

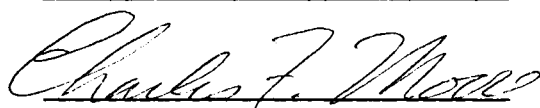
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THE BURP TEST: A METHOD FOR IN-SITU RESPONSE-TIME TESTING
OF PRESSURE-SENSING LINES IN NUCLEAR POWER PLANTS

A Thesis
Presented for the
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Degree
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Guy Elliott Ragan

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ABSTRACT

Commercial nuclear power plants are equipped with instrumentation designed to detect unsafe conditions and, if required, to initiate protective action. The time elapsed between the realization of an unsafe condition and the initiation of protective action is known as the "response time" of the instrumentation involved. The Nuclear Regulatory Commission (NRC) has issued guidelines which advise periodic in-situ "response-time" testing of "safety-related" instrumentation.

No method is presently available for in-situ response-time testing of "pressure sensing lines" (fluid-filled tubes connecting process to pressure transducer). A proposed method of doing just that is investigated in this thesis. The theoretical basis of the method is presented, followed by a description of the experimental investigation. It is concluded that, although the method appears to work, it is not justified in light of practical considerations.

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INTRODUCTION

Commercial nuclear power plants are equipped with instrumentation designed to detect unsafe conditions and, if required, to initiate protective action. The time elapsed between the realization of an unsafe condition and the initiation of protective action is known as the "response time" of the particular instrument channel involved. To insure the safety of the plant, the response time of all "safety-related" instrument channels must be less than a certain maximum for each channel.

The Nuclear Regulatory Commission (NRC) has issued guidelines [1] which advise periodic response-time testing of "safety-related" instrument channels. There have been recent successes in developing methods of in-situ response-time testing of thermometric instrumentation [2] and pressure sensors [3], but no such methods have been developed for pressure sensing lines (tubes for transmitting pressure signals from process to sensor). The present thesis investigates a method of in-situ response-time testing of pressure sensing lines: the "burp test." The theory behind the method is presented and the experimental development of the method is described. Although the method appears to work, it is found to be unnecessary and impractical.

CHAPTER 1

FUNDAMENTAL CONCEPTS AND DEFINITIONS

Nuclear Power Plant Instrumentation

For present purposes an "instrumentation and control system" is defined to be a system which measures and controls the operation of a process. In the case of a nuclear power plant, the process is that of producing electrical power from the nuclear fission of the fuel. An instrumentation and control system may be considered to be composed of distinct "channels." An instrumentation channel extends from the measurement of a particular operating parameter to the control action generated in response to measurement of that particular operating parameter.

Instrumentation Components as Input-Output Systems

Any component of an instrumentation channel can be viewed as an input-output system. For example, consider a pressure transducer. The input is the pressure (or pressure difference) at the input connection. The output is the electrical signal generated. However, taking a step back, the situation may not be quite that simple. Connecting the transducer (or sensor) to the location at which the pressure is to be measured (the process), there is a section of fluid-filled tubing known as the sensing line. If the dynamics of sensor and sensing line are such that they may be considered individually, this can be indicated by drawing a block diagram of the system as in Figure 1-1. However, in some cases it is necessary to consider various interactions among components as in for example, Figure 1-2. In the latter situation, the components having significant interactions must be considered as a whole.



Figure 1-1. Simple Pressure Measurement System.

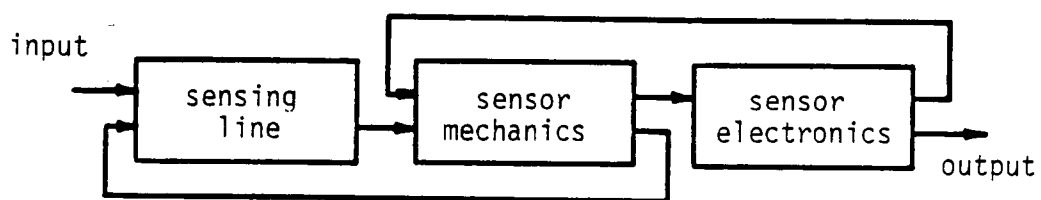


Figure 1-2. Complex Pressure Measurement System.

The Concept of Asymptotic Ramp Delay Time

A system with one input and one output may be depicted as shown in Figure 1-3.

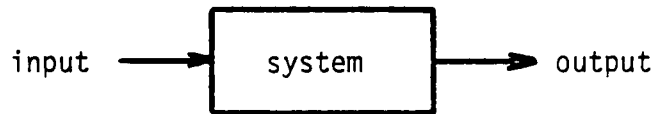


Figure 1-3. Input-Output System.

Suppose that the input $x(t)$ is an unterminated ramp as defined below:

$$x(t) = \begin{cases} 0, & t \leq 0 \\ at, & t > 0 \end{cases}$$

where

a = ramp rate constant,

t = time.

The response $y(t)$ of an asymptotically-stable, linear system will asymptotically approach the line

$$\hat{y}(t) = ka(t - \tau)$$

where

k = gain constant,

τ = asymptotic ramp delay time.

Assuming that the gain is unity, or that proper scaling has been done, τ can be measured after a suitable lapse of time as shown in Figure 1-4. The gain and asymptotic ramp delay time are both dependent on the system,

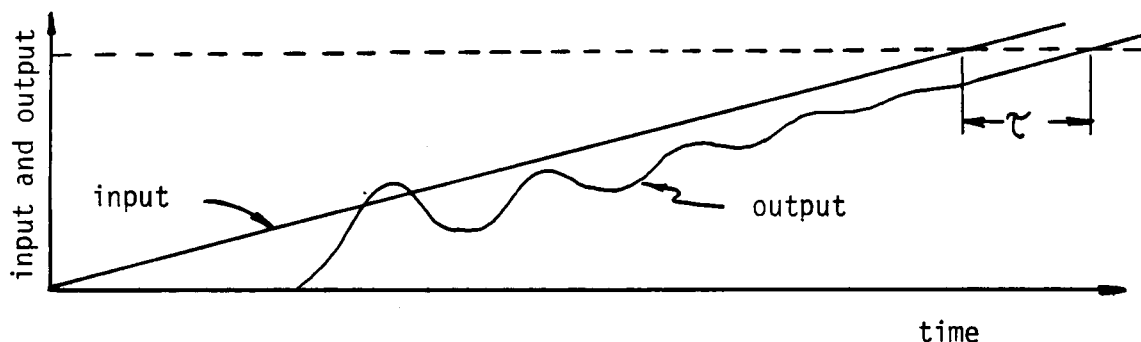


Figure 1-4. Illustration of Asymptotic Ramp Delay Time.

but both are independent of the ramp rate (for any asymptotically-stable, linear system). The term "ramp delay time" will be used to denote τ , but the term "delay time" will be avoided since it can lead to confusion.

Dynamic Testing of Nuclear-Safety-Related Instrumentation

When a nuclear power plant is designed, or when major modifications are made, safety analyses must be performed. Certain "design basis" accident scenarios are simulated using a mathematical model of the system. In the course of an accident simulation, a certain assumption is made as to the time required for an instrumentation channel to respond to an unsafe condition. For the conclusions of the safety analysis to be valid, the assumed response time of the channel must not be exceeded by the actual instrumentation. It is for this reason that dynamic testing of all "nuclear-safety-related instrumentation" is mandated by the NRC [1].

Nuclear-safety-related instrumentation is defined [4] to be instrumentation

which is essential to:

1. emergency reactor shutdown
 2. containment isolation
 3. reactor core cooling
 4. containment or reactor heat removal
 5. prevention or mitigation of significant release of radioactive material to the environment
- or is otherwise essential to provide reasonable assurance that a nuclear power plant can be operated without undue risk to the health and safety of the public.

The purpose of dynamic testing is to determine the "response time" of the instrument channel, defined to be [4] "the time interval from when the monitored variable exceeds its trip setpoint until a protective action is initiated." The testing may be done in a variety of ways. It is preferable that the entire instrument channel be tested in a single test. But testing of individual components is permissible provided that [4] "the response time obtained by adding the individual response times [is] greater than or equal to the actual instrument channel response time."

Tests are classified as either "direct" or "indirect." A direct test is one which measures the response time directly, i.e., by the application of an "unterminated ramp" [4] function

(a ramp that starts at the variable initial value, becomes linear, and continues to a higher or lower value beyond the set-point of interest, such that the instrument or channel desired output signal is obtained while the input ramp is still linear)

at the input to the instrument channel. This definition reveals that "response time" and "ramp delay time" as previously defined are synonymous. In many cases, a direct test is not possible, and an indirect test must be performed. An indirect test depends upon a mathematical model of the instrument channel (or component of the channel) which relates the measured characteristic (frequency response, step response, impulse response, etc.) to the characteristic of interest, i.e., the ramp delay

time of the instrument channel. In Chapter 4 an indirect method of testing the response time of a sensing line is presented. The model used to relate the indirect measurement of Chapter 4 to the ramp delay time of the sensing line is the subject of Chapter 3. Chapter 2 amplifies certain aspects of pressure measurement in nuclear power plants.

CHAPTER 2

PRESSURE MEASUREMENT IN NUCLEAR POWER PLANTS

Nuclear-Qualification of Instrumentation

Pressure is measured in nuclear power plants in much the same fashion as in other types of power plants. The major difference is that nuclear-safety-related instrumentation is required to be "nuclear-qualified." This means that the instruments must pass a series of tests intended to simulate postulated accident conditions. These include subjection to extremes of radiation, temperature, humidity and vibration.

Types of Sensors

Many different classifications of electronic pressure sensors are possible. Here they will be classified as either (a) force-balance, or (b) motion-balance. In either case there is a flexible element which deforms in response to a differential pressure. This element may consist of a diaphragm, bellows, Bourdon tube, or other flexible element. Next, in both cases, there is a means by which the displacement is converted into an electrical signal. This may be done by exploiting the geometric dependence of the inductance, capacitance, or resistance of specially designed circuit elements. The classification of the sensor as force- or motion-balance type depends on the fate of the electrical signal thus generated. In a motion-balance sensor, this signal itself, after amplification or other transformation, serves as the sensor's output signal. In a force-balance sensor this electrical signal is used as an error signal which actuates a force motor which, in turn, restores the flexible element to its nominal position. The output signal of the force-balance

sensor is the current supplied to the force motor which is required to balance the force due to the applied pressure differential.

Applications of Pressure Measurement

Pressure sensors are used in nuclear power plants to measure gage pressure directly, for example, coolant pressure and containment building pressure. Pressure sensors may also be used indirectly to measure flow rates and liquid levels. In any case a sensing line (or pair of lines) is required to transmit the pressure signal from the process to the sensor. Frequently, the sensing line is required to be rather long (on the order of 100 ft.) so that the sensor can be conveniently located for maintenance and testing purposes.

Sensor Dynamics

Pressure sensors can be complicated devices. They may consist of many subsystems with significant dynamic characteristics which contribute to the overall behavior of the system [3]. From the point of view of a sensing line, however, the sensor's flexible element which contacts the fluid may be the only part of the sensor which affects the sensing line significantly. This is the point of view taken here. That is, the only significant interaction between sensor and sensing line if any, which need be taken into account, is that between the flexible element of the sensor and the sensing line.

Sensing Line Dynamics

Chapter 3 is entirely devoted to a theoretical description of sensing-line dynamics. An overview of general principles will be presented here.

There are at least four characteristics of importance to the dynamic behavior of sensing lines:

1. the viscosity of the fluid,
2. the compressibility of the fluid and the distensibility of the tube,
3. the inertia associated with fluid motion, and
4. the "boundary conditions" at each end of the line.

The first characteristic is a mechanism of energy dissipation, while the second two are means of energy storage. Stored energy can be converted from one form to the other in an oscillatory fashion much like the interchange between potential and kinetic energy in a spring-mass system. However, (unlike the case of the idealized spring-mass system) the potential and kinetic energies in a sensing line are distributed throughout the length of the line. Thus, a more pertinent analog is a vibrating string. Like a vibrating string, a sensing line has an infinite number of resonant frequencies. In many cases, the natural viscous damping in a sensing line prevents resonant behavior. Sometimes, however, the viscous damping is intentionally increased in order to reduce unwanted resonances. A pulsation dampener (snubber) may be installed for this purpose. A snubber is essentially a severe constriction of the flow path. Commercial snubbers are available with a self-cleaning mechanism (a little rod rattles around in the constriction).

The technique to be described in Chapter 4 for sensing-line testing exploits the resonant behavior of the line. It is therefore assumed that the sensing line is not fitted with a snubber or that only the section of line on the process side of the snubber is to be tested. It is

recognized that, in the latter case, the dynamic effect of the snubber is not taken into account. This is significant since the snubber would be an ideal place for the sensing line to become blocked.

The boundary conditions are dependent upon the way in which the line is terminated. This determines the reflective properties of the termination, which in turn determine the resonant frequencies of the line.

Sensing Line Troubles

An industry survey was conducted [5] to discover what problems were actually being experienced with pressure sensing lines in commercial nuclear power plants. Three sources of blockage were found: (a) freezing, (b) inadvertently closed valves, and (c) boric acid deposition. Freezing is possible since the lines are long and may pass out of the containment building. A valve that was shut while the process sensor was being tested, repaired, or replaced may not have been re-opened. In a pressurized-water reactor (PWR), boric acid is dissolved in the primary-coolant-loop water as a means of controlling the reactivity of the core as fuel is consumed. Since boric acid is much more soluble in hot water than in cold, it can deposit itself on the interior of the sensing line, causing partial or total blockage.

The presence of air can also degrade the dynamic performance of liquid-filled sensing lines. However, it was decided at the outset of this project to concentrate on blockage detection. The decision to disregard the dynamic effects of air bubbles was justified on the basis of the following observations:

1. due to the high solubility of air in water at typical reactor pressures, air bubbles would quickly dissolve;

2. the compressibility of air, and therefore its dynamically degrading effect, decreases with increasing pressure.

Although there may be some nuclear-safety-related pressure sensing systems which operate at low pressure (e.g., those measuring containment building pressure), problems with these systems will not be addressed. The burp test is (at best) considered to be applicable only to high-pressure (1000 psi or greater) sensing lines.

CHAPTER 3

THEORETICAL DESCRIPTION OF A PRESSURE SENSING LINE

The Water Hammer Problem

The problem of fluid transients in pipelines has received considerable attention in relation to the water-hammer problem. Water-hammer may occur when there is a sudden change in flow conditions in a pipe. The classic example [6] is a reservoir which is being drained through a pipe with a valve at its downstream end. If the valve is shut suddenly, there will be pressure oscillations in the pipe. If the time taken to shut the valve is comparable to or less than the sonic transit time of the pipe, the oscillations may be severe and potentially damaging to the pipe. The name "water-hammer" is an allusion to the sound made when this happens, which is like someone banging on the pipes.

In order to understand the water-hammer problem, it has been necessary to develop mathematical models of pipelines [7,8,9]. Even though the problem of water hammer is not particularly applicable to sensing lines, the mathematical models of pipelines are of great value. Virtually all theoretical developments up to now have concerned uniform, cylindrical pipes.

Modeling the Sensing Line

In a review paper by Goodson and Leonard [10], three basic types of mathematical model of pipelines are identified and discussed:

1. lossless fluid line model (no dissipation or heat transfer effects included),

2. linear friction model (losses proportional to mean velocity, no heat transfer effect), and
3. dissipative model (includes frequency dependent viscosity and heat transfer).

Model (1) leads to the simple wave equation

$$\frac{\partial^2 p}{\partial t^2} = c_0^2 \frac{\partial^2 p}{\partial x^2}$$

where

$p = p(x, t)$ = pressure at axial position x and time t ,

c_0 = speed of sound in the fluid.

Model (2) leads to a damped wave equation:

$$LC \frac{\partial^2 p}{\partial t^2} + RC \frac{\partial p}{\partial t} = \frac{\partial^2 p}{\partial x^2}$$

where

R , L and C are parameters to be defined later.

Model (3) leads to a wave equation with frequency dependent parameters which is not normally written explicitly in the time domain.

The model chosen to represent the sensing line in this study is of the second type since

1. the first model type (simple wave equation) does not include the resistance parameter, which (as will be demonstrated) is directly related to the asymptotic ramp delay time of the sensing line;
2. the second type accounts for all the effects of interest and has been studied thoroughly in connection with electrical transmission lines [11, 12, 13, 14].

This model will be referred to as the "fluid-transmission-line equation."

Derivation of the Fluid-Transmission-Line Equation

Momentum balance. Consider a uniform, cylindrical, fluid-filled tube as depicted in Figure 3-1. The forces acting on the incremental volume of fluid are:

1. the force caused by the pressure differential,
2. the frictional force, assumed to be proportional to fluid-flow velocity (laminar flow), and
3. the inertial force.

Thus, the momentum balance is

$$[p(x,t) - p(x + \Delta x,t)]A - Fv = \rho_0 A \Delta x \frac{\partial v}{\partial t} \quad (3-1)$$

where

$p(x,t)$ = pressure at position x at time t ,

A = cross sectional area of tube,

F = friction-force proportionality constant,

v = average axial-fluid-flow velocity (volumetric flow rate divided by flow area),

Δx = incremental length,

ρ_0 = average fluid density.

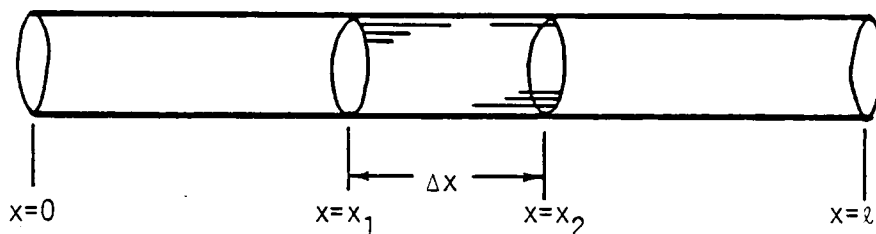


Figure 3-1. Uniform Sensing Line.

The Poiseuille equation relates the volumetric flow rate, Q , to the steady-state pressure drop, Δp , for a cylindrical tube of length Δx as follows

$$Q = \frac{\Delta p \pi r_o^4}{8 \mu_o \Delta x} \quad (3-2)$$

where

r_o = inside radius of tube,

μ_o = dynamic viscosity.

The frictional force in this case is

$$\begin{aligned} A \Delta p &= (\pi r_o^2) \frac{(8 Q \mu_o \Delta x)}{\pi r_o^4} \\ &= \frac{8 Q \mu_o \Delta x}{r_o^2} . \end{aligned}$$

Since the volumetric flow is also given by

$$Q = v A ,$$

the frictional force is

$$\begin{aligned} A \Delta p &= \frac{8 (\pi r_o^2 v) \mu_o \Delta x}{r_o^2} \\ &= (8 \pi \mu_o \Delta x) v ; \end{aligned}$$

and therefore the frictional-force proportionality constant, F , in Equation (3-1) is given by

$$F = 8 \mu_o \pi \Delta x .$$

Equation (3-1) can now be rewritten as

$$\frac{p(x,t) - p(x + \Delta x, t)}{\Delta x} - \frac{8\mu_o \pi v}{A} = \rho_o \frac{\partial v}{\partial t} . \quad (3-3)$$

We now take the limit as Δx approaches zero in Equation (3-3) to obtain

$$\frac{\partial p}{\partial x} + \frac{8\mu_o}{r_o^2} v + \rho_o \frac{\partial v}{\partial t} = 0 . \quad (3-4)$$

Continuity equation. Conservation of mass leads to the relation

mass inflow - mass outflow = rate of accumulation of mass.

Thus for the incremental section

$$v(x,t) A \rho_o - v(x + \Delta x, t) A \rho_o = \frac{\partial \rho}{\partial t} \Delta x A ;$$

or

$$\frac{v(x,t) - v(x + \Delta x, t)}{\Delta x} \rho_o = \frac{\partial \rho}{\partial t} .$$

Letting Δx approach zero,

$$\rho_o \frac{\partial v}{\partial x} + \frac{\partial \rho}{\partial t} = 0 . \quad (3-5)$$

Equation of state. Liquids behave approximately according to

$$\frac{d\rho}{d\rho} = \frac{\rho_o}{\beta} \quad (3-6)$$

where

β = bulk modulus of the fluid.

Equations (3-4), (3-5), and (3-6) above can be combined to yield

$$\frac{\rho_o}{\beta} \frac{\partial^2 p}{\partial t^2} + \frac{8\mu_o}{\beta r_o^2} \frac{\partial p}{\partial t} = \frac{\partial^2 p}{\partial x^2} . \quad (3-7)$$

Equation (3-7) is a simple form of the fluid-transmission-line equation. It applies for laminar flow through a rigid tube. Depending on the diameter, wall thickness, and distensibility of the tube, and on the compressibility of the fluid, the rigid-tube assumption may not be accurate. In such a case it is helpful to consider the sensing line as an analog of an electrical transmission line [13] in order to derive a more complete fluid-transmission-line equation.

Electrical-Transmission-Line Analogy

The following analogy is made between dependant variables:

<u>electrical</u>		<u>fluid</u>
potential difference	↔	pressure difference
current	↔	mass flow rate.

We now define the parameters of the transmission-line equation to account for the effects of interest.

Resistance. For a given length, Δx , of line the resistance is defined to be

$$\begin{aligned}\tilde{R} &\equiv \frac{\text{steady-state pressure drop}}{\text{steady-state mass flow rate}} \\ &= \frac{\Delta p}{Q \rho_o} .\end{aligned}$$

The Poiseuille equation (Equation 3-2) leads to

$$\tilde{R} = \frac{8 \mu_o}{\pi r_o^4 \rho_o} \Delta x .$$

The resistance per unit length is

$$\begin{aligned}
 R &\equiv \frac{\tilde{R}}{\Delta x} \\
 &= \frac{8\mu_o}{\pi r_o^4 \rho_o} \\
 &= \frac{8\nu_o}{\pi r_o^4} \quad (3-8)
 \end{aligned}$$

where

ν_o = kinematic viscosity (μ_o/ρ_o).

Inductance (inertance). The fluid-line analogy of inductance is called inertance since it is related to the inertia of the fluid. The inertance of a length, Δx , of line is defined by the instantaneous pressure difference required to cause a unit mass flow acceleration.

$$\begin{aligned}
 \tilde{L} &\equiv \frac{\text{instantaneous pressure difference}}{\text{rate of change of mass flow rate}} \\
 &= \frac{\Delta p}{\frac{d}{dt}(Q\rho_o)} \quad (3-9)
 \end{aligned}$$

The rate of change of mass flow rate is an acceleration of the fluid.

The acceleration is related to the pressure difference by

$$A\Delta p = (\rho_o A \Delta x) \frac{dv}{dt} \quad (3-10)$$

Since

$$\frac{dQ}{dt} = A \frac{dv}{dt} ,$$

Equations (3-9) and (3-10) can be combined to yield

$$\begin{aligned}\tilde{L} &= \frac{\Delta p}{\frac{dQ}{dt} \rho_o} \\ &= \frac{\Delta p}{\left(\frac{A \Delta p}{\rho_o \Delta x}\right) \rho_o} \\ &= \frac{\Delta x}{A} \quad .\end{aligned}$$

The inertance per unit length is

$$\begin{aligned}L &= \frac{\tilde{L}}{\Delta x} \\ &= \frac{1}{A} \\ &= \frac{1}{\pi r_o^2} \quad .\end{aligned}$$

Capacitance (compliance). The fluid analogy of capacitance is herein called compliance. It is related to distension of the tube walls and compression of the fluid. The compliance of a section of line is defined to be the change in mass of fluid in a section of unit length resulting from a unit change in static pressure throughout the section.

$$\tilde{C} \equiv \frac{\text{change in fluid mass in the section}}{\text{change in static pressure}}$$

Two components of the total compliance $\tilde{C}$ will be accounted for:

1. the compliance due to the compressibility of the fluid, $\tilde{C}_f$,
2. that due to distensibility of the tube, $\tilde{C}_t$.

The equation of state provides the relation for the first component

$$\frac{d\rho}{dp} = \frac{\rho_o}{\beta}$$

$$(A\Delta x) \frac{d\rho}{dp} = A\Delta x \frac{\rho_o}{\beta}$$

$$\tilde{C}_f = \frac{\pi r_o^2 \rho_o}{\beta} \Delta x \quad .$$

The compliance due to tube distension can be estimated by treating the tube as a thin-walled pressure vessel. In this case, the change in radius, Δr , due to a change in pressure Δp is given by [15]

$$\Delta r = \frac{\Delta p r_o^2}{hY}$$

where

h = tube wall thickness,

Y = Young's modulus for the tube wall material.

Thus, the tube-distension component is

$$\begin{aligned} \tilde{C}_t &= \frac{\pi [(r_o + \Delta r)^2 - r_o^2] \Delta x \rho_o}{\Delta p} \\ &\cong \frac{2\pi r_o^3 \rho_o}{hY} \Delta x \quad . \end{aligned}$$

The compliance per unit length in each case denoted by C_f and C_t , while the total compliance per unit length is written without the subscript.

We can now take the electrical-transmission-line equation [14]

$$\frac{\partial^2 v}{\partial x^2} = RGv + (RC + GL) \frac{\partial v}{\partial t} + LC \frac{\partial^2 v}{\partial t^2}$$

and substitute pressure, p , for voltage, v , and let $G = 0$ (no shunt conductance, i.e., no fluid leakage) to obtain the fluid-transmission-line equation

$$\frac{\partial^2 p}{\partial x^2} = RC \frac{\partial p}{\partial t} + LC \frac{\partial^2 p}{\partial t^2} \quad (3-11)$$

in which the parameters are defined in Table 3-1. If we ignore the tube-distension component of compliance, Equation (3-11) is identical to Equation (3-7), which was derived without using the electrical analogy. This gives confidence that the electrical analogy is meaningful and that the parameters in Table 3-1 have been defined properly. Table 3-1 summarizes the derived parameters.

Table 3-1. Sensing Line Parameters

Parameter	Symbol	Formula	Dimensions
Resistance per unit length	R	$\frac{8\mu_0}{\pi r_o^4 \rho_0}$	$\text{length}^{-2} \text{ time}^{-1}$
Inertance per unit length	L	$\frac{1}{\pi r_o^2}$	length^{-2}
Compliance per unit length due to:			
1. fluid compression	C_f	$\frac{\pi r_o^2 \rho_0}{\beta}$	time^2
2. tube distension	C_t	$\frac{2\pi r_o^3 \rho_0}{hY}$	time^2

Limitations of the Model

Equations of greater accuracy than Equation (3-11) have been derived [10], but they lack the valuable constant-coefficient property which will be required in subsequent manipulations. All three parameters are, in fact, frequency dependent due to heat transfer and nonparabolic radial velocity profiles. Heat transfer need not be considered when a liquid is used [9]. Nonparabolic radial velocity profiles occur at high frequency [16]. This is due to the fact that the flow near the walls is dominated by viscous effects while inertial effects are small. Inertial effects are more important in the central region. The region near the wall is thus able to respond more rapidly to changes in pressure gradient. This effect leads to much greater attenuation of oscillatory pressures at high frequencies [10]. This effect can even be important at frequencies below the quarter-wave resonance of the line. However, this effect is not present when the axial pressure gradient is not oscillating. As will be shown in the next section, a time-independent pressure gradient is necessary for the concept of asymptotic ramp delay time of a sensing line to have meaning. Thus, the model is limited to describing low-and zero-frequency (e.g., unterminated ramp) pressure oscillations in a uniform liquid-filled tube.

A summary of assumptions made in deriving the model is given below:

1. no radial flow,
2. frictional effects described by Poiseuille equation (laminar flow),
3. equation of state for liquids,
4. tube considered to be a massless thin-walled cylinder, and
5. no heat transfer.

Asymptotic Ramp Delay Time of a Sensing Line

It is the purpose of this investigation to develop a method of measuring the asymptotic ramp delay time, τ , (as defined in Chapter 1) of a sensing line. Previous developments in this chapter will be used to derive an approximate analytical expression for τ for comparison with experimental measurements. Two separate derivations will be provided: the first because of its simplicity, and the second because it gives a more general form which will be required later.

First derivation. Consider a sensing line of length ℓ with an open end (origin) and a closed end (terminus). Assume that the pressure at the origin is increasing linearly with time, i.e.,

$$p(0,t) = at$$

where

a = pressure ramp rate;

$p(x,t)$ = pressure at position x at time t .

Also assume that the pressure has been increasing in this fashion for a sufficient period of time so that the pressure at every point in the system is rising at exactly the same rate, but displaced in time, i.e.,

$$p(x,t) = a[t - \tau(x)] \quad (3-12)$$

where

$\tau(x)$ = asymptotic ramp delay time at location x .

Equation (3-12) implies that

$$\frac{\partial p}{\partial t} = a \quad ; \quad (3-13)$$

$$\frac{\partial^2 p}{\partial t^2} = 0 \quad ; \quad (3-14)$$

$$\frac{\partial^2 p}{\partial x^2} = -a \frac{\partial^2 \tau}{\partial x^2} \quad . \quad (3-15)$$

If (3-13), (3-14), and (3-15) are substituted into the pressure-transmission line equation (3-11), the result is

$$\frac{\partial^2 \tau}{\partial t^2} = -RC \quad . \quad (3-16)$$

We may solve (3-16) for $\tau(x)$ if appropriate boundary conditions are known. One possible set is

$$\tau(0) = 0 \quad ; \quad (3-17)$$

$$\left. \frac{\partial \tau}{\partial x} \right|_{x=\ell} = 0 \quad . \quad (3-18)$$

The first of these is obvious, the second follows from

$$\left. \frac{\partial p}{\partial x} \right|_{x=\ell} = 0$$

which must be the case for a sensing line terminated by a closed end.

Integrating (3-16) and using (3-17) and (3-18), the result is

$$\tau(x) = RC \left(\ell x - \frac{x^2}{2} \right) \quad , \quad (3-19)$$

and at the terminus,

$$\tau(\ell) = RC \frac{\ell^2}{2} \quad . \quad (3-20)$$

Second derivation. Consider an infinitesimal section of line of length Δx , at a distance x away from the lines origin (Figure 3-1, page 15). Again assume that

$$p(x,t) = a[t - \tau(x)] \quad .$$

The mass flow rate through the section is

$$\dot{M} = \frac{1}{R(x)\Delta x} [p(x,t) - p(x + \Delta x,t)] \quad (3-21)$$

where

$R(x)$ = resistance per unit length of line at position x .

Notice that inertia effects are not present because there is no acceleration through the section. There is no acceleration because, by assumption, the pressure is rising at precisely the same constant rate at every point in the line.

The flow through the section must have somewhere to go. Space is provided by fluid compression and tube distension downstream of the section. Thus, the flow through the section is also given by

$$\begin{aligned} \dot{M} &= \int_x^{\ell} [C(x) \frac{dp}{dt}] dx \\ &= a \int_x^{\ell} C(x) dx \end{aligned} \quad (3-22)$$

where

$C(x)$ = compliance per unit length of line at position x .

Equating (3-21) and (3-22) yields

$$\frac{1}{R(x)\Delta x} [p(x,t) - p(x + \Delta x,t)] = a \int_x^{\ell} C(x) dx \quad ,$$

and inserting (3-12) we get

$$\frac{a[t - \tau(x)] - a[(t - \tau(x + \Delta x))]}{a\Delta x} = R(x) \int_x^l C(x) dx$$

$$\frac{\tau(x + \Delta x) - \tau(x)}{\Delta x} = R(x) \int_x^l C(x) dx .$$

Now let Δx approach zero,

$$\lim_{\Delta x \rightarrow 0} \frac{\tau(x + \Delta x) - \tau(x)}{\Delta x} = R(x) \int_x^l C(x) dx$$

$$\frac{d\tau}{dx} = R(x) \int_x^l C(x) dx .$$

Integrating and letting $\tau(0) = 0$,

$$\tau(x) = \int_0^x [R(x') \int_{x'}^l C(x'') dx''] dx \quad (3-23)$$

In order to integrate (3-23) the functions $R(x)$ and $C(x)$ must be known continuously throughout the line. However, for a uniform line, the integration yields

$$\tau(x) = RC(lx - \frac{x^2}{2}) . \quad (3-24)$$

and at the terminus,

$$\tau(l) = RC \frac{l^2}{2} . \quad (3-25)$$

Note that (3-24) and (3-25) are identical to (3-16) and (3-17) respectively. Equation (3-23) will be used later.

Predicted Effect of Blockage on Ramp Delay Time

Consider a sensing line with constant compliance per unit length C , but with resistance per unit length, R , varying as depicted in Figure 3-2. Assume that the blockage is a uniform constriction with inside radius r_b in a cylindrical sensing line of inside radius r_n . Using Equation (3-23) an expression can be developed which relates the ramp delay time of the blocked line, τ_b , to that of the line in its normal unblocked condition, τ_n (see Appendix). The expression is, in terms of the nominal cross-sectional flow area A_n of the sensing line and the flow area of the blocked section, A_b ,

$$\frac{\tau_b}{\tau_n} = 1 + \frac{\Delta x}{\ell} \left[2 \left(1 - \frac{x_1}{\ell} \right) - \frac{\Delta x}{\ell} \right] \left[\left(\frac{A_n}{A_b} \right)^2 - 1 \right] \quad (3-26)$$

This equation is plotted in Figure 3-3 for values of the dimensionless ratios $(\Delta x/\ell)$ and (A_n/A_b) which correspond to an inch-long constriction in a 100-foot-long sensing line. The equation is plotted for various blockage locations as described in the figure.

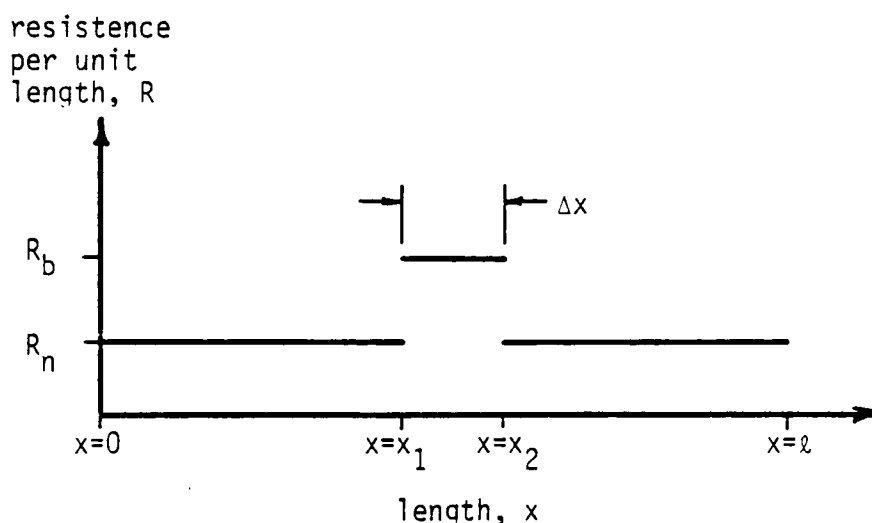


Figure 3-2. Variation of Resistance versus Length for an Idealized Sensing Line with Localized Blockage.

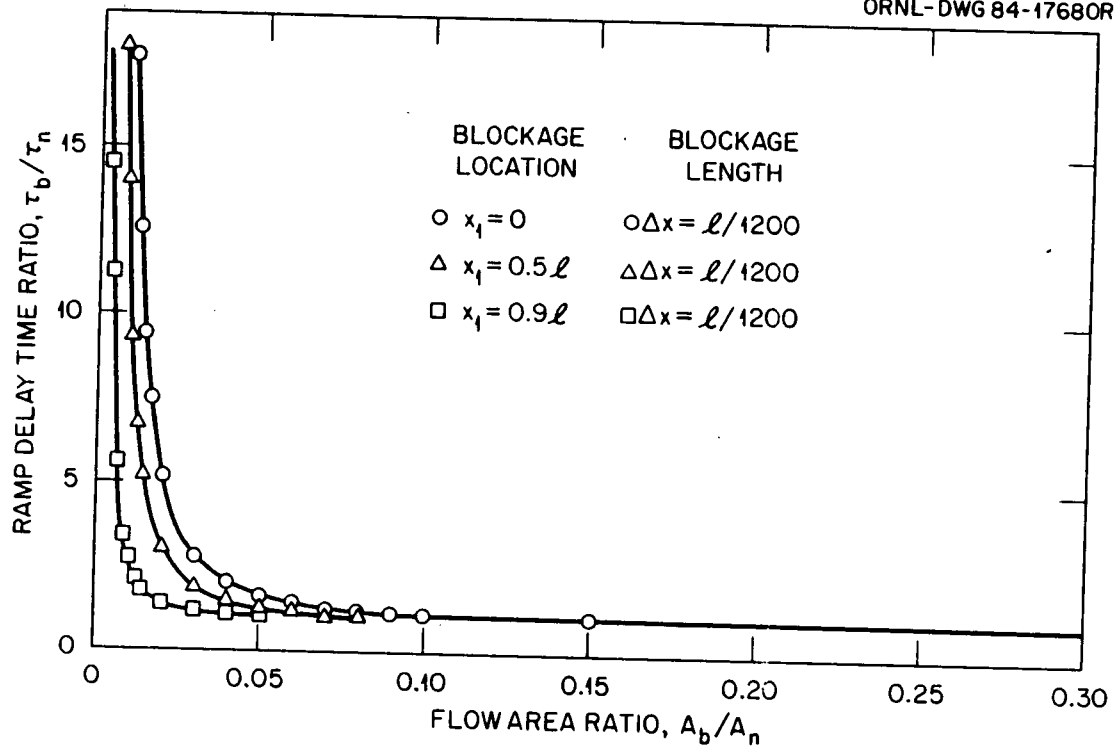


Figure 3-3. Plot of Equation (3-26).

Equation (3-26) is based on fully-developed laminar flow and therefore ignores entry effects into the blocked region. The intent of the figure is to demonstrate that a sensing line blockage must be nearly complete before any significant dynamic effect is observed. It also reveals the importance of the location of the blockage.

CHAPTER 4

THE BURP METHOD

Description of the Method

The burp test is a proposed method for in-situ dynamic testing of liquid filled pressure sensing lines. It involves connecting a small air-filled chamber at atmospheric pressure (the burp chamber) to a test port near the pressure sensor, as shown in Figure 4-1. Flow from the pressurized sensing line into the burp chamber is inhibited by a fast-acting solenoid valve, which is initially shut. A test is initiated by opening the solenoid valve. As pressure equilibrium is established between the process and the burp chamber via flow through the sensing line, a reference pressure transducer records the burp chamber pressure. Since the sensing line is the vehicle for the establishment of pressure equilibrium, the recorded pressure transient may be used to infer the dynamic characteristics of the sensing line. The purpose of the test is to measure the asymptotic ramp delay time, τ , of the sensing line.

The burp-test concept has been investigated experimentally in a series of experiments conducted at The University of Tennessee and at the Oak Ridge National Laboratory. These experiments will be described in a later section. First the theoretical basis of the burp test will be presented.

Theoretical Basis

It was shown in a previous section that the asymptotic ramp delay time, τ , of a sensing line depends on the magnitude and distribution of resistance and compliance throughout the sensing line (Equation 3-23).

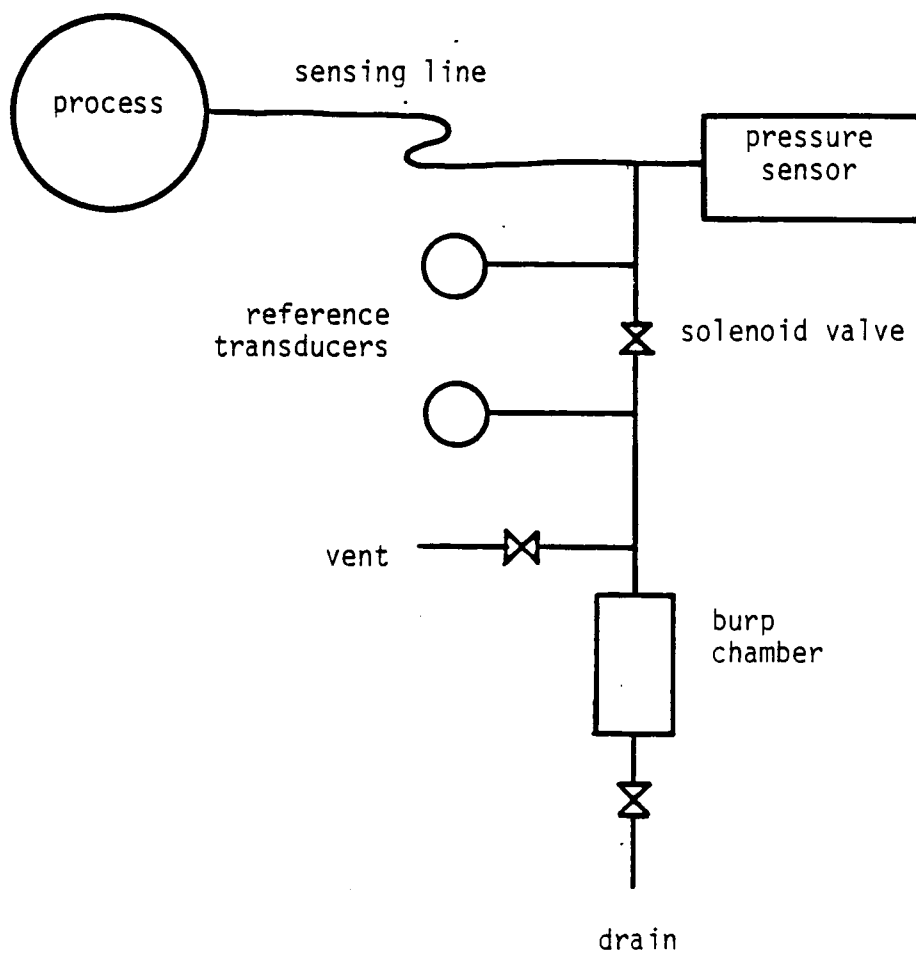


Figure 4-1. Simple Burp-Test Schematic.

Yet the burp test is expected to yield an estimate of τ based on pressure measurements at one point and a model of the sensing line. It is apparent that a simplified model is required. Thus, the fluid-transmission-line equation (which is strictly valid only for a uniform line) was employed. The burp test, coupled with the uniform-line model, is to be used to determine the effect of blockages (which are non-uniformities) on the dynamic behavior of the sensing line.

Consider the fluid-transmission-line equation

$$\frac{\partial^2 p}{\partial x^2} = RC \frac{\partial p}{\partial t} + LC \frac{\partial^2 p}{\partial t^2} . \quad (4-1)$$

Assume, in accordance with the method of separation of variables [17], that the solution to (4-1) under some set of initial and boundary conditions obeys

$$p(x,t) = f(t) \phi(x) . \quad (4-2)$$

Equation (4-2) is substituted into (4-1) with the result that

$$f(t) \frac{d^2 \phi}{dx^2} = RC \phi(x) \frac{df}{dt} + LC \phi(x) \frac{d^2 f}{dt^2} ,$$

or

$$\frac{1}{\phi(x)} \frac{d^2 \phi}{dx^2} = \frac{1}{f(t)} (RC \frac{df}{dt} + LC \frac{d^2 f}{dt^2}) . \quad (4-3)$$

Equating both sides of Equation (4-3) to the separation constant K^2 gives

$$\frac{d^2 \phi}{dx^2} + K^2 \phi(x) = 0 \quad (4-4)$$

and

$$LC \frac{d^2 f}{dt^2} + RC \frac{df}{dt} + K^2 f(t) = 0 \quad . \quad (4-5)$$

The general solution of (4-4) is

$$\phi(x) = A \cos Kx + B \sin Kx \quad . \quad (4-6)$$

A possible set of boundary conditions for Equation (4-6) is

$$\phi(0) = 0 \quad (4-7)$$

and

$$\left. \frac{d\phi}{dx} \right|_{x=\ell} = 0 \quad . \quad (4-8)$$

The first of these, Equation (4-7), results from the assumption that the sensing line originates in an infinite medium at constant pressure. The second boundary condition is known as a reflected boundary condition and strictly applies to a completely closed end which admits no mass flow. In fact the situation is more complicated than that since the pressure sensor and burp chamber both are compliant and admit mass flow. The burp-chamber compliance is, by design, small compared to that of the sensing line. In many cases the compliance of the pressure sensor is likewise small. For now, therefore, we ignore these end effects and apply the boundary conditions (4-7) and (4-8) to Equation (4-6).

This yields an expression for the separation constant

$$K = \frac{n\pi}{2\ell} \quad (4-9)$$

where

$$n = 1, 3, 5, \dots$$

Particular values of n correspond to different modes of oscillation of the sensing line. It is easy to identify the value of n since the wavelength of the oscillation is given by

$$\lambda = \frac{c}{f}$$

where

c = speed of sound in the fluid,

f = observed frequency.

Since $\lambda = 2\pi/K$, the value of n is approximately given by

$$n = \frac{4\ell}{\lambda}$$

Experimental observations (to be described) have indicated that the best value to use for n is unity. This corresponds to the quarter-wave resonance of the line.

Taking $n = 1$ in Equation (4-9) yields

$$\left(\frac{2\ell}{\pi}\right)^2 LC \frac{d^2f}{dt^2} + \left(\frac{2\ell}{\pi}\right)^2 RC \frac{df}{dt} + f(t) = 0 \quad . \quad (4-10)$$

The parameters R , L , and C in Equation (4-10) are the same as those in the fluid transmission line equation. If the product RC can be identified from the application of Eq. (4-10) to a burp-test transient, the ramp delay time of the line can be estimated by [see Equation (3-25)]

$$\tau = RC \frac{\ell^2}{2} \quad (4-11)$$

which applies for a uniform sensing line. Provided that oscillatory behavior is observed, Equation (4-10) can be written in terms of the roots of the characteristic equation

$$z = \alpha + \omega j$$

$$z^* = \alpha - \omega j$$

as

$$\frac{1}{zz^*} \frac{d^2 f}{dt^2} + \frac{z^* + z}{zz^*} \frac{df}{dt} + f(t) = 0$$

or

$$\frac{1}{\alpha^2 + \omega^2} \frac{d^2 f}{dt^2} + \frac{2\alpha}{\alpha^2 + \omega^2} \frac{df}{dt} + f(t) = 0 \quad . \quad (4-12)$$

By comparing Equations (4-10) and (4-12), we have

$$\left(\frac{2\ell}{\pi}\right)^2 RC = \frac{2\alpha}{\alpha^2 + \omega^2} \quad (4-13)$$

Solving Equation (4-13) for

$$\tau = RC \frac{\ell^2}{2} \quad ,$$

we have

$$\tau = \left(\frac{\pi^2}{8}\right) \left(\frac{2\alpha}{\alpha^2 + \omega^2}\right) \quad . \quad (4-14)$$

Using Equation (4-10), a simplified representation of the sensing line can be developed from which Equation (4-14) can be deduced. Equation (4-10) is equivalent to a lumped network with parameters

$$\hat{R} = R\ell \quad (4-15a)$$

$$\hat{C} = \frac{4}{\pi^2} C \ell \quad (4-15b)$$

$$\hat{L} = L \ell \quad (4-15c)$$

as shown in Figure 4-2. This model will be called the quarter-wave lumped model. It approximately represents the unforced behavior of the sensing line at frequencies near the quarter-wave resonance of the line.

The choice of parameters for the network in Figure 4-2 which yields the behavior represented by Equation (4-10) is not unique. Any choice which preserves the values of the products $(2\ell/\pi)^2 RC$ and $(2\ell/\pi)^2 LC$ will do. The choice in Equation (4-15) is made so that a load compliance can be included as shown in Figure 4-3. This load compliance may represent the burp-chamber, the process sensor, or both. The advantage to this particular choice is that, as the load compliance increases and the

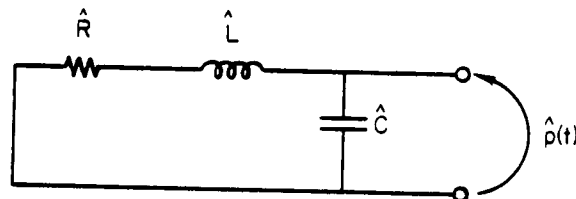


Figure 4-2. Quarter-Wave Lumped Model.

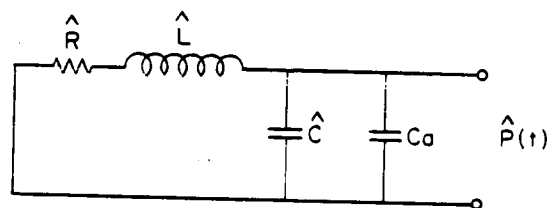


Figure 4-3. Quarter-Wave Lumped Model with Load Compliance C_a .

sensing-line compliance becomes insignificant, the proper relationship emerges between the inertance and resistance of the line and the load compliance.

The natural frequency of the network in Figure 4-3 is given by

$$\begin{aligned}
 \omega_n &= \frac{1}{\sqrt{L(\tilde{C} + \tilde{C}_a)}} \\
 &= \frac{1}{\sqrt{L\ell(C\ell\frac{4}{\pi^2} + \tilde{C}_a)}} \\
 &= \frac{\pi}{2\ell\sqrt{LC}} \frac{1}{\sqrt{1 + (\frac{\pi^2}{4}) \frac{\tilde{C}_a}{C\ell}}} \\
 &= \omega_{1/4} \frac{1}{\sqrt{1 + (\frac{\pi^2}{4}) \frac{\tilde{C}_a}{C\ell}}}
 \end{aligned}$$

where $\omega_{1/4}$ is the quarter-wave frequency (rad/sec) of the sensing line. Thus the ratio of the natural frequency of the system to the quarter-wave resonant frequency of the sensing line is given by

$$\frac{\omega_n}{\omega_{1/4}} = \frac{1}{\sqrt{1 + \frac{\pi^2}{4} \frac{\tilde{C}_a}{C\ell}}} \quad (4-16)$$

Equation (4-16) was derived from the intuitive conception of Figure 4-3. A mathematically rigorous treatment would include the load compliance as a boundary condition for the fluid-transmission-line equation. However, this would be difficult and will not be explored. It will be shown in a later section that Equation (4-16) agrees well with experiment.

Critical condition for the success of the method. Figure 4-4 depicts the relationship between ω_n and the parameters ω and α . As shown in Equation (4-16), ω_n is independent of $\hat{R}$, the resistance term of the lumped model. However, α is related to $\hat{R}$ by

$$\alpha = \frac{\hat{R}}{2L} . \quad (4-17)$$

Thus, increases in $\hat{R}$ are accompanied by increases in α and decreases in ω until the limiting condition

$$\alpha = \omega_n \quad (4-18)$$

is reached. At this point, the system is no longer underdamped and no pressure oscillations can be brought about by a burp test. This limiting condition can be translated into a statement of the maximum ramp delay time measureable by the burp-test method:

$$\tau_{\max} = \left[\left(\frac{\pi^2}{8} \right) \left(\frac{2\alpha}{\alpha^2 + \omega^2} \right) \right] \begin{matrix} \alpha = \omega_n \\ \omega = 0 \end{matrix}$$

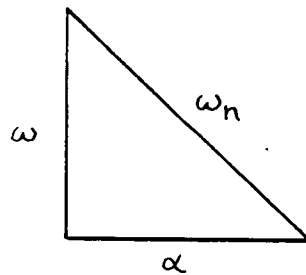


Figure 4-4. Depiction of the Natural Frequency.

$$\begin{aligned}\tau_{\max} &= \left(\frac{\pi^2}{8}\right) \frac{2}{\omega_n} \\ &= \frac{\pi^2}{4\omega_n} .\end{aligned}\tag{4-19}$$

Under ideal conditions, the burp chamber volume approaches zero and the natural frequency approaches the quarter-wave frequency, as evidenced by Equation (4-16). The quarter-wave frequency is approximately (in the rigid line case) given by

$$\omega_{1/4} = \frac{\pi c_o}{2\ell}$$

where

c_o = speed of sound in the fluid,

ℓ = sensing line length.

Thus Equation (4-19) becomes

$$\tau_{\max} = \frac{\pi \ell}{2c_o} .\tag{4-20}$$

Equation (4-20), for example, yields $\tau_{\max} \cong 30$ ms for a 100-foot-long sensing line filled with water at room temperature.

Identification of α and ω . After the solenoid valve is opened, the burp chamber takes some time to fill. During this time, the pressure recorded by the reference transducer is approximately constant. Then, the chamber fills up, and a decaying pressure oscillation is observed. Time zero can be chosen so that the pressure during the decaying oscillation is approximately described by

$$f(t) = f_o e^{-\alpha t} \cos \omega t\tag{4-21}$$

where

$f(t)$ = pressure at time t

$f_0 = f(0)$

$z = \alpha + \omega j$

= one root of the characteristic equation from the single lump model.

The imaginary component, ω , is simply the angular frequency of oscillation. The real component, α , can be estimated by plotting the amplitude of the pressure oscillation versus time on semilog paper and drawing a straight line through the points. The slope of the line determines α . Finally, the product $\hat{R}\hat{C}$ is given by

$$\hat{R}\hat{C} = \frac{2\alpha}{\alpha^2 + \omega^2} \quad (4-22)$$

and Equation (4-14) is used to determine τ , the asymptotic ramp delay time of the sensing line.

Sources of error. Energy is dissipated as the fluid flows into and out of the burp chamber through the solenoid valve's constriction. This additional energy loss increases the rate of damping above that for the sensing line alone. This effect tends to increase the estimate of τ .

The presence of the burp chamber also changes the distribution of compliance in the system. The effect of this is apparent by examination of Equation (3-23). The integral

$$w(x) = \int_x^l C(x) dx \quad (4-23)$$

can be viewed as weighting factor. Then Equation (3-23) can be written as

$$\tau(x) = \int_0^x [R(x)w(x)]dx \quad .$$

The weighting factor is equal to the total downstream compliance. If an additional compliance is added to the downstream end, the dynamic effect of the sensing line resistances throughout the line is increased. This effect also tends to increase the estimate of τ .

We have seen that, ideally, the burp chamber compliance should be small in comparison to that of the sensing line. However, the presence of the burp chamber always tends to increase damping and to decrease the resonant frequency. Both of these effects tend to increase the estimate of τ derived from a burp test. This insures that the burp test always gives a conservative estimate of τ .

Experimental Facilities

Three experimental facilities were used in the investigation of the burp-test method. The facilities can be described according to pressure regime as (a) low-pressure (< 30 psig), (b) medium-pressure (< 1000 psig), and (c) high-pressure (< 2300 psig).

Low pressure (30 psig). The first facility to be built was the low-pressure facility. It was to be used to determine whether or not the burp test concept merited further investigation. The facility consisted of a 42 gallon tank (to act as a pressure reservoir), a 50 foot length of 3/8 inch I. D. copper tubing (as a sensing line), a regulating valve (to introduce line blockage), a solenoid valve (to initiate the tests) and interchangeable burp chambers of 50, 100, and 1000 cc volume. A schematic diagram of the low-pressure facility is provided in Figure 4-5.

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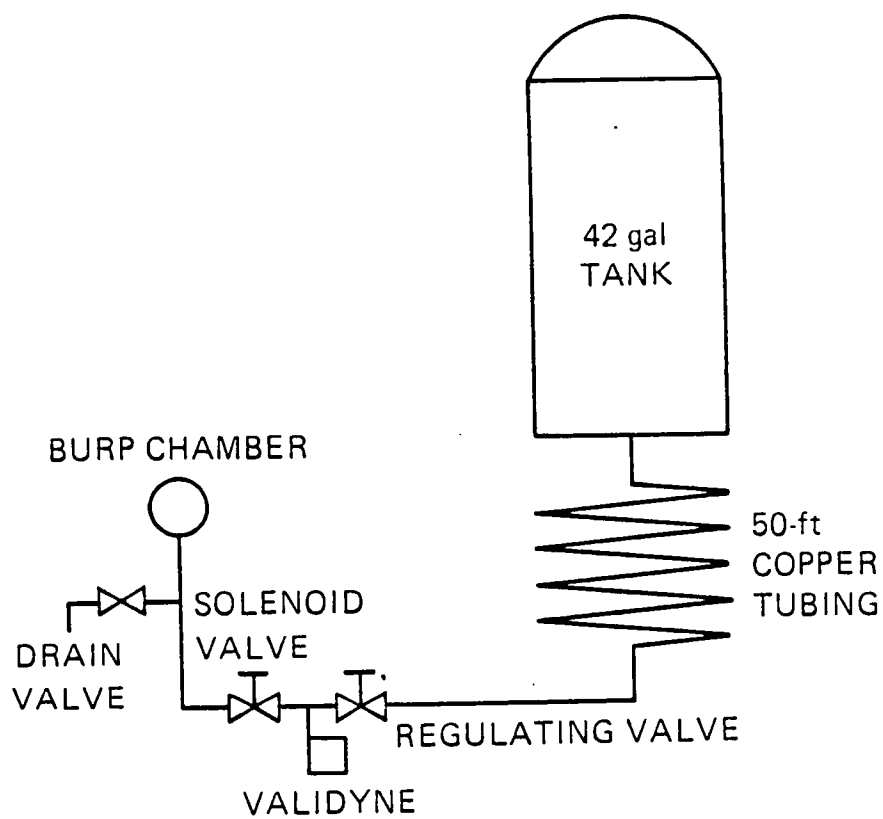


Figure 4-5. Schematic Diagram of Low-Pressure Facility.

Medium pressure (1000 psig). The medium-pressure facility was several steps beyond the low pressure facility. The pressure was increased by a factor of 20. This is only a factor of 2 less than typical reactor pressures. The major difference resulting from the increase in pressure was that the compliance of the burp chamber is greatly reduced in the higher pressure case. This means that the compliance of the sensing line itself became significant and the distributed nature of the sensing line came into play. For this reason the facility was designed so that the location of the blockage could be varied. The facility was also designed so that entrained air could be introduced in the sensing line.

A schematic diagram of the facility is provided in Figure 4-6. Table 4-1 describes the facility in detail.

High pressure (2300 psig). The high-pressure facility came the closest to simulating the conditions of a commercial power reactor. The Forced Convection Test Facility (FCTF) at ORNL was used as a pressure reservoir. The FCTF is a forced-circulation water loop which can be operated at temperatures and pressures up to 650°F and 2250 psig, with a flow of up to 50 gal/min. A schematic diagram of the high-pressure facility is provided in Figure 4-7. The sensing line is 100 ft. of stainless steel tubing which has an outside diameter of 3/8 in. and wall thickness of 0.065 in. The tubing used for the burp-test fixture has an outside diameter of 1/4 in and wall thickness of 0.065 in.

Burp-Test Instrumentation

The burp test requires one or more fast-responding pressure sensors so that the induced pressure transient may be recorded. The most important

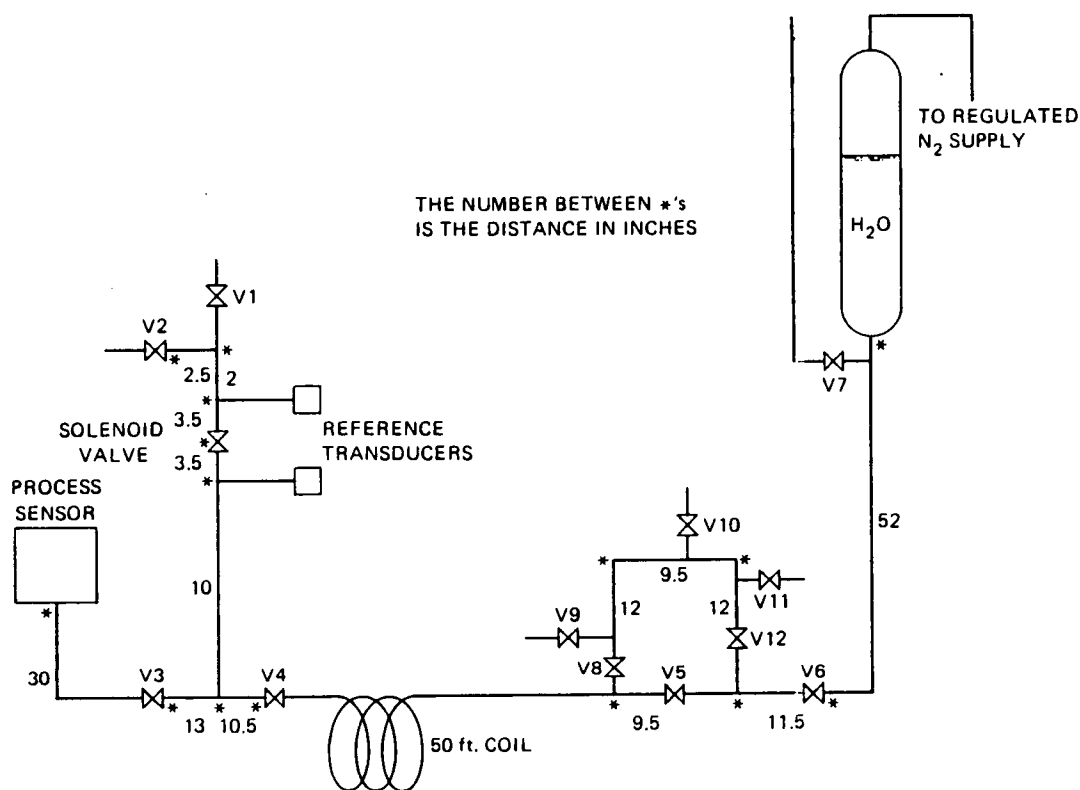


Figure 4-6. Schematic Diagram of Medium-Pressure Facility.

Table 4-1. Detailed Description of Medium Pressure Facility

Item	Function	Description
Tubing	Sensing line	1/4-in.-OD, 0.03-in. wall copper tubing
Solenoid valve	Test initiation	—
Burp chamber	See text	Section of 3/8-in.-OD copper tubing
Reference transducers	Data acquisition	Validyne, DP15TL
Pressure reservoir	Pressure reservoir	1 gal tank, pressure rating 1800 psi
Process sensor	Not used	—
Valve V4	Blockage introduction "near sensor"	Whitey (See Figure 4-12 for characteristics)
Valve V5	Blockage introduction "near process"	Whitey (See Figure 4-12 for characteristics)
Valves Vmn	Open and shut use only	Valves are of differing type
Level indicator	Indication of tank water level (when depressurized)	Plexiglass tube
To regulated He supply	Provides pressure source for reservoir	2000-psi He tank with regulator

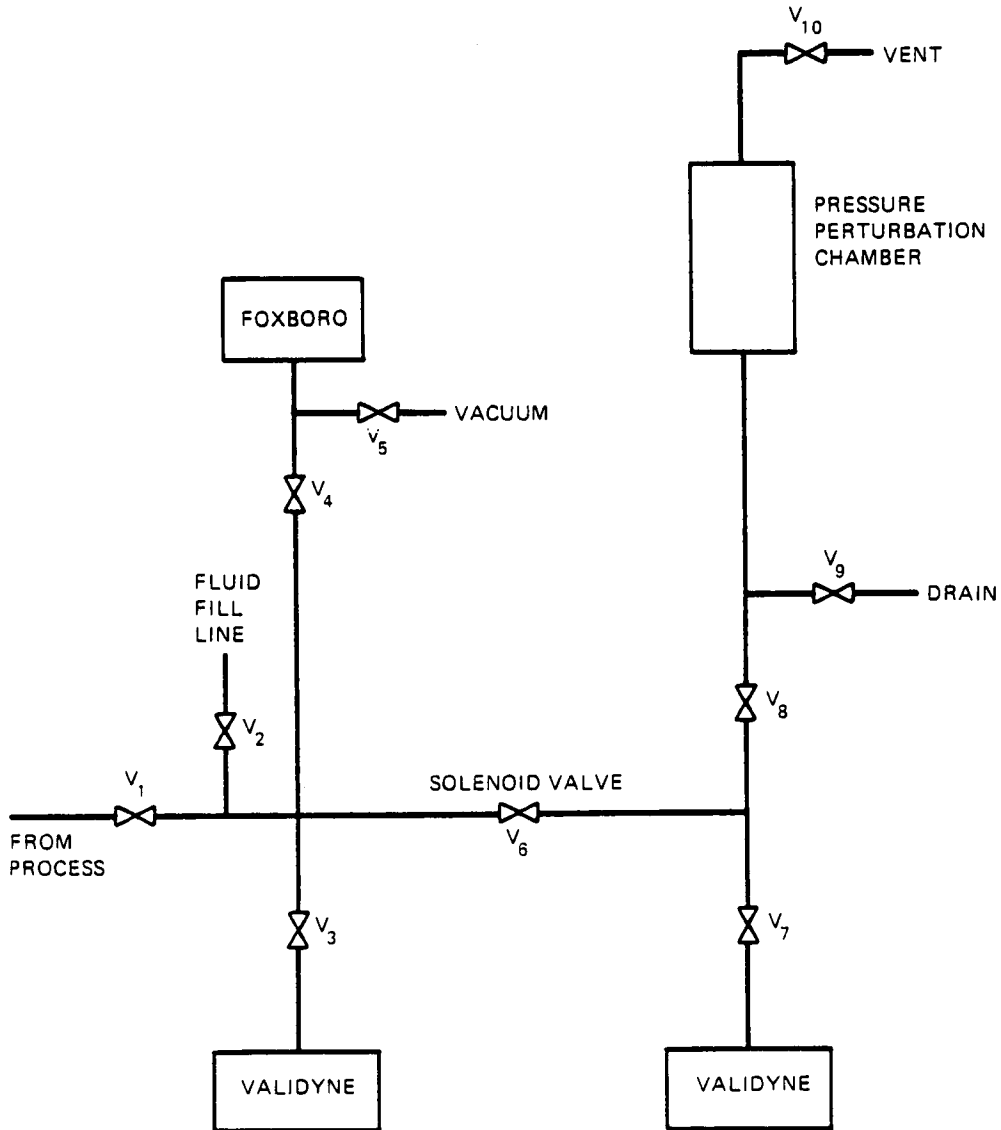


Figure 4-7. Schematic Diagram of High-Pressure Facility.

consideration in choosing this (these) sensor(s) is that the burp-test resonant frequency be well within the dynamic range of the pressure sensor(s). The process sensor, which would be an integral part of instrumentation channel to be tested, is not expected to have the required dynamic range. Therefore the fast-responding sensor(s) are part of the test equipment required. The fast-responding pressure sensor used in all three experimental facilities was a Validyne DPI5TL. This sensor is equipped with interchangeable diaphragms so that the very same sensor may be used for a wide range of pressure. A pair of these sensors was used in the medium- and high-pressure facilities so that the effect of the solenoid valve's constriction could be investigated. The presence of a process sensor was allowed for in the medium-pressure facility, but none was available for use. Therefore, the branch of the sensing line intended for a process sensor was kept valved shut. A nuclear-qualified Foxboro NE11GM was used in the high-pressure facility. This force-balance model is used in several nuclear plants throughout the country.

In all three facilities data was recorded using a Hewlett-Packard (HP) 5420 signal analyzer. This sophisticated device is capable of recording and analyzing electrical signals in many ways (e.g., auto-correlation, cross-correlation, power spectral density, transfer function). But for our applications it was mostly used as a glorified oscilloscope. The HP 5420 has two separate channels. Each channel was used to record a signal, in digital form. The records were then stored on magnetic tape as a routine part of the operation of the signal analyzer. Graphs of these records were conveniently made using an attached auxillary device. The recorded transients shown here were made in this fashion.

In addition to the HP 5420 signal analyzer, an Ampex tape recorder was used to record noise signals in the high pressure facility. This data was later sampled (converted to digital) and analyzed using computer programs developed at Oak Ridge National Laboratory's Instrumentation and Controls Division [18].

Burp Test Evaluation Results

Low pressure. The purpose of the low-pressure facility was to determine whether further investigation was merited. The burp test was shown, in low pressure tests, to be sensitive to changes in sensing line blockage. On the basis of that encouraging qualitative result, further experiments were planned.

Medium and high pressure. The development of the theory behind the burp test was greatly aided by results from the medium- and high-pressure facilities. The theory requires that the compliance of the burp chamber be small in comparison to that of the sensing line itself. The small-burp-chamber-compliance assumption is satisfied in the medium-and high-pressure facilities but not in the low-pressure facility. When designing a burp test facility, the presence of the burp chamber may be accounted for by augmenting the compliance in the single-lump model as was shown in Figure 4-3, page 37. The relationship between burp-chamber compliance (as a fraction of total sensing-line compliance) and resonant frequency is plotted in Figure 4-8. The theoretical relationship was determined according to the model in Figure 4-3, page 37. The measurements are from the medium- and high-pressure facilities. The wide range of frequencies was attained by using different burp-chamber sizes and static pressures

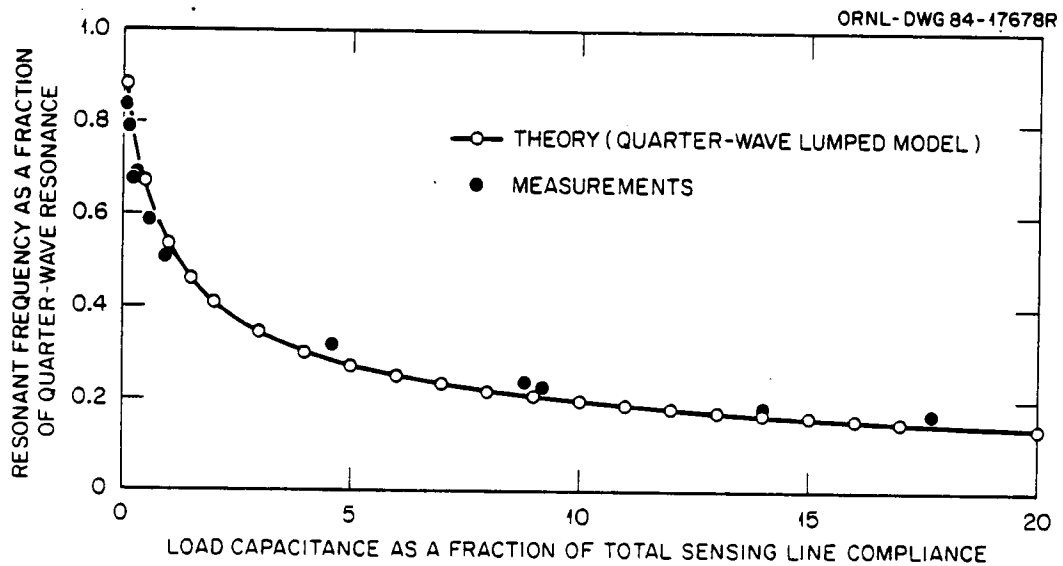


Figure 4-8. Comparison of Equation (4-16) with Experimental Measurements.

in the medium-pressure facility. The reason for the vast differences in burp-chamber compliances is that, for a given burp chamber, the compliance is inversely proportional to the square of the absolute pressure. The agreement between theory and experiment shown in Figure 4-8 lends confidence to the quarter-wave lumped model.

Since the compliance of the burp chamber is a function of pressure (being larger at lower pressure), there is a departure from linear behavior for large-amplitude pressure oscillations. However, since the burp-chamber compliance does not dominate the system (except for very large oscillations), this effect is usually not significant. Another mechanism, then, must be responsible for the failure of the oscillations to decay exponentially. Figure 4-9 shows semi-logarithmic plots of amplitude versus time for burp tests with different blockage conditions in the medium-pressure facility. In some cases the data can be fitted rather well with two straight lines. (In other cases, the magnitude of the pressure oscillations is so small that only a few data points can be accurately read.) Although the magnitude of the pressure oscillations is different in different cases, the two lines do not appear to intersect at about the same pressure, which indicates that the effect is not dependent on the amplitude of the pressure oscillations. This effect could be related to the presence of two distinct but nearly equal frequencies which interact in a decaying beating pattern. A typical test record from the medium-pressure facility, Figure 4-10, shows the beating pattern. The second frequency could be the quarter-wave frequency of the sensing line or could be due to a standing wave being set up between the reservoir and a reflection site, such as the blockage simulation valve.

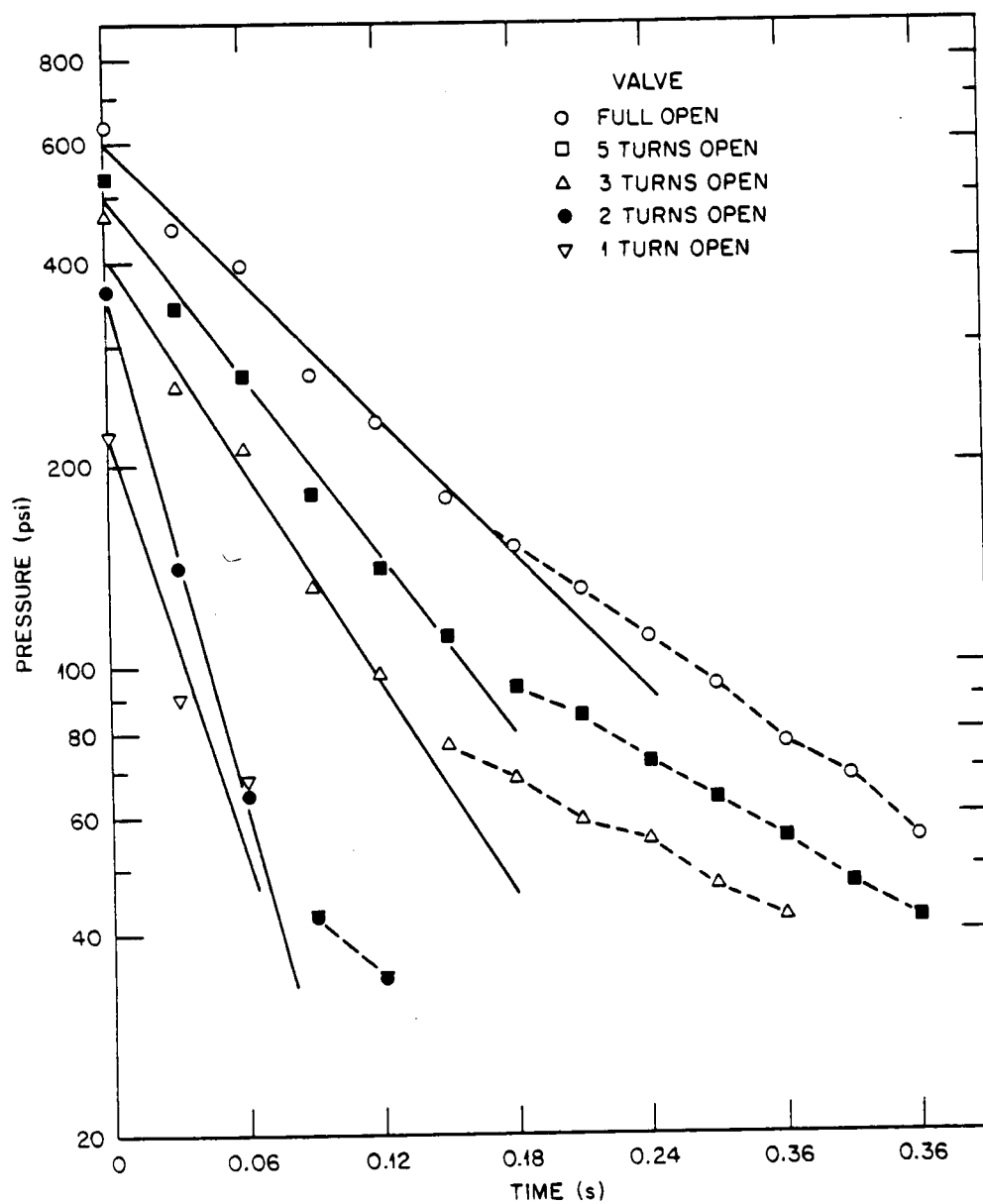


Figure 4-9. Amplitude Plots from the Medium-Pressure Facility.

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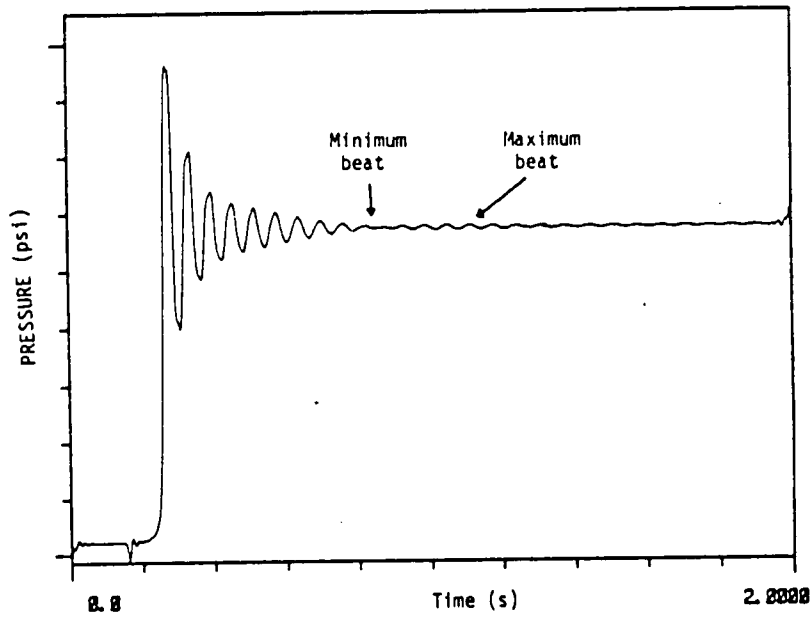


Figure 4-10. Burp-Test Record from the Medium-Pressure Facility with Medium Blockage.

Figure 4-11 shows a typical trace (from the medium-pressure facility) in which the blockage simulation valve near the process was open one turn out of nine. (The same valve was four turns open for the test in Figure 4-10.)

Since no single line fits the data well in some cases, there is an ambiguity in the determination of α . The steeper of the two lines yields the larger value. For the sake of conservatism, the steeper line should be used.

Figure 4-12 shows a characterization of the valves used for blockage simulation in the medium-pressure facility. The flow coefficient is defined to be the flow in gallons per minute which would result from a steady-state pressure difference of 1 psi across the valve.

The beat phenomenon was not observed in the high pressure facility. Typical traces (Figures 4-13 and 4-14) show the effect of ten and one turns open respectively of the blockage valve near the process. Note that the blockage is so severe in the one-turn case that there is no oscillation. In this case, there can be no determination of τ by the method described here. However, the absence of oscillation indicates that there is a severe line blockage.

Typical plots (from the high-pressure facility) for determining α are shown in Figure 4-15. There is considerably more scatter in the amplitude versus time data from the high-pressure facility as compared to the data from the medium-pressure facility (see Figure 4-15). The cause of the scatter is apparently that the data sampling frequency was too low to accurately record the pressure transient. Some of the curves in Figure 4-15 exhibit the same sort of nonlinearity observed in the

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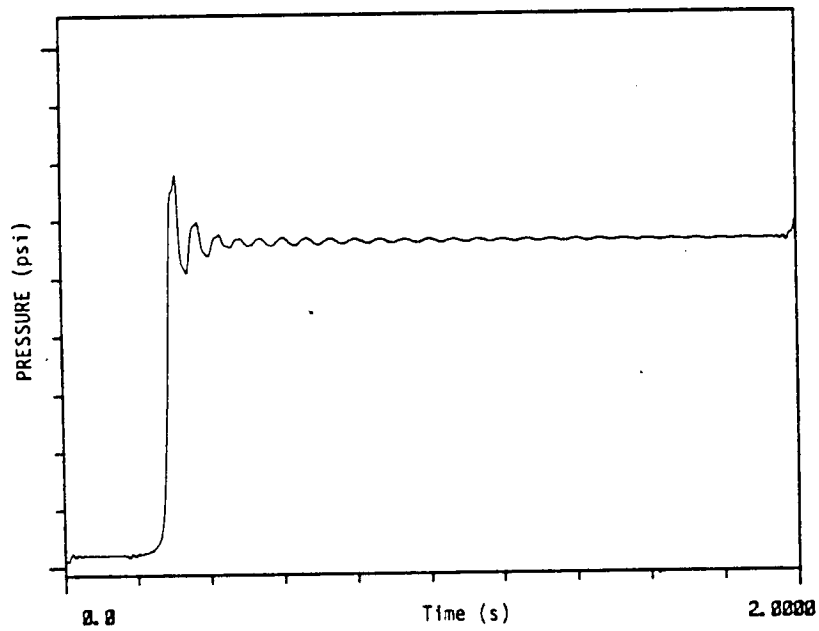


Figure 4-11. Burp-Test Record from the Medium-Pressure Facility with High Blockage.

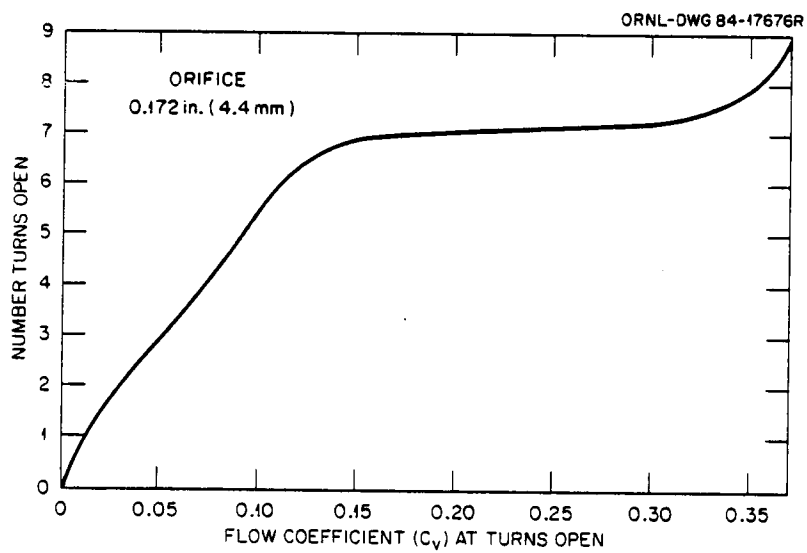


Figure 4-12. Valve Flow-Coefficient Characterization for the Medium Pressure Facility.

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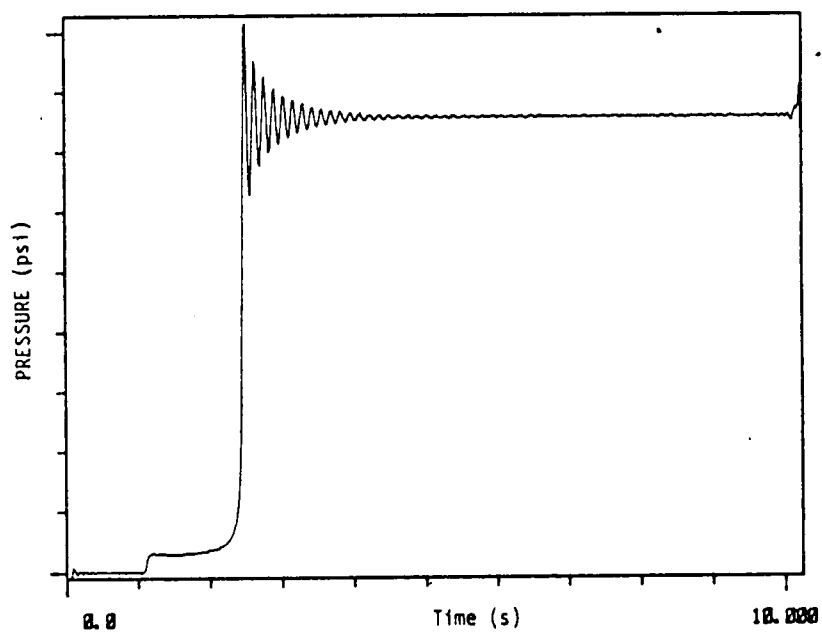


Figure 4-13. Burp-Test Record from the High-Pressure Facility with Low Blockage.

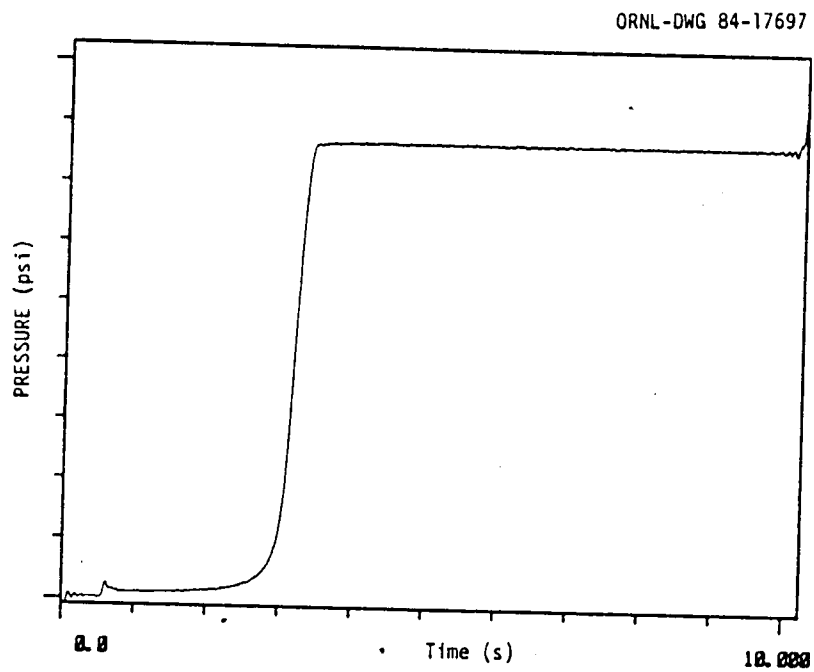


Figure 4-14. Burp-Test Record from the High-Pressure Facility with High Blockage.

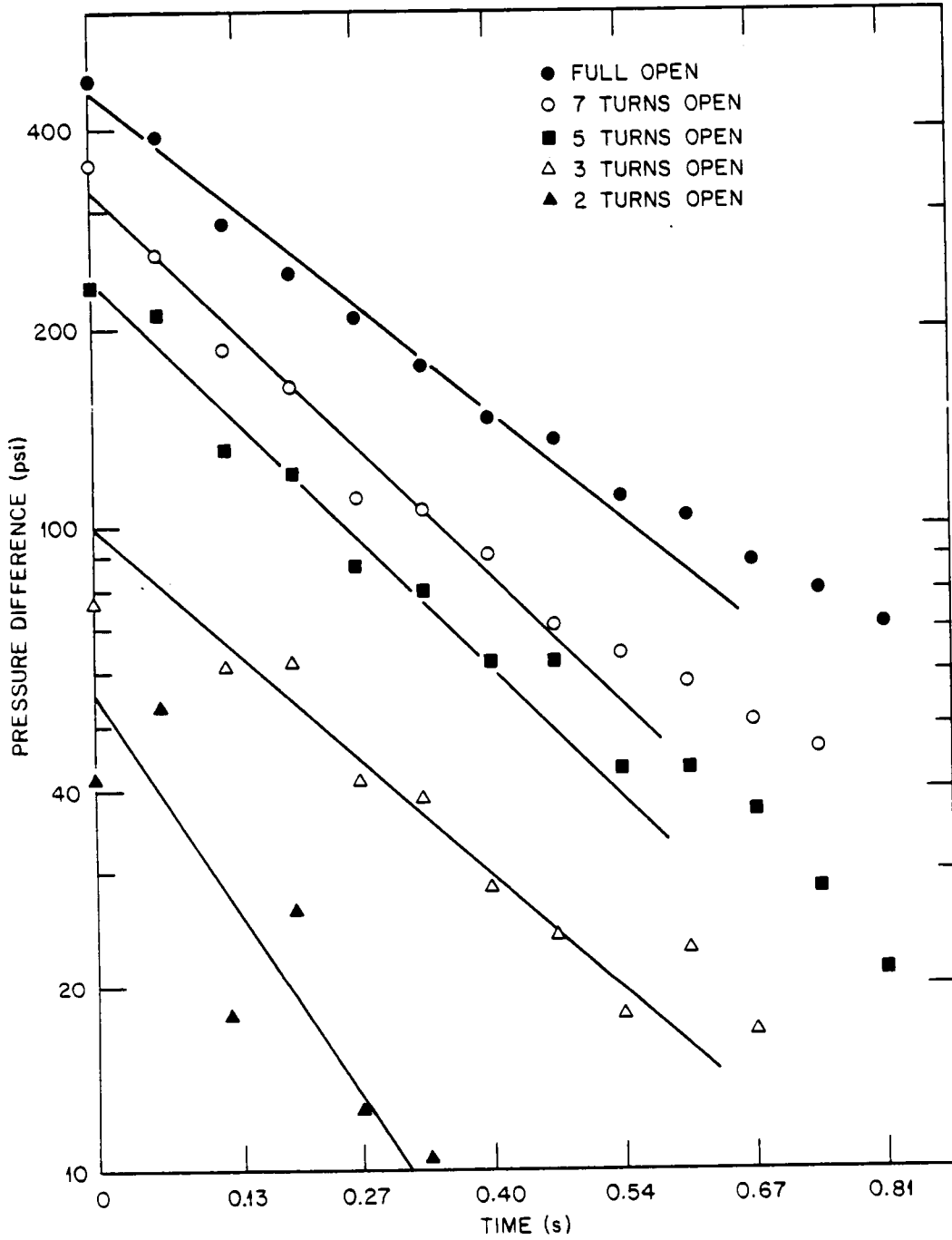


Figure 4-15. Amplitude Plots from the High-Pressure Facility.

low-pressure facility. In some highly blocked cases the amplitude of a "trough" exceeds that of the preceding "peak." This could be due to the fact that the burp chamber takes longer to fill under high-blockage conditions and therefore that the oscillations are superimposed on a general upward trend. In these cases (notably the two-turns-open case in Figure 4-15), it is still possible to draw a line which crudely fits the data, and thereby estimate τ .

The blockage simulation valve near the process had a much greater dynamic effect than the one near the sensor in both the high-pressure and the medium-pressure facilities. This is readily explained by noting that the dynamic effect of resistance is dependent on the total downstream compliance [see Equation (3-23)]. No results are presented for blockage near the sensor.

Figure 4-16 is a plot of asymptotic ramp delay time, τ , (as estimated from burp tests) versus valve flow coefficient for the medium-pressure facility.

Using Equation (3-25) and formulas for R and C (Table 3-1, page 22), theoretical predictions of τ can be made for an unblocked sensing line. The formulas take no account of bends, valves or other irregularities. Comparisons of the theoretical predictions of τ with those obtained from burp tests are provided in Table 4-2.

It is not surprising that the theoretical predictions of τ are much smaller than the burp-test results. The theoretical prediction would be expected to underestimate τ since the damping effects of valves and bends are ignored. The burp test is expected to overestimate τ . This leads to the conclusion that the true value is probably somewhere in between these two.

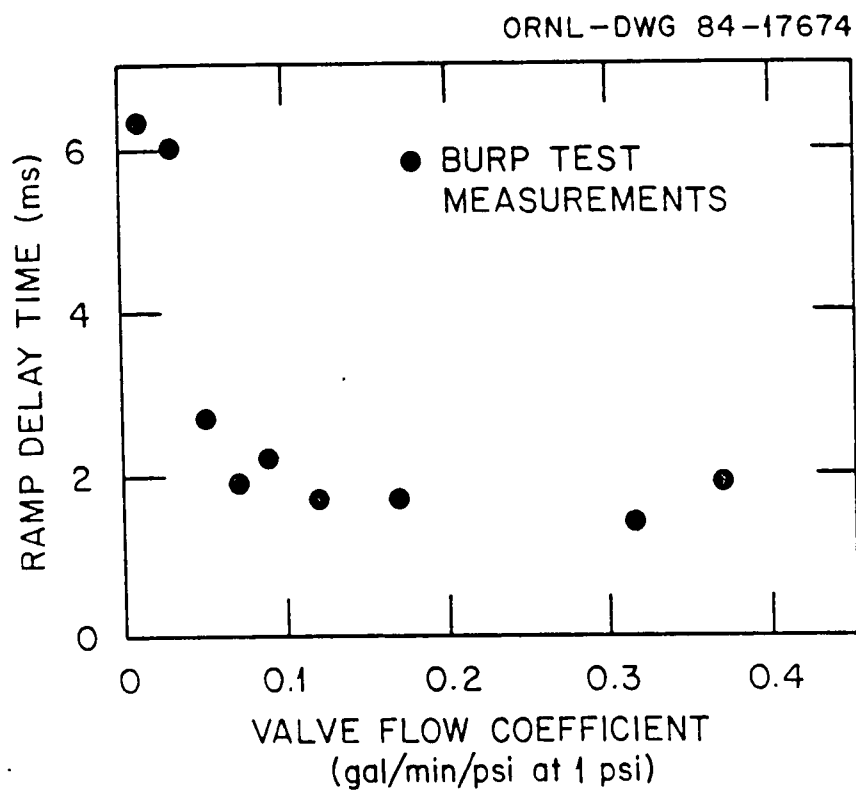


Figure 4-16. Burp Test Measurements of Asymptotic Ramp Delay Time versus Flow Coefficient for the Medium Pressure Facility.

Table 4-2. Comparison of Theoretical Estimates
with Burp-Test Measurements of Ramp Delay Time

	Theoretical*	Burp-Test Measurements
High-pressure	0.25 ms	2.7 ms
Medium-pressure		
6.2 cm ³ burp chamber	0.11 ms	1.9 ms
12 cm ³ burp chamber	0.11 ms	2.1 ms
19 cm ³ burp chamber	0.11 ms	2.2 ms

*Theoretical estimates are based on Equation (3-20), which does not include the effect of the burp chamber. Therefore the theoretical estimate does not depend on the size of the burp chamber.

CHAPTER 5

CONCLUSIONS

The objective of the burp test is to measure indirectly the ramp delay time of a sensing line. The measured value may then be added to the ramp delay times of the other components of the instrument channel to obtain the response time of the entire channel. It is the latter value which must be below a certain maximum, as discussed in Chapter 1. Typical pressure sensors used in nuclear power plants have response times on the order of 100 ms [3]. It has been demonstrated here that a typical sensing line has a response time on the order of 1 ms. The maximum allowable response time of the entire channel is typically 2 seconds. Thus we see that the contribution from the sensing line to the total response time of the channel is negligible under normal circumstances. It has also been shown here that the response time of a sensing line remains negligible unless the line is severely blocked. Depending on the severity of the blockage, a determination of τ by the burp test may or may not be possible, since such a determination requires the observance of damped pressure oscillations.

The burp test allows one to distinguish between three possible sensing line conditions: (a) unblocked, (b) partially blocked, and (c) severely blocked. An unblocked line would exhibit decaying pressure oscillations and would be determined to have a negligible response time. A partially blocked line would also exhibit the decaying oscillations and would also be determined to have a negligible response time. However, its response time would be greater than that of the unblocked line. A

severely blocked line would not exhibit the damped oscillations, and therefore no response time determination would be possible.

The partially blocked condition would presumably be distinguished from the unblocked condition by means of comparison with a benchmark case. Although the measured response time of a partially blocked line would almost certainly be negligible, it might be wise to flush out the line so that the partial blockage does not become more severe. Remedial action would certainly be required in the case of a severely blocked line.

Should the burp test be implemented in commercial power plants in partial fulfillment of NRC recommendations? To answer this question first consider whether or not it represents significant advantages over simpler methods. A crude method of sensing line testing would be to open a valve on a test port near the sensor end of the sensing line, and observe the resulting flow. By this method, an unblocked line could easily be distinguished from a severely blocked one. Also a partially blocked line could be diagnosed by measuring the flow rate, provided that a benchmark flow rate measurement had been made. Since the response time of a properly functioning sensing line is negligible, the burp test provides no more useful information than the crude flow rate test, and at considerably more expense. Second, in addition to the amount of information gained in a test, one should also consider possible harm to the system. Damage from water-hammer could be a problem in some cases. Also, since the coolant water in a PWR contains dissolved boric acid, each test would introduce a fresh supply of concentrated solution, which would then cool and possibly precipitate solid boric acid. Third, consider whether or not the information is really needed. A sensing line can tolerate

much blockage before ceasing to work well (see Figure 3-3, page 29). And severe blockages have obvious symptoms (e.g., dead sensor response, very limited frequency response) which could be detected in other ways. Fourth, there are unavoidable, inaccurate assumptions implicit in the mathematical model upon which the burp test is based. Perhaps the most fundamental of these is the uniform-line assumption. This assumption requires that blockages be uniformly distributed throughout the line. The errors introduced by violations in this assumption have not been investigated. Another deficiency of the burp-test method is its inability to test sensing lines with pulsation dampeners or snubbers.

It is concluded on the basis of the foregoing discussion that the burp test should be withdrawn from consideration as a candidate for dynamic testing of sensing lines in nuclear power plants. A possible method of sensing-line testing based on noise analysis was investigated concurrently with the burp test by M. E. Buchanan, et al. [18]. Though far from ideal, the noise-analysis method was found to be superior to the burp-test method.

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APPENDIX

APPENDIX

DERIVATION OF EQUATION 3-26

Consider a sensing line with localized blockage as depicted in Figure 3-2 (page 28). Neglect the change in compliance in the blocked region. The ramp delay time of the sensing line as a function of x is given by Equation (3-23) (page 27). With constant compliance C , resistance $R(x)$ as given in Figure 3-2 (page 28), Equation (3-23) (page 27) evaluated at the terminus of the line becomes

$$\begin{aligned}
 \tau_b &= C \int_0^{\ell} R(x)(\ell-x)dx \\
 &= C \left[\int_0^{x_1} R_n(\ell-x)dx + \int_{x_1}^{x_2} R_b(\ell-x)dx + \int_{x_2}^{\ell} R_n(\ell-x)dx \right] \\
 &= C \{ R_n [\ell(x_1-x_2) + (x_2^2 - x_1^2 + \ell^2)/2] \\
 &\quad + R_b [\ell(x_2-x_1) + (x_1^2 - x_2^2)/2] \} \quad . \quad (A-1)
 \end{aligned}$$

In the absence of blockage [$R(x) \equiv R_n$], the ramp delay time would be

$$\begin{aligned}
 \tau_n &= C \int_0^{\ell} R_n(\ell-x)dx \\
 &= R_n C \ell^2 / 2 \quad . \quad (A-2)
 \end{aligned}$$

The ratio of (A-1) to (A-2) is

$$\frac{\tau_b}{\tau_n} = [2(x_1 - x_2)/\ell + (x_2^2 - x_1^2)/\ell^2 + 1](1 - R_b/R_n) + 1 \quad . \quad (A-3)$$

From Table 3-1 (page 22) we have

$$R = 8\mu_o / \pi r_o^4 \rho_o$$

and therefore

$$\begin{aligned} R_b/R_n &= (r_n/r_b)^4 \\ &= (A_n/A_b)^2 \end{aligned} \quad (A-4)$$

where r_n , r_b , A_n , and A_b are the nominal and blocked radii and cross sectional flow areas respectively. By introducing the definition

$$\Delta x \equiv x_2 - x_1$$

into Equation (A-3) and by using the substitution (A-4) we get

$$\frac{\tau_b}{\tau_n} = 1 + \frac{\Delta x}{\ell} [2(1 - x_1/\ell) - \Delta x/\ell] [(A_n/A_b)^2 - 1] \quad . \quad (A-5)$$

VITA

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