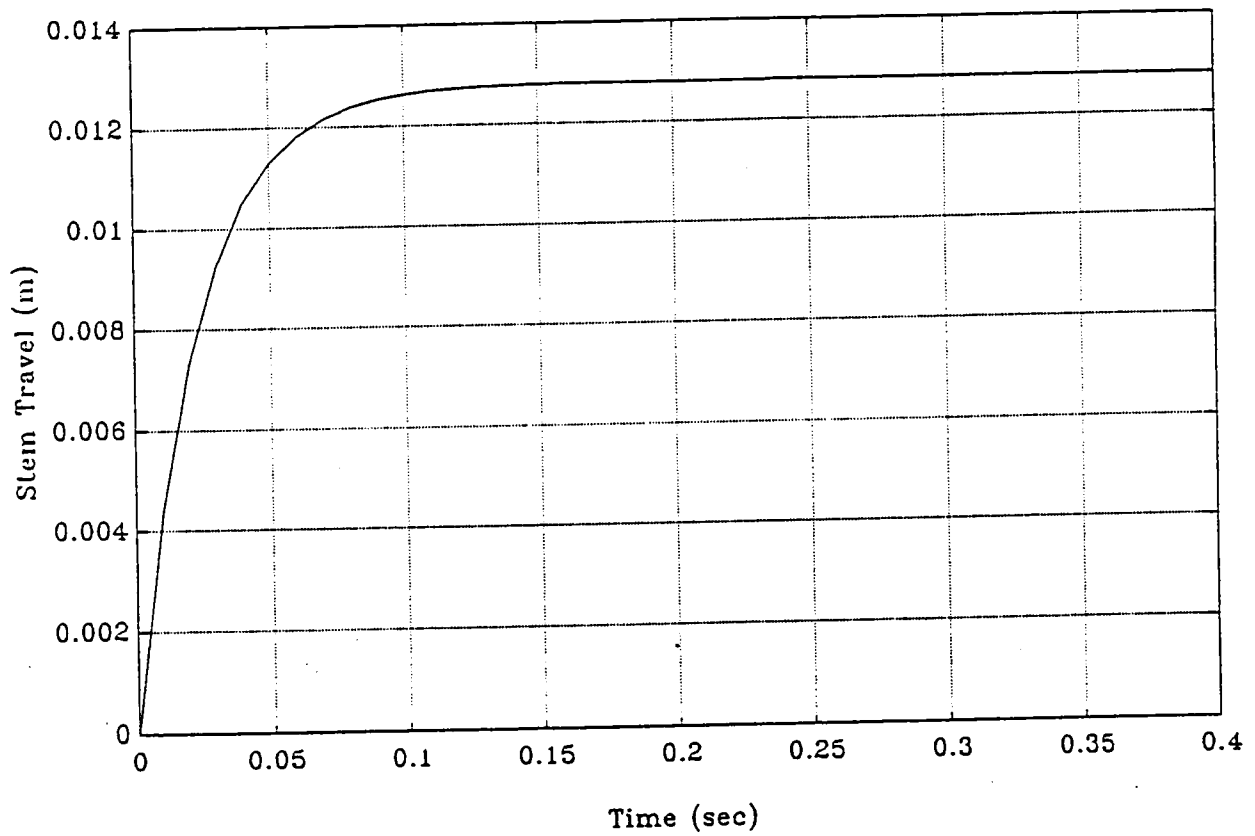
Friction coefficient (c) = 3780

Figure 5.8. Model step response with friction increase.



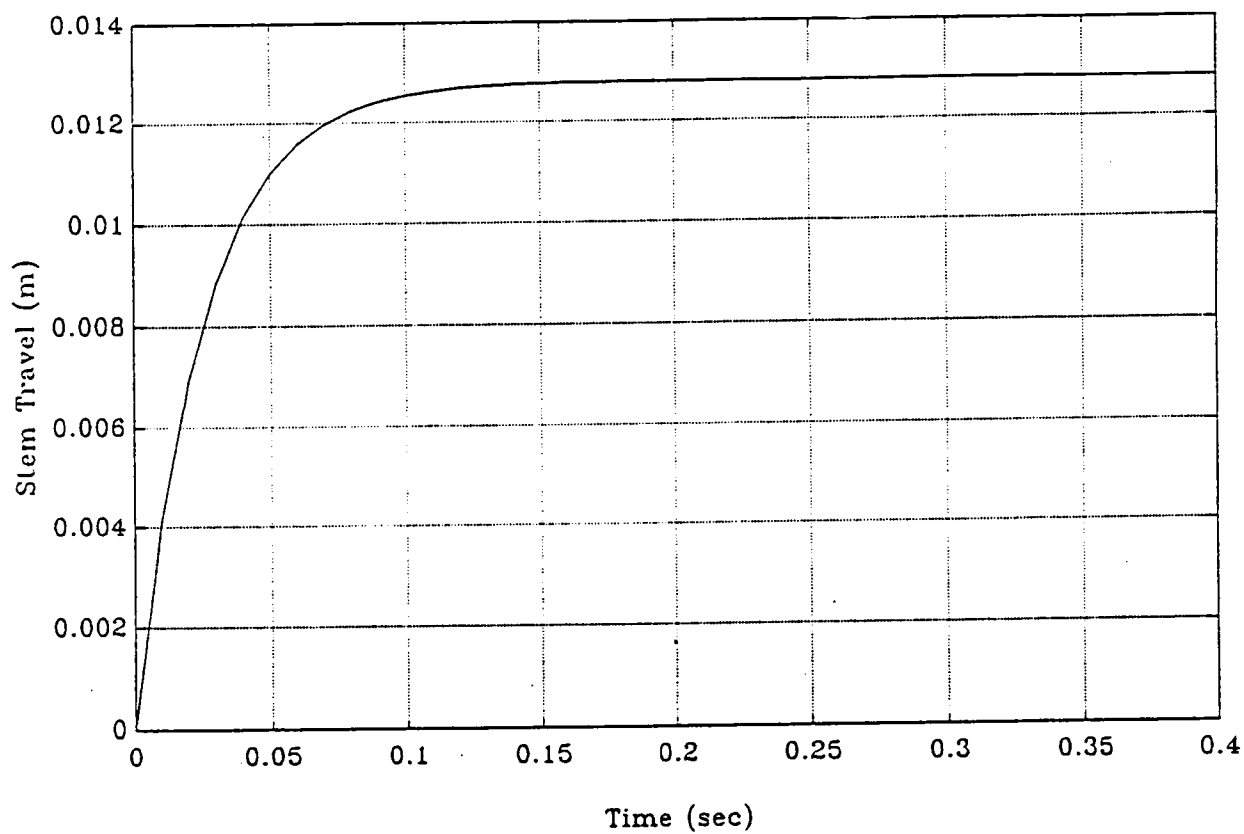
Friction coefficient (c) = 3960

Figure 5.8. (Continued)



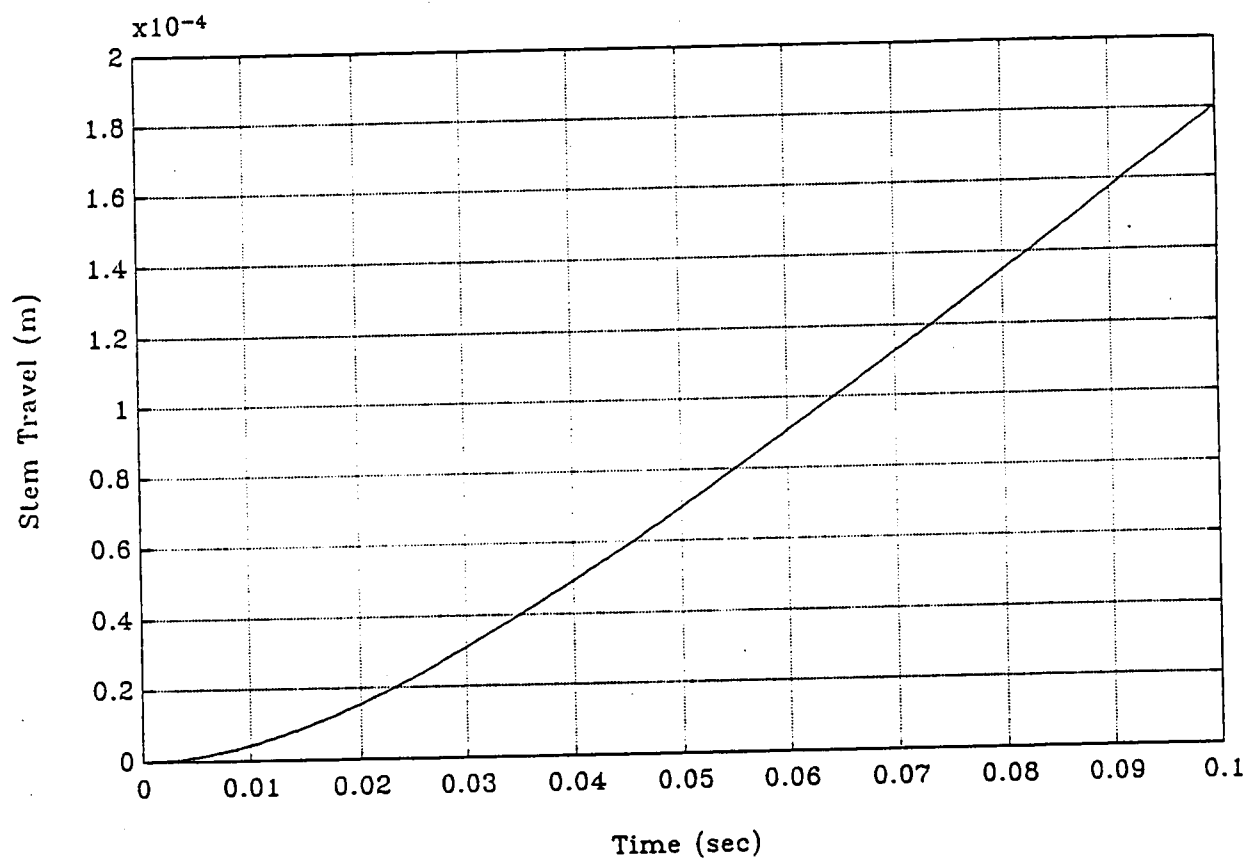
Friction coefficient (c) = 4140

Figure 5.8. (Continued)



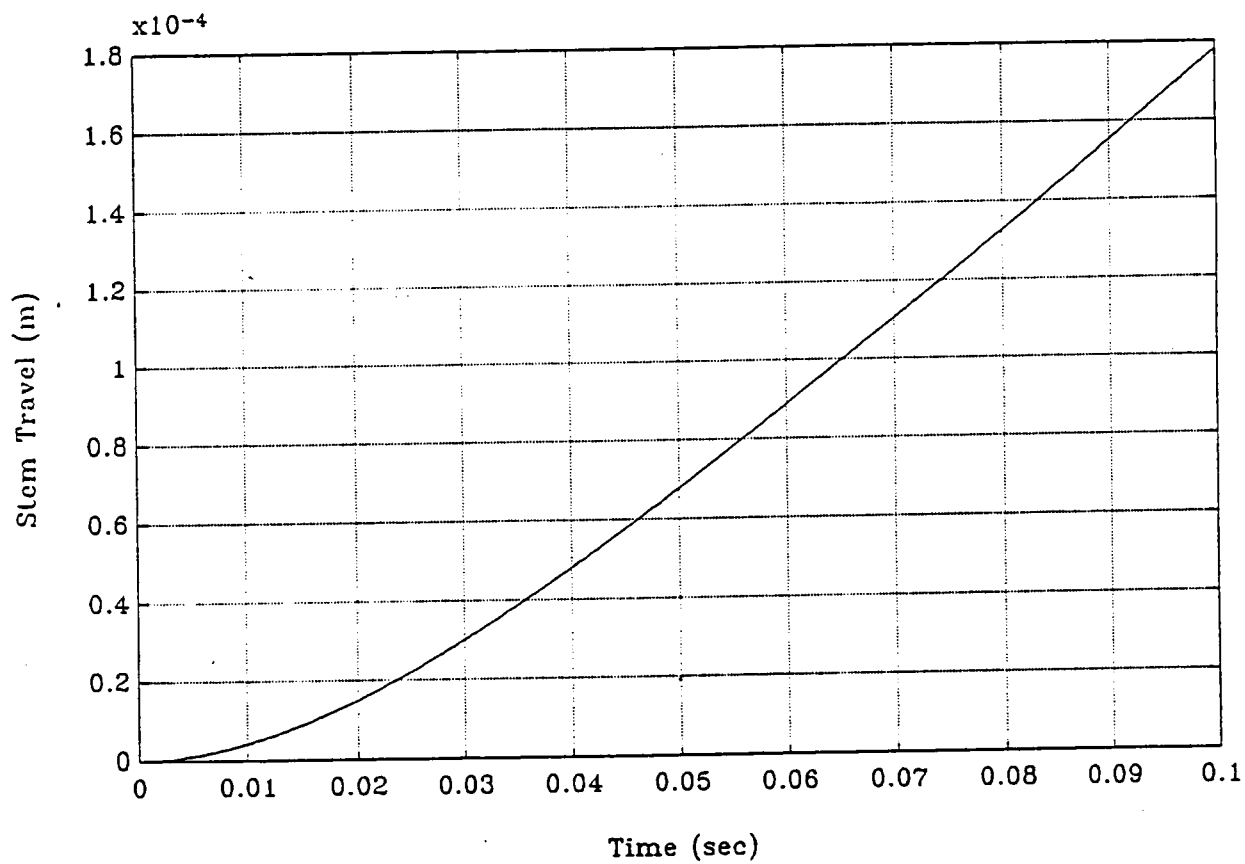
Friction coefficient (c) = 4320

Figure 5.8. (Continued)



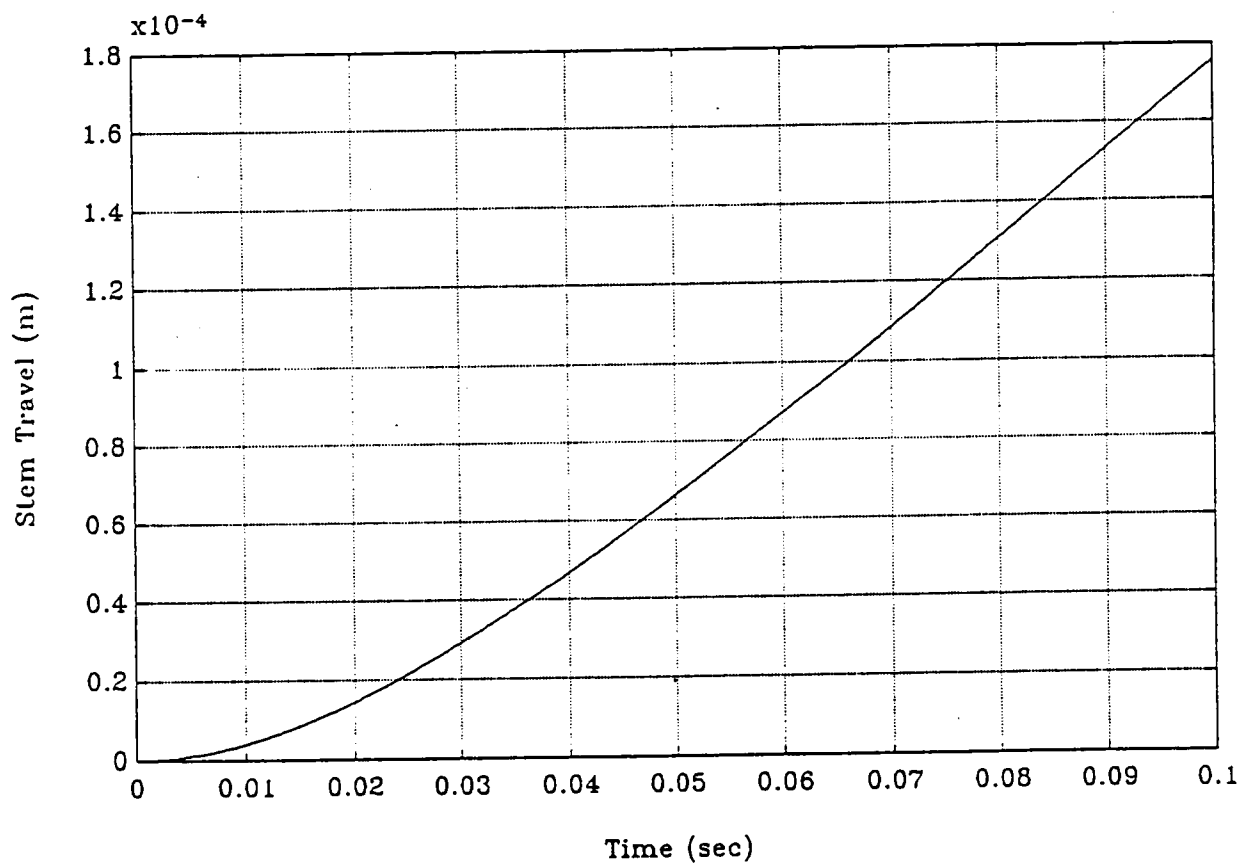
Friction coefficient (c) = 3780

Figure 5.9. Model ramp response with friction increase.



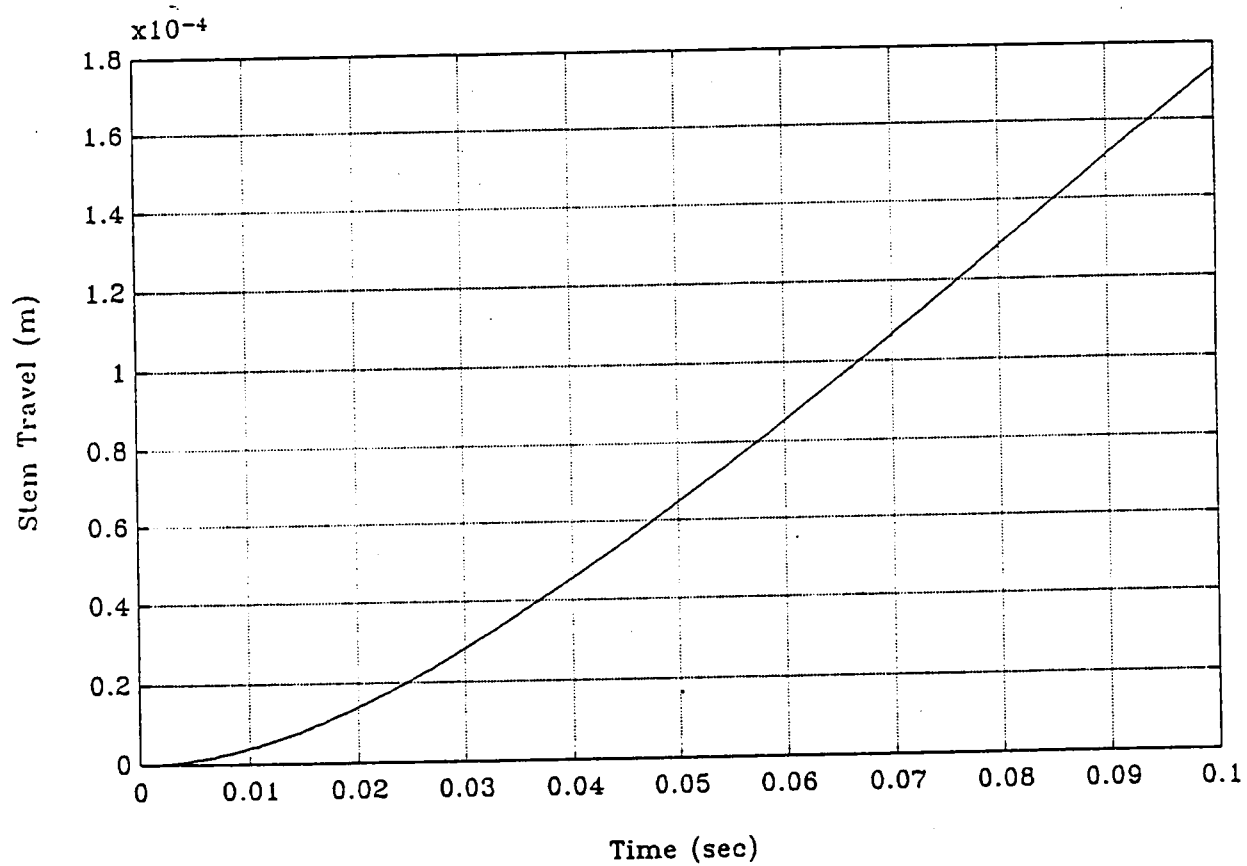
Friction coefficient (c) = 3960

Figure 5.9. (Continued)



Friction coefficient (c) = 4140

Figure 5.9. (Continued)



Friction coefficient (c) = 43290

Figure 5.9. (Continued)

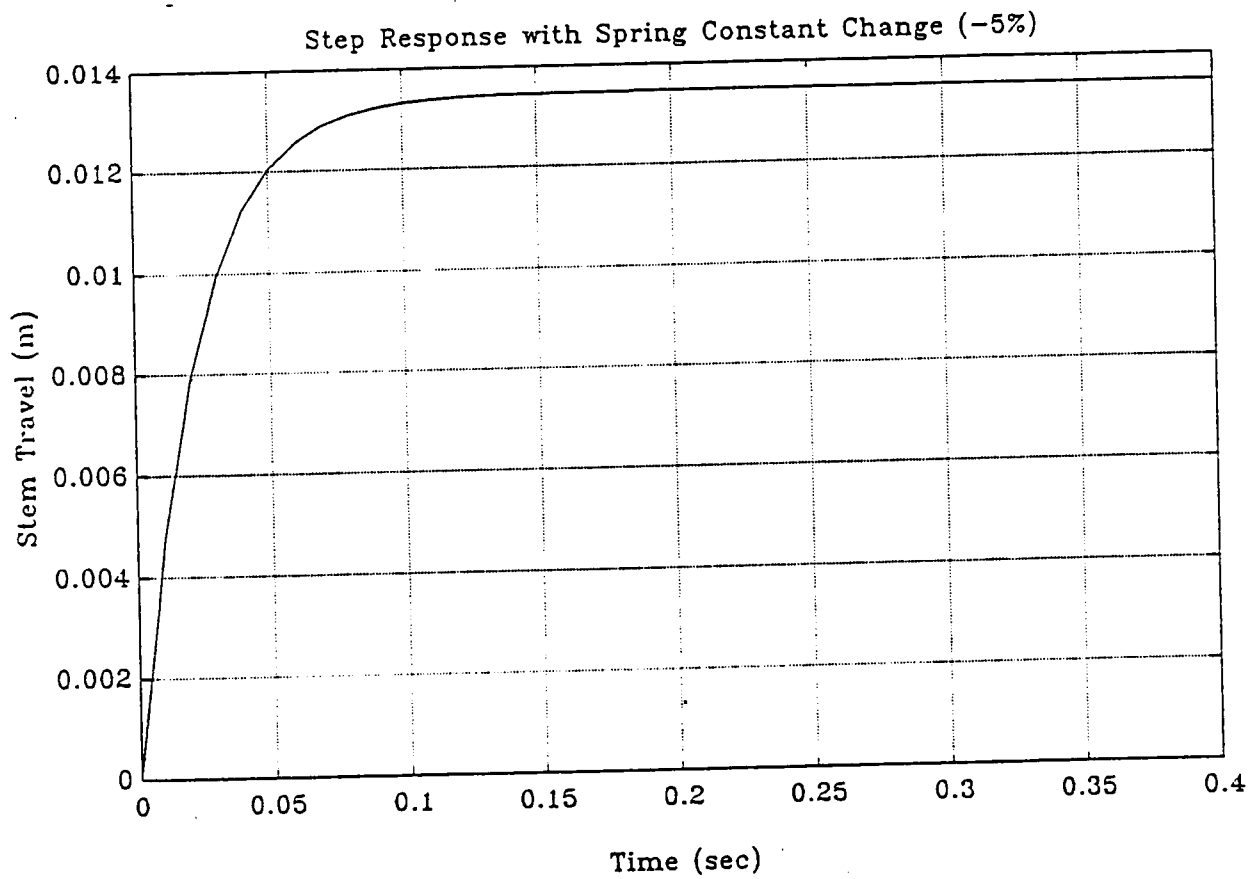


Figure 5.10. Model step response with actuator spring degradation.

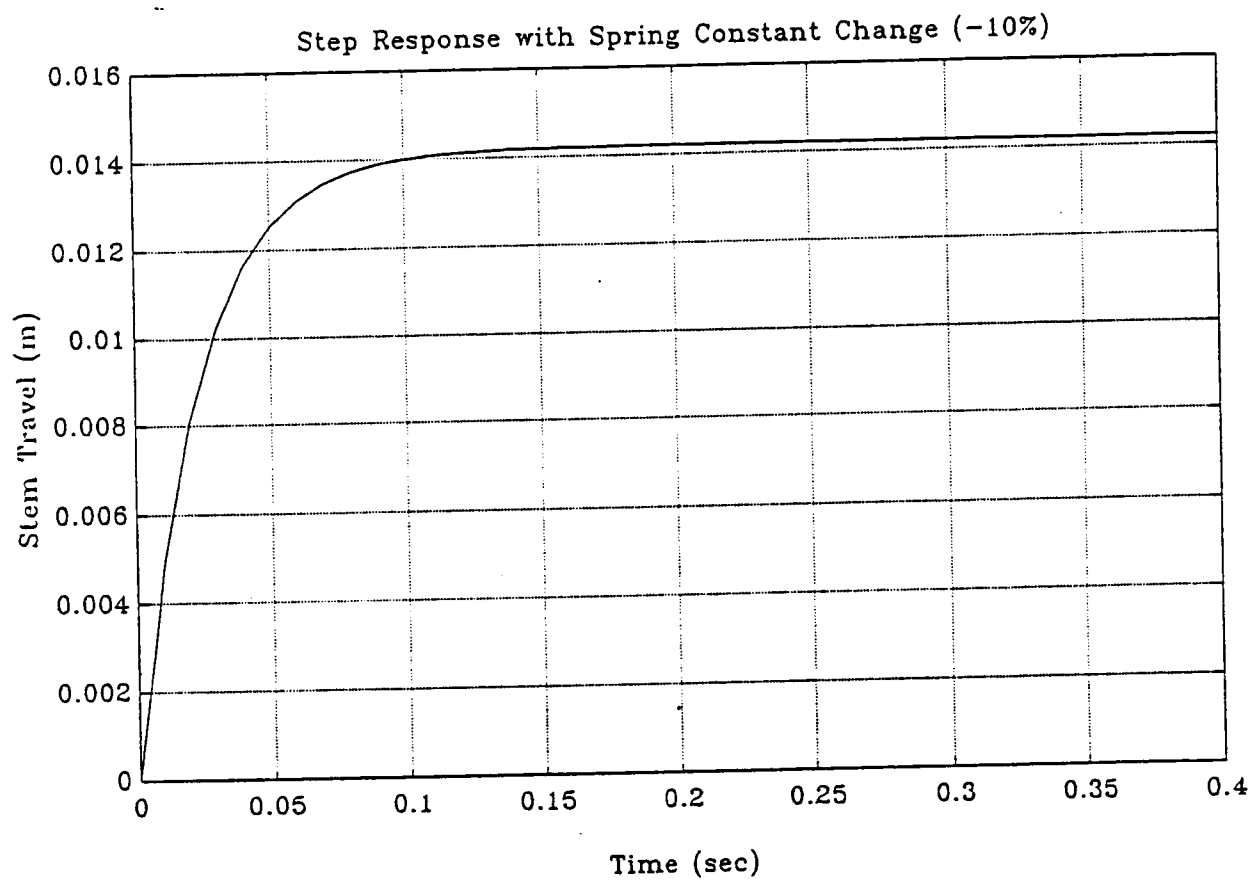


Figure 5.10. (Continued)

5.6 Comparison of Model Simulation and Experimental Analysis

The model responses to step and ramp pressure inputs are compared with the experimental measurements of valve stem travel. These responses match very well. The stem travel traces to different pressure inputs in the experimental analysis are shown as the voltage outputs of the LVDT. Using the calibration constants, at the fully open position, the total stem travel is found to be about 1.6 cm (measured) corresponding to a pressure change from 3 to 12.5 psig. The model response indicates a stem travel of 1.3 cm (Figure 5.5). The ramp response of the model (Figure 5.6) compares very well with the experimental result (Figure 3.14). These results indicate that the valve model is a valid representation of the actuator system.

The results of the model simulation with various anomalies indicate clearly the changes in the stem displacement to different perturbations. The degradation of actuator spring assembly is also simulated using the valve model. The simulation results show that the model represents the actuator system very accurately, even for the degraded case.

CHAPTER VI

A METHODOLOGY FOR MONITORING AIR-OPERATED VALVES

The primary objective of the research and development under this project is to perform experimental and analytical studies of AOV condition monitoring, and to identify one or more variables that are sensitive to the changes in the behavior of AOVs. It is shown in the present study that the measurements, valve differential pressure and stem motion as measured by the linear variable differential transformer (LVDT), reflect the changes in the AOV performance.

Both valve transient test data and the steady-state fluctuation data were analyzed, for normal valve operation and for the case of anomalies. These include air leakage and increased stem friction. The following characteristics reflect the valve behavior:

1. Step response to a change in the actuation air pressure. Stroke time, stem travel, and the rate of response may be calculated for both the valve DP and the stem motion signals.
2. Plots of air pressure vs. valve DP, and air pressure vs. stem motions (LVDT) show varying characteristics with changing performance due to air leakage and friction parameters. The hysteresis effects are seen in the area enclosed by these plots. Changes in these parameters provide a quick indication of changing valve behavior.

3. Valve testing with ramp pressure input can also be performed. Since ramp input is more gradual than the step input, a different valve characteristic is exhibited by this test. Again ramp delay time as given by valve DP and LVDT signals may be measured and monitored.
4. Random data analysis was performed during steady-state operation of the AOV. The stem motion is limited during these measurements. These tests are very useful in estimating the degree of modulation in the flow pressure caused by valve movements.
5. Various time-domain and frequency-domain signatures may be monitored to establish possible anomalies in an AOV. This study showed that signal RMS (root-mean-squared) values are very sensitive to valve stem modulation. It is recommended that other quantities such as auto- and cross-power spectral densities, coherence functions, and amplitude probability density functions should be monitored to establish any cause and effect relationship.

Univariate autoregression time series modeling was also performed using the stationary random data. This technique is useful in processing short data samples, and for estimating signal power spectrum and step response functions. The response time may be calculated and monitored for possible changes. No system perturbation is necessary in this approach.

6. A preliminary analysis using a second order model of the AOV under test must be performed. This is useful in understanding the basic response characteristics of the valve, and also for designing valve tests.

The test results from the transient and random noise signal analysis may be compared with a baseline database, and decision about the status of the valve can be made. This procedure can be automated in a PC-based diagnostic system. The valve design can be modified to include measurements of stem motion and valve DP signals. If the price of this modification is prohibitive, at least such a design change must be considered for safety-related valves in nuclear and chemical process systems.

CHAPTER VII

CONCLUSIONS AND SUGGESTIONS FOR FURTHER RESEARCH

7.1 Summary

Various problems related to air-operated valve (AOV) operations in nuclear power plants were reviewed and typical valve related events were summarized. To develop AOV condition monitoring techniques, an AOV test loop was constructed. The valve operations were monitored using an absolute pressure gauge, a linear variable differential transformer (LVDT), and a differential pressure gauge. These sensors were installed, calibrated, and interfaced with a PC-based data acquisition system. A comprehensive transient study of AOVs is presented in this thesis. The responses of the system to different pressure perturbations for both normal and abnormal conditions are studied. The anomalies of the test valve can be identified from the results of transient analysis. A preliminary analysis of random signal components was performed to detect the changes in AOV characteristics. A detailed analytical study of AOV dynamics was performed. A linearized model of the stem displacement dynamics was also developed. The model was used to simulate both normal and abnormal operating characteristics of an AOV. Furthermore, model studies may be used to verify experimental results.

7.2 Concluding Remarks

The results of studies reported here have shown that the operating characteristics of AOVs can be monitored by measuring air supply pressure, stem position, and valve differential pressure. These measurements are sensitive to degradations and anomalies of AOVs. Random signal analysis, transient analysis, and analytical model simulations are very good tools for solving valve monitoring and diagnostic problems.

Transient results clearly indicate AOV characteristics for various operating conditions. Different transient pressure perturbations may be used in identifying the characteristics of the test valve. The responses of the system to a step pressure perturbation show the dynamics and valve degradations more clearly compared to the responses to ramp or sinusoidal pressure perturbations. These tests provide more information for identifying different anomalies. No obvious difference is found in the responses to slowly applied ramp and sinusoidal pressure perturbations. The slope and shape of the hysteresis plots, especially at the beginning and at the end of the strokes, are very useful in studying the valve degradation and anomaly diagnostics. The simulations of air leakage and friction effects are clearly seen in the transient analysis plots.

Random signal analysis during valve steady-state operation indicates the feasibility of applying this technique for monitoring valve condition. Both time-domain and frequency-domain signatures are applied in this project.

The analytical modeling of the test valve operation is useful in simulating

model response to various pressure perturbations. The model is also effective in simulating various valve anomalies and for designing experiments.

7.3 Suggestion for Implementation of AOV Condition Monitoring Techniques in Nuclear Power Plants

The basic methodology for condition monitoring of AOVs is presented in this thesis. For effective diagnostics of AOVs in operating power plants, a monitoring system must consider the following:

1. Identification of important AOVs for monitoring in a power plant.
2. Selection of variables and sensors. The measurements of air line pressure, stem position, and pressure drop across the valve must be considered. The sensors should be selected based on the applications.
3. Selection of appropriate signal conditioning hardware for both transient and random data acquisition.
4. Development of a data analysis and automated diagnostics software system.

A methodology for monitoring AOVs, based on valve stroking and normal fluctuation signal analysis is outlined in Chapter 6.

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APPENDIXES

APPENDIX A

TYPICAL AOV ANOMALIES IN NUCLEAR POWER PLANTS

1. BWR, Operator: Commonwealth Edison, Co. (LER 31)

Power level 93%. On August 21, 1987, the reactor scrambled on a low reactor water level signal of +8 inches. The root cause of the event was the failure of the 2a feedwater regulating valve (FWRV). The stem and plug assembly of the 2a FWRV separated due to a fatigue crack.

2. BWR, Operator: Commonwealth Edison Co. (LER 32)

Power level -0%. On May 17 of 1988, for a scheduled maintenance outage, during the performance of a required procedure, all eight unit 2 main steam isolation valves (MSIV) were declared inoperable due to their failure to fully close on a loss of pneumatic supply. The root caused of the MSIV failure to fully close has been attributed to high drag forces acted on the valve stem by the valve packing.

3. PWR, Operator: Southern California Edison Co. (LER 6)

Power Level 92%. Plant procedures require that each loop of the safety injection system be vented on quarterly basis in order to avoid a water hammer in the event of system actuation. Upon the completion of venting of the loop B header on 9/10/87, it was noted that the remotely operated, normally closed, containment

isolation valves in the vent line for that header initially did not fully close upon demand. The valves failed to fully closed as the result of worn and warped valve seats.

4. PWR, Operator: Virginia Electric & Power Co. (LER 140)

Power level 23%. The feed flow to "A" steam generator increased to full flow. The operator was not quick enough to close the valve and the "A" steam generator high level trip initiated a turbine trip, which tripped the reactor. An elbow in the air supply piping to the "A" feed regulating valve was damaged and leaking. The 1/2 inch line was not secured and vibration had caused it to hit against its support.

5. PWR, Operator: Southern California Edison Co. (LER 5)

Power level 92%. On 5 occasions from 3/87 to 9/87, during in-service testing, 4 air operated valves (one containment isolation valve, one containment spray valve, and 2 letdown isolation valves) failed to stroke. On each occasion, the normally-energized automatic switch company 3-way solenoid air valve had failed to vent from the valve operator. Inspection of the solenoid valves revealed that there was a thin hard film that had formed between the top of the slug and slug housing.

6. PWR, Operator: Florida Power & Light Co. (LER 69)

Power level 0%. On August 24, 1986, during performance of test of "Auxiliary Feedwater Train 2 Operability Verification", the AFW flow control valve to the "A"

steam generator failed to open fully, therefore, one train of AFW was not available to the "A" steam generator. The valve opened to about 70% of fully travel instead of the required 80% to 85%. The valve failed to open fully due to the current to pressure(I/P) convertor having a terminal with some corrosion resulting in high electrical resistance and lower than required current. The low current resulted in low air pressure to the valve operator, with the consequent failure to open fully.

7. PWR, Operator: Wisconsin Public Service Co. (LER 189)

Power level 100%. On 11/13/1985, the "A" train main feedwater control valve failed shut causing a low-low water level in SG 1A, and subsequent reactor trip. Two of the four cap screws, which hold the valve actuator to the yoke on the 1A main feedwater control valve, sheared causing the valve to fail shut. The failure mechanism of the cap screws is suspected to be low cycle stress fatigue.

APPENDIX B

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APPENDIX C

MAJOR COMPONENTS AND SPECIFICATION OF A LINEAR VARIABLE DIFFERENTIAL TRANSFORMER

The major components of LVDT are: DC power supply, carrier exciter, transducer, demodulator, and amplifier. The following description of a LVDT transducer instrumentation is borrowed from vendor catalog, Schaevitz catalog # 701 [20].

A. Excitation

A LVDT requires constant-amplitude AC input, at a frequency not readily available. This means that an oscillator of the appropriate frequency must be connected to an amplifier with amplitude regulation on its output. Usually the excitation output is a constant voltage, but special circumstances may require a constant current.

B. Amplification

A LVDT's output is usually too low-level to operate ordinary readouts, so it is necessary to amplify the output of the LVDT. Sometimes this is done in two steps, using an AC carrier-amplifier prior to demodulation and a DC amplifier after demodulation.

C. Demodulation

Most readouts are DC-operated devices, so the amplified AC output must be converted into filtered DC before it can be used with ordinary DC-type readout devices. Furthermore, taking advantage of the phase polarity of an LVDT requires the use of a synchronous demodulator. Demodulators are usually followed by one or more stages of low-pass filtering.

D. DC Power

All of the electronic circuits require stable DC voltages for proper operation.

LVDT Specification:

Power requirement: 115 VAC rms

Transducer excitation: 3V rms nominal

Excitation frequency: 2.5 KHz (carrier frequency)

Full scale output: ± 10 V DC maximum

Operating temperature range: 30 °F to 130 °F (0-55°C)

VITA

I was born on October 1, 1961, in JangSu, China. I graduated from Nanjing Institute of Chemical Technology, in August, 1982 with a Bachelor of Science degree in Mechanical Engineering. I had been a mechanical engineer in Beijing Institute of Nuclear Engineering for six and half years before I came to U. S. A. to pursue higher education. In Spring of 1989, I entered the University of Tennessee, Nuclear Engineering Department, as a full-time graduate student. I received a research assistantship for two years and two months during which I completed this thesis. In May 1991, I expect to be awarded the Master of Science degree with a major in Nuclear Engineering and a minor in Mechanical Engineering in May 1991.