SYNFUEL SYSTEMS :

A Review of Letdown Devices and Other Multiphase-Flow Problems

by

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Report No. E200.5

Division of Engineering and Applied Science
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September 1981

OUTLINE

I. INTRODUCTION

In an effort to develop domestic sources of energy, a number of processes to convert coal to clean fuels have been investigated. Products of these processes span the range from clean coal substitutes, to synthetic crude oil, to synthetic natural gas. Some of these processes, after successful operation in the laboratory, are now operating on a pilot plant scale to determine the feasibility of further scale up. These pilot plants range in capacity from 1/1000 to 1/10 of that projected for commercial operation.[1] At the pilot plant scale, problem areas which are generic to the conversion process have come to light. These problem areas include; handling of highly abrasive slurries, three phase heat transfer, high temperatures, high pressure coal feeding, and high pressure slurry letdown.

In this report the problems associated with high pressure slurry letdown from the reactor/disolver to the fractionating section will be discussed. The operating experience and the current state of the art for high pressure letdown valves in the coal conversion industry and the process industry will be examined. Also, the commonly used valve sizing techniques will be examined in relation to the problem of sizing for high pressure letdown. A summary of the instrumentation needed to monitor the letdown process for valve development, and ultimately for process control, is also included.

II. BACKGROUND

II.1 TYPICAL SLURRY LETDOWN CONDITIONS

High pressure letdown of a highly abrasive, high temperature slurry is a generic requirement of the coal conversion process. The SRC II (Solvent Refined Coal), H-coal, and EDS (Exxon Donor Solvent) liquefaction processes all require letdown systems capable of controlling the flow of a coal/ash-solvent slurry. Temperatures range from 500F to 900F with pressures of 2000 psi to 3000 psi let down to near atmospheric.[2] Similar letdown requirements are quoted by Hatchet et al in a survey of letdown valve availability.[3] As an example, letdown valve service conditions for one stage of a multi-stage slurry letdown system are given in table II.1.

Letdown valves used in the conversion process generally function as control valves. Types of letdown control valves used include differential pressure control, pressure blowdown, and slurry tank level controls.[3] The valve styles that have been used for letdown control include the globe, the angle, and the oil field choke valve. Modified oil-field choke valves and angle valves have had the greatest success in pilot plant operation.[3] Valves used in these applications at the pilot plant level have been one- or two-inch size.[1]

II.2 EXPERIENCE IN PILOT PLANTS

II.2.1 Erosion

The lack of reliability of system components in the pilot plants is due, primarily, to their inability to resist erosion from flowing coal/ash- solvent slurries.[4] Two approaches are available to combat the effects of erosion. Either sufficient material is provided such that the ability of the component to function is not hindered by erosive losses over a normal lifetime, or steps must be taken to prevent erosion.[5] Both tactics have been used with limited success. Valves have been constructed with sacrificial target plugs which are allowed to erode, thereby dissipating the kinetic energy of the particles in the stream.[6][5] Figure II.1 is a letdown valve used in the Louisiana, MO Bergius pilot plant. This valve was designed to handle a one step pressure letdown of 10,300 psi. Under these conditions valve life was approximately three days.[6]

Attempts at reducing erosion have also been made. Streamline flow valves have been tried with some degree of success.[8] One of the more common attempts at reducing erosion has been the installation of hard face trim in standard valve bodies.[3] In general, standard valve trim materials have provided unsatisfactory service, some operating as little as a few hours in the highly erosive environment.[6] Replacement of standard trim with tungsten carbide trim has greatly increased life under these conditions, but performance is still short of that desired for commercial plants (2.5 years).[6][9] A tabulation of valve trim life in four pilot plants is presented in table II.2.

Recently, an attempt has been made to test candidate materials' erosion resistance using actual letdown products from a SRC pilot plant.[4][10] In these tests an effort was made to provide a letdown condition similar to that encountered in actual letdown service. Samples tested in this manner showed equivalent erosion to that experienced when a free jet of the same flow rate and velocity impinged on a sample at an angle of 20 degrees. Data was also taken using ash-free anthracene oil to determine whether cavitation effects could account for a prominent portion of the erosion. It was found that erosion was negligible, indicating that the presence of the coal ash is the primary factor in the erosive behavior of the slurry.

II.2.2 Thermal Problems

High temperatures also present a problem in letdown system design. Difficulties occur both due to the nature of the slurry at high temperature, and mechanical problems. Letdown temperatures are generally greater than the normal boiling point of many components in the stream. This leads to flashing of the slurry within the valve during pressure reduction.[9] Also, in some processes, the letdown stream operates near the coking temperature. Problems may occur if large conglomerates of coke form in the stream, plugging pipes and valves.[3]

Mechanical problems stem from degradation of material properties and differential thermal expansion. The strength of most materials used in the manufacture of valves is reduced at high temperature. This effect has completely eliminated the use of some materials in the letdown systems.[11] Also, the resistance to erosion/corrosion of most valve body and trim materials has been found to decrease with increasing temperature.[4] Problems with differential thermal expansion have been so severe in some cases as to prevent valve operation.[12]

TABLE II.1	
SLURRY LETDOWN VALVE SERVICE CONDITIONS SCR-II PROCESS FIRST STAGE PRESSURE LETDOWN [7]	
Operating Temperature	775 to 800°F
Inlet Pressure	2,000 psig
Outlet Pressure	460 psig
Inlet Size	8" max.
Normal Flow Rate	1,300 gpm
Flow Turndown Ratio	2:1
Outlet Conditions	
Vapor	
Flow	22,700 lbm/hr.
Density	1.68 lbm/ft^3
Liquid	
Flow	1,250 gpm
Density	67 lbm/ft^3
Solids	
Percent Solids	22.8
Type	Ash and Untreated Coal
Particle Size Distribution	Weight Percent
1/8" max.	1 to 3 during lifetime
> 50 micron	0.5
15-50	2
10-15	2
6-10	10
4-6	15
2-4	35
1-2	21
0.2-1	11
< .2	< 4

TABLE II.2

COMPARISON OF TUNGSTEN CARBIDE TRIM WITH TRIM SUPPLIED WITH STOCK VALVE [9]

Process	Pressure drop (psi)	Temperature (^{0} F)	Original trim life (hr)	Tungsten carbide trim life (hr)
Synthane	600	670	72	770
HYGAS	1000	180	144	2000
SRC-I	1500	625-750	unsatisfactory	4000
SRC-II	1000	400-650	unsatisfactory	2000

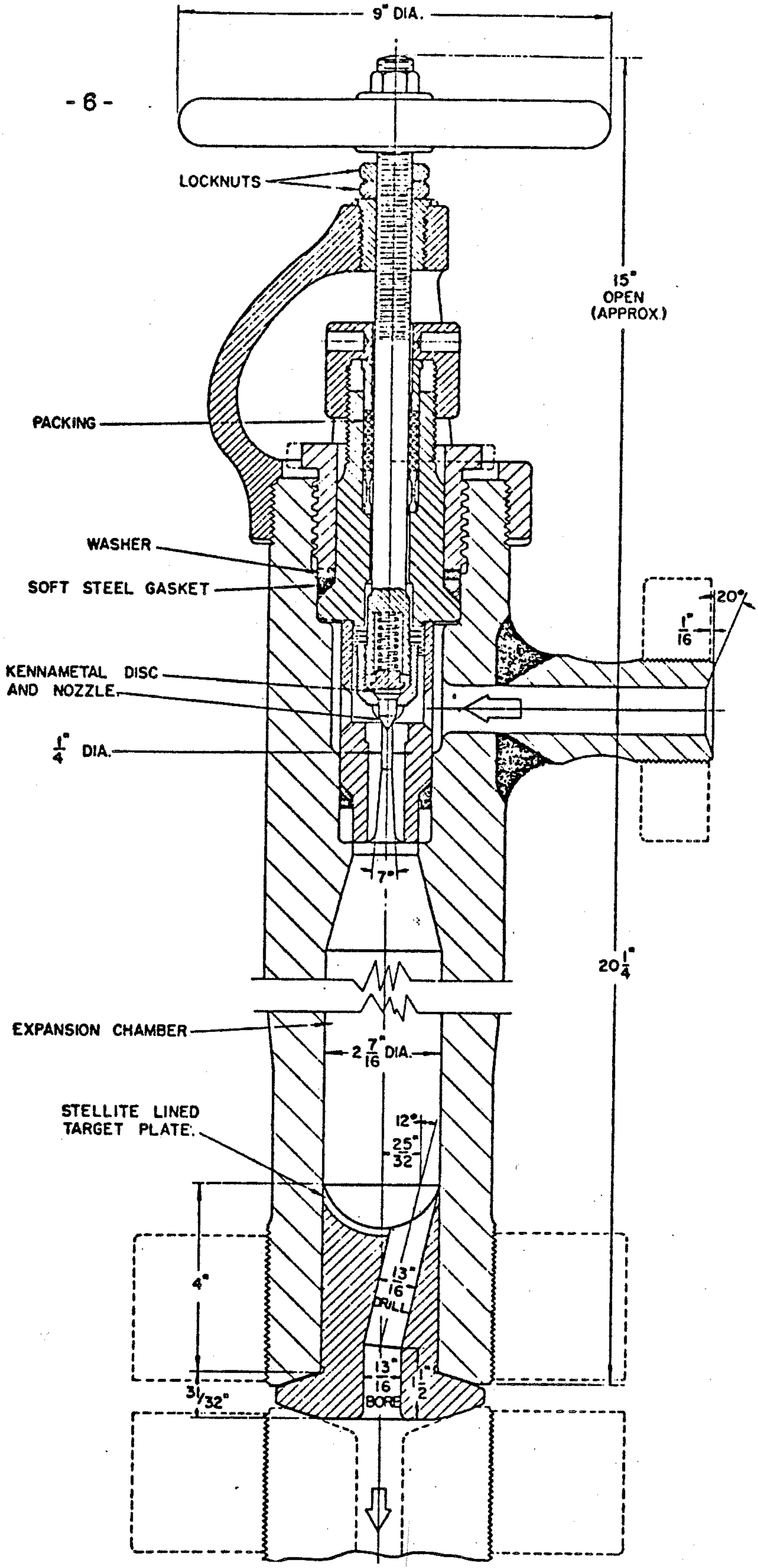


Fig II.1

Drawing of a Manual
High Pressure Letdown Valve
from Louisiana, Mo. [6]

III. CURRENT TECHNOLOGY / STATE-OF-THE-ART

An attempt is made here to examine the state-of-the-art in high pressure letdown valves from a different point of view. In the past, state- of-the-art surveys have consisted mainly of detailed accounts of individual valve performance in the various pilot plants.[1][3] In this report possible letdown valve types are generically classified. The performance and potential of the various generic groups is then examined. The objective is a greater overall understanding of the available alternatives.

III.1 GENERIC CLASSIFICATION

All passive letdown valves operate on a common principle, that is, the conversion of potential energy, stored as a result of the fluid differential pressure, into heat. The methods employed in this conversion process may be used to provide a generic classification system for letdown valves. The generic catagories to be used are: 1) Skin friction, 2) Jet formation, 3) Controlled vortex formation, 4) Tortuous path, and 5) Choked flow. The final generic class, choked flow, is not so descriptive of the means of dissipation as it is of a means of control. It is included here since devices employing this method of control are not adequately described by the other classifications. Active devices, such as turbines, in which energy is recovered from the process stream, have not been explored.

Very few valves in use may be said to fall in just one group, but most may be classified by their principle means of dissipation. The classifications are defined as follows:

1) Skin friction: Skin friction devices are those which rely on a shear force between the fluid and a fixed boundary to dissipate energy. These devices generally employ large surface areas in contact with relatively low velocity fluid.

2) Jet formation: Jet formation devices are those which employ the formation of a free jet, or a number of jets, to dissipate energy. Higher velocities are employed in these devices. Energy is converted from potential, to kinetic, and then dissipated by fluid shear in the body of the fluid.

3) Controlled vortex formation: Vortex devices are those which impart rotation (angular momentum) to the fluid and then dissipate energy through high shear rates induced by the destruction of this momentum. As in the case of the previous group, dissipation occurs in the body of the fluid.

4) Tortuous path: Tortuous path devices are those which dissipate energy through many sudden changes in direction. Jet formation is prevented by the presence of obstructions.

5) Choked flow: Choked flow devices are those which rely on the choking of a nozzle or orifice to control flow. These devices are independent of downstream pressure.

III.1.1 Skin Friction

Skin friction devices have several favorable characteristics when used for high-pressure letdown. Skin friction allows the dissipation of energy without high velocities. The absence of local high velocities lessens erosion problems, and tends to reduce cavitation caused by local low pressures. Another benefit of skin-friction throttling is that the fluid velocity normal to the surface is kept quite small, minimizing the impact angle and velocity of entrained particles.[13]

One problem inherent in skin friction throttling is the large area required for significant pressure reductions at low fluid velocities. If this area is formed by a large number of parallel paths having a small hydraulic diameter, plugging may be a problem when solids are entrained in the fluid. Entrained particles present further problems due to abrasion.

A novel approach to the control of fluid flow by the use of skin friction is the "silent valve". As shown in figure III.1, the control element of the valve consists of a cylindrical elastomer plug having a large number of parallel passages, through which fluid must flow. The fluid flow is controlled by varying the diameter of the passages in the plug by compressing it with a single-acting hydraulic cylinder.

Pressure reduction is accomplished through two processes; skin friction in the passage, and jet formation at the passage outlet. The relative magnitudes of the two effects may be seen by a simple comparison. Assuming turbulent flow the components of the pressure drop are:

$$\Delta P = \Delta P_{jet} + \Delta P_{friction}$$

$$\Delta P_{jet} = C_l . \tfrac{1}{2} . \rho . V^2$$

$$\Delta P_{friction} = f\left[\frac{l}{d}\right] . \tfrac{1}{2} . \rho . V^2$$

where C_l is the head loss coefficient ($C_l \leq 1$), ρ is the fluid density, V is the mean fluid velocity, f is the Darcy friction factor, l is the passage length, and d is the passage diameter. Thus, the pressure drop is dominated by skin friction losses for $\frac{l}{d}$ sufficiently large.

III.1.2 Jet Formation

Jet formation is the most common mode of energy dissipation used in throttling valves. Valves employing jet formation include; the common globe, the angle, the butterfly, and the ball valve. The simple orifice restriction also employs this principle.

Jet formation requires the acceleration of low velocity fluid to a higher velocity. The energy required to accelerate the fluid is acquired through the reduction of static pressure in accordance with the Bernoulli equation.

The fluid is then decelerated, converting some of the kinetic energy into a static pressure rise, while the remainder is dissipated as heat. Two important characteristics of throttling by jet formation are indicated above. The maximum amount of energy which may be dissipated in one stage is equal to the kinetic energy of the fluid in the jet. Secondly, the static pressure in the jet is generally lower than the valve outlet pressure. Both of these characteristics pose problems in high pressure letdown valves.

Large pressure reductions, require the jet to possess a large amount of kinetic energy, implying that high velocities must exist within the valve. The problem of erosion under these conditions becomes very serious when an abrasive slurry is throttled. Erosion problems become so severe in globe valves that their use is not recommended for highly abrasive slurry service.[11]

Cavitation induced by low pressures within the valve may also cause considerable erosive damage. In an effort to reduce damage caused by cavitation two techniques have been used.[14] The first method is the elimination of cavitation by taking the pressure drop in stages so that lower kinetic energy jets are required at each stage. This method also reduces erosion due to abrasive slurries. Examples of multi-stage valve trim designs which employ jet formation are shown in figure III.2.

The second method is to confine cavitation to the bulk of the fluid, away from walls where damage might occur. Examples of this approach are the use of diametrically opposed jets which impinge on one another[14], and the discharge of an angle valve into a vessel, using the large mass of fluid to "absorb" the energy.[14]

Erosion damage due to abrasive slurries may also be reduced in these valves by the use of streamlined valve bodies.[8] Erosion problems, in this case, may also occur in downstream piping due to high exit velocities. Figure III.3 depicts an experimental streamlined angle valve used in the EDS pilot plant.

III.1.3 Controlled Vortex Formation

Devices using controlled vortex formation as a means of energy dissipation are the rotational analogue of jet formation devices. In controlled vortex formation devices, angular momentum is created and destroyed as a means of energy dissipation.

Important differences between the two categories involve the dependence of angular momentum on the radius from the center of rotation, and radial pressure gradients due to centrifugal force.

One use of controlled vortex formation to regulate flow in high pressure letdown has been in a boiler feed water pump bypass valve.[11] A picture of this type of valve is shown in figure III.4. In this application the valve was designed as a multi-stage unit. Fixed vanes located in an annular flow passage impart rotation to the fluid within each stage. As the fluid passes from one stage to the next the flow rotation is reversed, inducing high shear rates to dissipate energy. This type of valve performs well in flows which tend to cavitate since vapor bubbles are kept away from the outer wall by the radial pressure gradient.[11]

A fluidic device which may be used for flow control is the "vortex valve".[15][16] The vortex valve requires no moving parts in the main fluid stream, only the control stream need be metered. A schematic diagram of a vortex valve is shown in figure III.5. In the valve shown, fluid enters around the periphery of a cylindrical chamber. The entering fluid is divided into two streams; the main stream enters radially toward the center, while the control stream is injected tangentially. The flow rate of the control stream determines the rotation rate of the fluid within the chamber. The rotation of the fluid produces a radial pressure gradient, controlling the fluid flow rate in the main stream. An outlet is provided at the center of the vortex chamber to discharge the low pressure fluid.

Energy is dissipated by high shear rates generated as fluid is forced to follow an inwardly spiralling path. These high shear forces occur due to the principle of conservation of angular momentum, which requires the angular velocity to increase as fluid travels toward the center of rotation in the absence of external torques.

The application of vortex valves to fluids containing suspended solids may present problems if the solids are more dense than the liquid. Radial pressure gradients may tend to encourage phase separation.

III.1.4 Tortuous Path

Globe or angle valves with tortuous path type trim are often used when cavitation is anticipated.[17] The dissipation mechanism used is that of small scale turbulence. Each time the fluid direction is forced to change, turbulence is generated locally. The tortuous path device, in the limit, effectively becomes a porous plug.

A problem arises in the use of tortuous path valve trim when solids are present in the fluid. Small passages in the trim are easily plugged, eventually leading to valve blockage. For this reason tortuous-path type devices are generally not considered acceptable for slurry letdown.

III.1.5 Choked Flow

Devices which operate on the principle of choked flow are often used in high-pressure letdown.[17] Choked or "critical" flow is not a phenomenon which is determined solely by the valve geometry, as are the other classifications. Choking is a phenomenon associated with the fluid. For this reason, valves using choking as a means of control must be designed for a particular process.

The oil-field choke valve has been used for high pressure letdown of two-phase fluids in the oil industry. The oil-field choke valve is characterized by a long tube commonly referred to as a "choke". The choke is made long enough so that the critical flow condition is independent of small changes in choke length. The diameter of the choke determines the flux for any given upstream stagnation conditions. Also available are variable choke valves in which a means of throttling is provided at the choke entrance. A variable oil-field choke valve is shown in figure III.6. It should be noted that the choke acts as a jet-formation device when the outlet pressure is to high to allow critical flow.

Some advantages of the choke valve are its simple design, and relatively high resistance to damage from entrained solids.[17] Chokes are usually arranged to discharge into a downstream receiver to dissipate the high discharge kinetic energy.

A method which has been suggested for high-pressure letdown of an erosive slurry is the use of a number of fixed chokes.[17] This concept involves the use of a number of chokes operating in parallel. Each choke is equipped with an on-off valve upstream allowing it to operate in either a full-on or full-off condition. This method has the advantage of eliminating moving parts near the choke which are subject to high local velocities resulting in erosion damage.

By sizing the chokes such that each successive choke have twice the flow as the last, equal increments of flow may be obtained. In such a system having n chokes there are 2^n discrete operating points and a turndown ratio of 2^n-1. This type of system, being "digital" in nature, is especially suited to computer control.

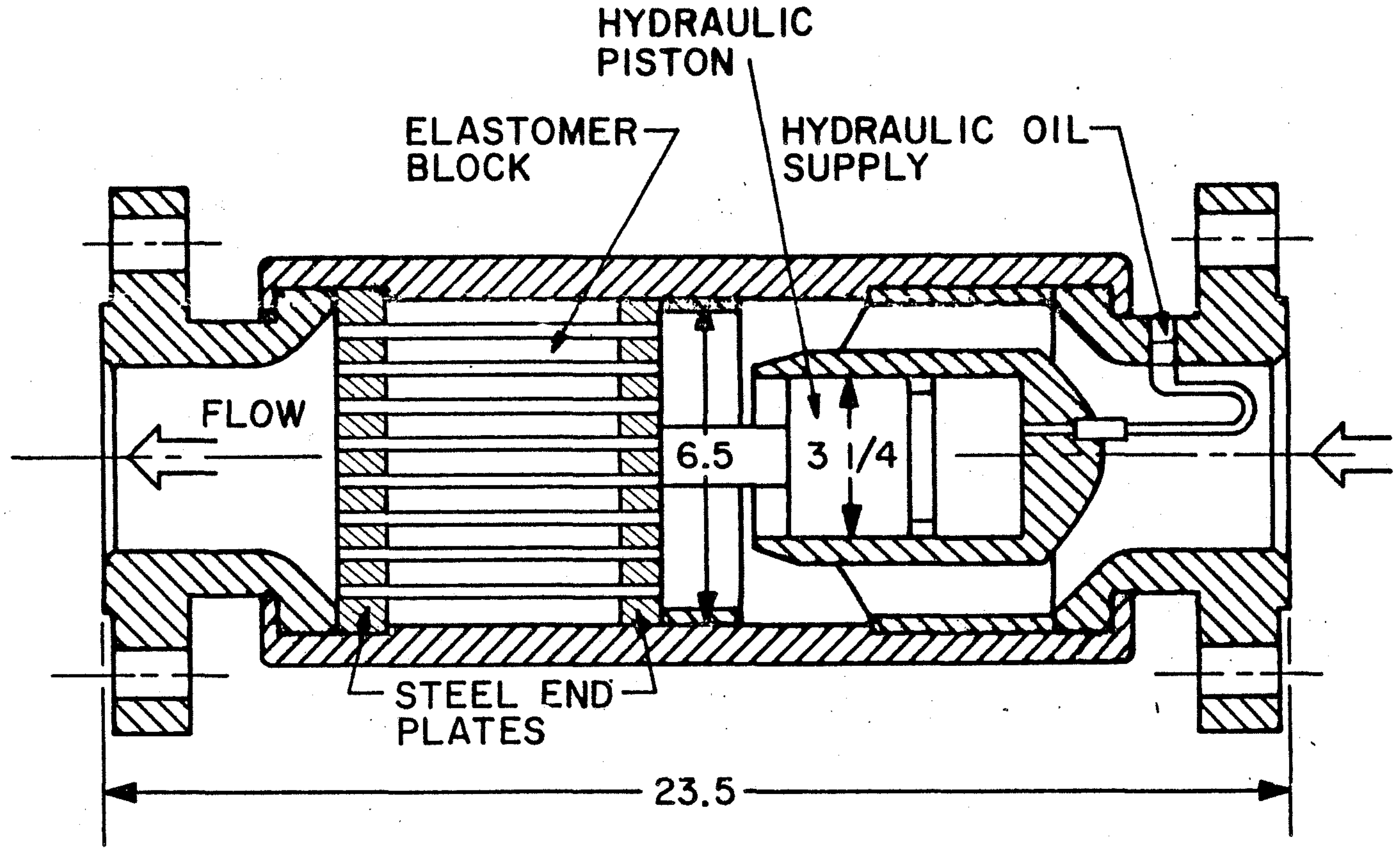
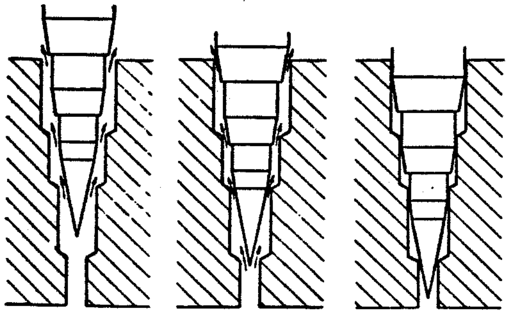
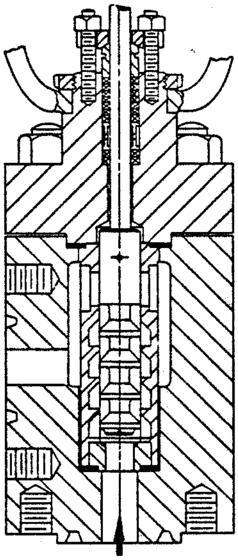


Fig III.1 Schematic drawing of the silent valve. [50]



MULTIPLE VELOCITY
HEADLOSS TRIM.

Fig III.2 a



Abrasive and cavitation erosion in high pressure drop liquid service are practically eliminated in this valve design which reduces pressure in multiple small steps thus maintaining much lower velocity than conventional valves. Conventional single plug orifice valves exhibit an average velocity head loss coefficient of 0.8 whereas this design has a K value of 20 due to the continuous turns the fluid is subject to when passing through. (Head loss $H = V^2/2g$ Each port of the plug sees only 5% of the pressure drop experienced in a conventional valve. The throttling process is nearly identical to a constant enthalpy flow with friction much like pressure drop in a long pipe line. Long guiding is restricted to 4 corners of the circumference which prevents jamming when flowing particle entrained liquids. Working pressures range to 6000 psi in sizes 1/2 to 2". Flow capacity is 1/16 to 1/4 that of a conventional low pressure drop valve. Available in TFE soft seat design for drop tight shutoff.

(b) Cascade Flow Restricting Orifices

Courtesy
Masoneilan International, Inc.

Fig III.2 b

Multi-stage Trim Designs [11]

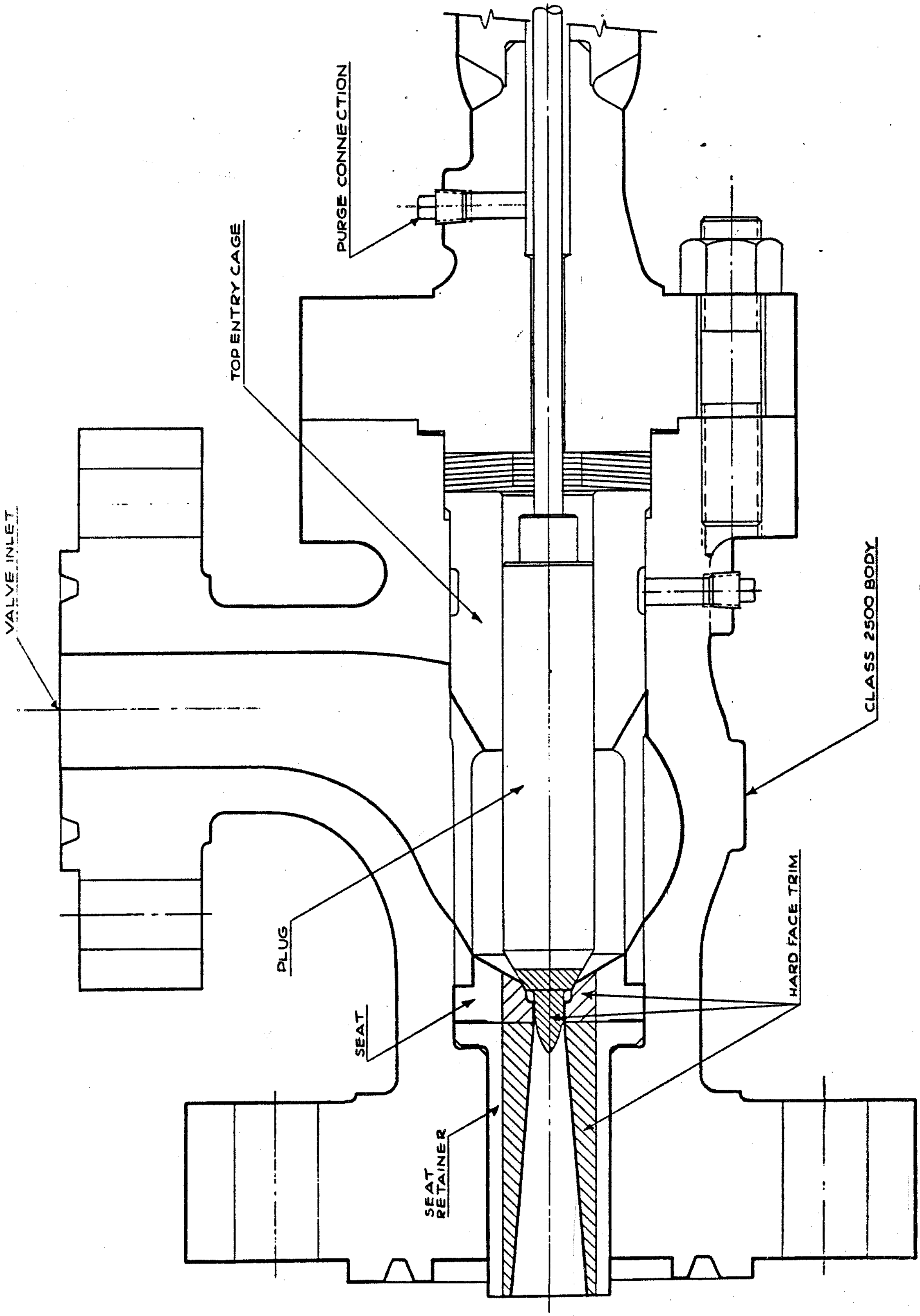
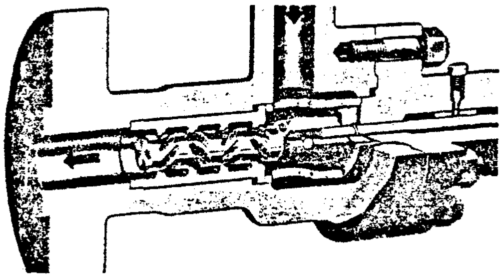


Fig III . 3

Streamlined Angle Valve [8]



(a) Turbo-cascade® Control Valve

Fluid vortexes are produced within each stage which may take up to 1000 psi pressure drop. Fluid is turned 90° as it passes through the flutes from one stage to the next, and flutes spiraling around the plug force fluid against the containment wall by centrifugal force. Cavitation vapor bubbles tend to remain in the main stream away from the walls since they are lighter than the fluid. This design has been applied to regulating boiler feed water pump bypass.

Courtesy Yarway Corporation

Fig III.4 [11]

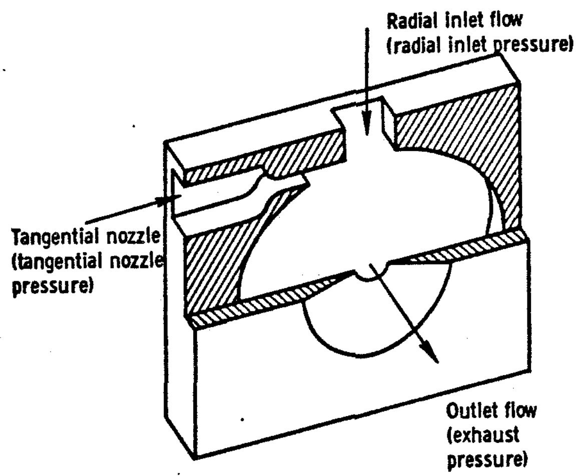


Fig III.5 SCHEMATIC ILLUSTRATION OF THE VORTEX VALVE [16]

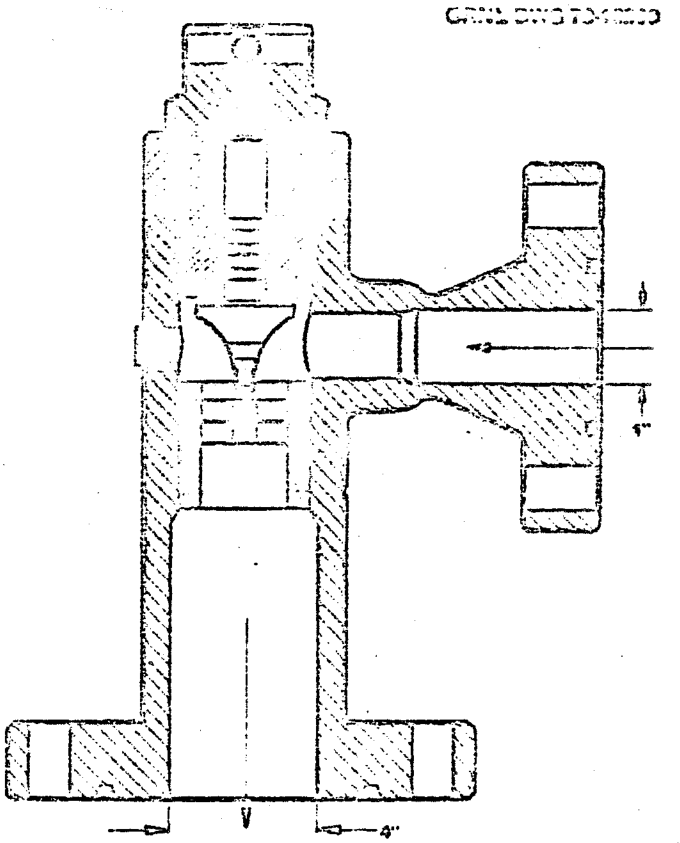


Fig III.6 , Willis M-2 choke valve (manufactured by Willis Oil Tool Co.). Source: S. E. Carson and O. D. Runnels, *Valve Applications at the Synthane Plant In Mixed Phase Erosive Service*, COO-8883-23, The Lummus Company, June 1978. [3]

IV. CONTROL VALVE SIZING METHODS

To appreciate the capabilities and limitations of today's control valve sizing methods, it is necessary to examine the premises on which they are based. The sizing equations commonly used for control valves are based on an analogy between a control valve and an orifice.[11] The sizing formulas consist of the equation for flow through an orifice modified by a string of multiplicative factors. These factors include an experimentally determined valve coefficient, C_v, and empirically determined modifiers to correct for flow regime.[18]

IV.1 SINGLE-PHASE FLOW

Valve sizing formulas for single phase fluid flow are readily available in the valve industry. The ISA (Instrument Society of America) has published a set of standards for control valve sizing single phase flow.[19][20] The sizing formulas presented in these standards will be examined here.

IV.1.1 Incompressible Flow

IV.1.1.1 Turbulent flow

In turbulent flow, the capacity of a valve is determined by a single experimentally determined coefficient called the valve coefficent, C_v. The equation relating the flow to the pressure drop is:

$$w = N.C_v\sqrt{\gamma_1.\Delta P} \quad (1)\ [11]$$

where w is the mass flow rate of fluid through the valve, N is a dimensional constant to account for units, ΔP is the pressure loss across the valve, and γ_1 is the specific weight of the fluid at the valve inlet.

IV.1.1.2 Viscous flow

At low Reynolds numbers the valve coefficient is no longer constant. [18] To account for this effect a Reynolds number correction is introduced. This correction factor is applied to appropriately reduce the valve coefficient for use in equation (1). An empirically developed correction curve is provided in the standard to determine the correction to the valve coefficient for a given Reynolds number. In an attempt to account for mechanical dissimilarity, a "valve style modifier" has been introduced to correct the Reynolds number. One criticism of this method is that the shape of the curve is not allowed to vary with valve type.[18] The dependance of the Reynolds number correction factor does appear to be asymptotically correct in that it predicts a linear relationship between flow and pressure drop at low Reynolds numbers.

Another problem encountered in sizing valves for laminar flow is that of non-Newtonian behavior. In practice, low Reynolds numbers are generally encountered due to extremely high viscosities. However, many liquids which exhibit high viscosities also exhibit non-Newtonian behavior.[18]

IV.1.2 Compressible Flow

Sizing control valves for compressible flow service is based on the same orifice model as used in the incompressible case, the only alteration being the admission of compressibility.[18] The sizing relation for compressible flow through a valve is generally written as:

$$w = N.C_v.Y\sqrt{\gamma_1.x.P_1}. \quad (2)\ [11]$$

where Y is the expansion factor, P_1 is the inlet pressure to the valve, and x is defined as $\frac{\Delta P}{P_1}$. It should be noted that if a value of unity is assigned to Y, equations (1) and (2) are identical.

The expansion factor, Y, may be calculated exactly for venturis and flow nozzles.[19] The value of Y is known to depend only on the isentropic exponent, k, and the pressure drop ratio, x, in these cases. For valves, the functional relationship of the expansion factor on the pressure drop ratio has been empirically determined to be:

$$Y = 1-\frac{x}{x_T}.\left[\frac{1}{3.F_k}\right] \quad (3)\ [11]$$

where F_k is the ratio of the isentropic exponents of the gas flowing in the valve and air, and x_T is the value of the pressure drop ratio at which choking occurs on air. The value of x_T is experimentally determined.

This relation yields a maximum value of unity, which is the incompressible case, and a minimum value of two-thirds at choked flow. (The minimum value of the expansion factor is determined by maximizing the mass flow rate in equation (2) with respect to the pressure drop ratio, assuming the expansion factor is a linear function of x.[11])

The functional dependence of the expansion factor exactly matches that commonly used for sharp edged orifices, but differs considerably from that predicted theoretically for venturis and nozzles.[19] Therefore, it is reasonable to suspect that the mechanical configuration of a valve has an influence on this functional dependence. This implies that the linear relationship might not be valid for some valve types.

IV.2 TWO-PHASE FLOW

In general, there are no widely accepted methods for sizing control valves used in two-phase flow service. A few methods do exist which allow proper sizing under a limited range of conditions.[18][11][21] Before discussing these methods it is instructive to examine the models commonly used in the study of two-phase flow.

IV.2.1 Flow Models

IV.2.1.1 Homogeneous flow

Homogeneous flow theory provides the simplest method for analyzing two-phase flows. In homogeneous flow theory the two-phase mixture is assumed to act as a pseudo-fluid which obeys the usual single-phase flow equations.[22] The properties of this pseudo-fluid are weighted averages of the component properties.[23]

A further simplification is often made by assuming the phases are in equilibrium (i.e. equal velocity, temperature, pressure, etc.). This model is known as the homogeneous equilibrium model (HEM).

The HEM has been shown to accurately predict critical mass flux in steam-water flows in long pipes.[22] On the other hand, rapid acceleration and pressure changes make the HEM inaccurate for describing the discharge of flashing steam-water mixtures through short nozzles or orifices, because of substantial relative motion between the phases under conditions of rapid acceleration.[57]

Modifications of the equilibrium assumptions are often made to include non-equilibrium effects. Examples of these models are; the "frozen flow" model in which no phase change is allowed[22], "slip flow" models in which a velocity ratio for the two phases is assumed[24], and models which assume some degree of thermal non-equilibrium between the phases.[23]

IV.2.1.2 Two-fluid models

The two-fluid model takes into account the fact that the two phases may have differing properties and different velocities.[23] The model is expressed in terms of two sets of conservation equations governing the balance of mass, momentum, and energy in each phase. These conservation equations, together with rate equations which describe the interphase transfer of mass, momentum and energy, are solved simultaneously to determine the macroscopic two-phase behavior.[25]

The primary advantage of the two-fluid model is that is can take into account dynamic interactions between the phases.[25] Also, processes which involve counter-current flow or phase stratification may be modeled accurately with this formulation.[23]

A difficulty which occurs in applying the two-fluid model to engineering problems is the prediction of flow-regime.[26] In two-phase flow many different flow-patterns may exist depending on constituent flux rates and a number of other parameters. As an example, the flow-patterns occurring in air-water flow in a vertical pipe may be classed as bubble, slug, churn, annular, or "wispy-annular" flow as shown in figure IV.1. These flow patterns have a great influence on the interphase transfer of mass, momentum, and energy, and must be know prior to or determined as part of the solution of the flow field.[26]

In theory, the two-fluid model may be used to predict flow-patterns if the three-dimensional formulation is used with proper constitutive relationships.[25] In practice, it is often not practical to model in the detail required for flow-pattern prediction, or the constitutive relations necessary are not available.[26][25] In this case the flow regime must be determined by some means external to the model. In an attempt to circumvent this problem a number of computer codes using two-fluid models have used flow-regime maps.[26][27]

Flow regime maps have been presented by numerous authors. These maps consist of a two-dimensional plot of flow-regimes as a function of two independent variables.[23] The most common coordinates for liquid-gas flows are the liquid flow rate and the gas flow rate.[28] The use of flow-regime maps has been criticized by many authors since the areas covered by a particular regime tend to be a function of other variables not included in the map.[28][26][23]

Although problems do exist with the two-fluid model, it appears to have the capability to solve engineering problems.[22] Two-fluid modeling is used extensively in nuclear reactor modeling computer codes.[29]

IV.2.2 Critical Flow

Critical flow phenomena may be the controlling factor in the discharge of a fluid from a high pressure reservoir to a receiver at a lower pressure. Critical flow is defined as flow conditions, under which, the mass flux is independent of downstream pressure.[28]

The critical flow rate of a single phase gas occurs when the fluid velocity equals the sonic velocity at some point in the system. The gas is generally regarded as being in thermodynamic equilibrium due to the short molecular relaxation time as compared to the gas velocity.[22]

Two-phase critical flow becomes more complicated. The equilibrium assumptions valid in single-phase gas dynamics are no longer valid in many cases.[22][23][29] Processes such as heat, mass, and momentum transfer involved in the evolution of new flow patterns may have relaxation times on the order of the fluid residence time in the "critical" region.[22] For this reason, although criticality may be mathematically indicated to occur at a point, a much larger region plays a role in how the "critical" condition is approached.[22]

The calculation of critical flux rates has received considerable attention in the nuclear industry.[22][27][29] Much of this work has been done in an attempt to model reactor transients during a loss-of-coolant- accident (LOCA). Most of the work involves steam-water or sub-cooled water. Both analytical and experimental work have been done.

Experimental investigation indicates the critical mass flux is geometry dependent owing to the long relaxation times involved.[29] The behavior of a a given fluid in a nozzle may be viewed as a function of the fluid stagnation conditions. Experiments involving the use of a single nozzle with various inlet stagnation conditions indicate that the behavior of flows with sub-cooled, near saturation, and saturated two-phase stagnation conditions are quite different.[29]

IV.2.3 Valve Sizing In Two-Phase Flow

Although no valve sizing equations exist which cover the entire range of two-phase flows, some special cases have received attention. Some of these special cases are listed in the following sections.

IV.2.3.1 Solid-liquid

The sizing of valves to handle slurries in industrial applications is normally handled by using an adjusted density in the single-phase liquid sizing equation.[17] This technique is just an application of homogeneous flow theory to the two-phase flow. Investigation into the hydrodynamics of two-phase solid-liquid flow has been conducted in connection with hydrotransport of coal and other materials.[30] Results of these investigations may prove to be of use in valve sizing, although most have not investigated accelerating flows which may be experienced in valves.

IV.2.3.2 Solid-gas

Sizing methods for solid-gas flows in the process industry seem to be limited to homogeneous flow theory. Driskell recommends the use of homogeneous theory with an expansion factor applied to the gaseous phase.[17] The single-phase gas sizing equations are then used to predict flow.

Research on the hydrodynamics of two-phase solid-gas systems has been conducted in regard to solid propellent particles in rocket nozzles.[23] This work considers high accelerations and might be applicable to valves. Work has also been done on the flow of solid-gas mixtures through pipes and venturi tubes in the field of pnuemotransport.[31]

IV.2.3.3 Liquid-gas

A significant amount of work has been done on liquid-gas flows. Generally these flows are handled differently depending on whether the gaseous phase is condensable or non-condensable.

IV.2.3.3.1 Non-condensable gas

Some early attempts at sizing for liquid-gas flow were made on the assumption that the net flow might be calculated by superposition.[21] In this method the required valve coefficient for the gas flow alone and for the liquid flow alone are calculated and summed to yield the required valve size. The result of these attempts, as one might expect, was to undersize the valve considerably.[21] This is due to the strong coupling of the liquid and gas phases being completely neglected.

More recently, the trend has been toward the use of a homogeneous flow model.[18] The average properties used in these model are derived from the mass flow rates of the two components, not the local void fraction.[18][21] Experimental data for an air-water mixture throttled at a variety of void fractions and pressure drops agrees well with the homogeneous model.[21]

IV.2.3.3.2. Condensable gas

Liquid-condensable gas systems tend to be more difficult to deal with than liquid-non-condensable gas systems. Difficulties arise due to non-equilibrium effects involved in the mass-transfer process.[18]

The most common liquid-condensable gas system is the presence of a liquid and its vapor. Liquid-vapor two-phase flows encountered in valves may be classified by the fluid condition at the inlet and outlet of the valve. The common classifications are: "cavitating" flows, "flashing" flows, and flows with two-phase inlet conditions.

IV.2.3.3.2.1 Cavitating flow

As used in the valve industry "cavitation" refers to the formation and subsequent collapse of vapor bubbles in the fluid stream.[14] In general, work done on cavitation in the valve industry assumes the fluid at the inlet and outlet are both all liquid, and the inlet is sub-cooled.[11]

Cavitation occurs when the local pressure becomes low enough so that vapor bubble growth is induced. These low pressures are usually due to local high velocities within the valve. As the fluid slows, the pressure increases causing the bubbles to collapse. The collapsing vapor bubbles induce strong pressure waves in the surrounding fluid which may cause severe damage to the valve.[14]

No known engineering material, available today, is capable of withstanding the stresses imposed by severe cavitation indefinitely.[11] It is for this reason that the ability to predict when cavitation will occur is desirable. The most commonly used measure of a valve's propensity to cavitate, is the "cavitation index", K_c .[14] The cavitation index is defined by,

$$K_c = \frac{\Delta P_c}{P_1 - P_v}, \quad (4)\ [11]$$

where P_v is the vapor pressure of the fluid at the valve inlet, P_1 is the inlet pressure, and ΔP_c is the pressure drop across the valve at which a two percent deviation from equation (1) occurs. The value of the cavitation index varies greatly with valve style, ranging from .15 to .80.[11]

IV.2.3.3.2.2 Flashing flow

The second class of liquid-vapor flows commonly encountered is "flashing" flow. In the valve industry, "flashing" refers to the formation of vapor bubbles in the valve which do not collapse upon traveling down stream. This condition occurs when the downstream pressure is below the vapor pressure of the upstream fluid.[11] Under this condition the flow does not continue to increase as the downstream pressure is reduced. The flow is therefore "critical" at some point in the valve.

Sizing valves in the "critical" flow region is covered in the ISA standards.[20] The sizing formula is based on the control valve, orifice analogy. The sizing formula assumes that no phase change occurs prior to the vena contracta (frozen flow). The pressure at the vena contracta is then calculated from an empirical relation for the minimum vena contracta pressure using the inlet vapor pressure and critical pressure. This vena contracta pressure, P_{vc} , is then used to calculate the flow. i.e.

$$w = N.C_v.F_L \sqrt{\gamma_1.(P_1 - P_{vc})} \quad (5)\ [18]$$

,where F_L is an experimentally determined coefficient relating the valve outlet pressure to the vena contracta pressure under single- phase operating conditions.

The accuracy of this method rests on several factors. Probably the most important factor is the determination of the vena contracta pressure. The formulae which are available to determine this pressure are only accurate when the pressure is far from the critical pressure.[18] When the fluid contains many components, the appropriate values for use in the formulae are a matter of conjecture.[17]

It should be noted that research has been conducted in the area of critical flows having sub-cooled stagnation conditions. Papers by Wallis and others present information concerning the problem of critical flow in these situations which should aid in the understanding of these flows.[32]

IV.2.3.3.2.3 Two-phase inlet conditions

The third class of liquid-vapor flows treated are those with two- phase inlet conditions. The case of critical flow with two-phase saturated stagnation conditions is covered by Wallis[22], Isbin[29], and others.

Two-phase pressure drop due to abrupt area changes has been investigated by Weisman, et al [33] using Freon 113 and its vapor. Weisman also modeled the flow using a one-dimensional momentum balance and found that good agreement was obtained by assuming "slip flow" upstream and downstream, but "homogeneous frozen flow" at the vena contracta.

Calculation of flow in low quality streams has presented great difficulty.[18] Calculations of the flow of low quality steam-water mixtures using the homogeneous equilibrium model have under-predicted the flowrate by as much as a factor of ten. Assuming "frozen flow" up to the vena contracta still under-predicted the flow by a factor of two.[11] These problems have also been encountered in low stagnation quality critical flow calculations and have been attributed to non-equilibrium effects.[29]

IV.3 THREE-PHASE FLOW

Information of valve sizing in three-phase flow does not seem to exist. The standard modeling techniques used in two-phase flow such at the "homogeneous flow" and "two-fluid" models may be easily extended to three- phase flow in theory. Problems encountered in applying these models will most likely be due to the lack of experimental data to test theory. Different flow regimes may be expected to appear in three-phase flow requiring modified constitutive relations.

A real limitation in the study of three-phase flow hydrodynamics is the lack of information at the most basic level. At present it is not possible, in general, to describe these flows even in the steady-state. If the complexities of the non-steady, reacting, three-phase flows encountered in the coal conversion process are to be effectively dealt with it is first necessary to understand the processes involved in these flows. A first step in this direction would logically involve the investigation of a three-component, non-reacting fluid flow in the steady state. Results of of this type of research are essential to the understanding of the non-steady, reacting, three-phase fluid flows which have been a generic problem in the coal conversion industry.

Some work in three-phase flow has been done in relation to liquid lift pumps to be used in hydro-mining.[34] Also, a bibliography of research in three-phase hydrodynamics has been prepared in conjunction with the development of the H-coal ebulated bed reactor.[35]

IV.4 SUMMARY

The state-of-the-art in control valve sizing is acceptable in the areas of turbulent, liquid flow with no phase change and turbulent gas flow.

Valve sizing methods for viscous liquids rely on factors such as "valve style modifiers" which have only been determined for a few valve types.[20] The ability to account for extensive mechanical differences using one empirical coefficient is open to question.

In two-phase flow the state-of-the-art in the valve industry appears to be limited to the various "homogeneous flow" models. The application of two-fluid models appears to have great potential for more accurate control valve sizing. These methods covered by Wallis[22], Isbin[29], and others.

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In two-phase flow the state-of-the-art in the valve industry appears to be limited to the various "homogeneous flow" models. The application of two-fluid models appears to have great potential for more accurate control valve sizing. These methods are more detailed than the simple "orifice- control valve" analogy in common use. This detail is required for improved accuracy, but is not likely to be used by process engineers for routine sizing of valves. The use of the two-fluid model is justified in critical applications such as high pressure slurry letdown encountered in the coal conversion process.

Problems encountered in "cavitating" and "flashing" flows are linked closely with "critical" flows having sub-cooled stagnation conditions. Work done in the nuclear industry on "critical" two-phase flow should be useful in dealing with these problems.

Difficulties in sizing valves for low quality flows are due in part to the existence of non-equilibrium effects. The understanding of these effects is necessary in this region.

No sizing methods for three-phase flow seem to exist. Knowledge of the hydrodynamics of three-phase flow is very poor compared with single- or two- phase flows. Limited experimental data makes it difficult to generate empirical correlations which might be used to test theory.

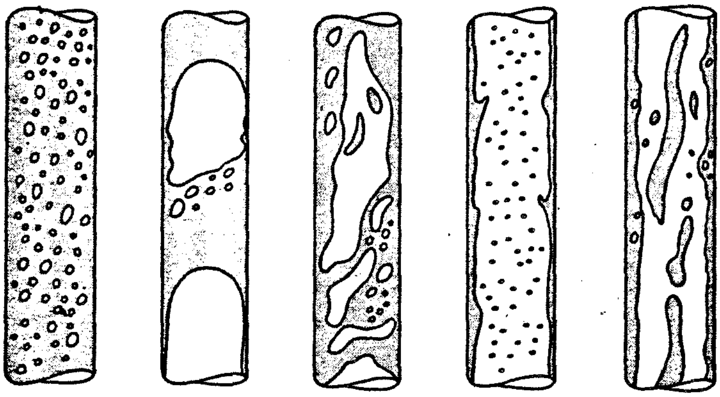


Fig IV.1

Flow patterns in vertical upward flow. [24]

V. ACTIVE LETDOWN DEVICES

The use of active devices for high-pressure slurry letdown is attractive. It has been estimated that the thermodynamic availability of the letdown stream from a 6000 ton per day coal liquefaction plant is on the order of 3600 horsepower.[3]

The German Bergius process, operated during World War Two, used a combination of active and passive letdown.[6] Letdown valves were used for the heavy oils and solids, while a machine consisting of side by side pistons was used to letdown clean liquids downstream of the secondary hydrogeneration reactor. The energy extracted from the flow was used to pump a feed-slurry into the reactor. Pumping power savings using this system were reported to be about seventy percent.

In a survey of industrial coal conversion equipment, Williams et al reported that no letdown energy recovery devices were in use in any of the pilot plants visited.[36] The reason given for this was the small plant size, making the energy recovery uneconomical. Williams et al also investigated the availability of hydraulic turbines for high-pressure slurry letdown and found no suitable equipment to be commercially available.[37]

Most hydraulic turbines used in the process industry are basically centrifugal pumps slightly modified to operate in reverse rotation.[36] Mattson and Glenn, in their industry survey of coal slurry pumps, have indicated some pumps which may have potential as letdown devices.[38]

Analytical and experimental work has been performed in the nuclear industry which may be of use in designing slurry turbines. A report by Wilson and Chan describes the operation of a centrifugal pump under two- phase flow conditions.[39] Data was taken operating in all four quadrants of the head-flow plane.

Impulse turbines, such as the Pelton wheel, are generally used for high head application in hydroelectric power generation. The impulse turbine, generally favored due to its high efficiency, has found little use in the process industry.[36]

One of the main problems which must be overcome in the development of slurry turbines is erosion. Erosion problems are expected to be even greater than those encountered in slurry pumps. For this reason, it has been suggested that slurry turbine development be postponed until the slurry pump erosion problem is resolved.[9] In an effort to alleviate the erosion problem, positive displacement devices have been suggested which have a long stroke, reducing the number of valve cycles, thereby reducing wear. Such a device is described in the patent literature.[40]

Another area of research in which energy must be extracted from a two-phase fluid is geothermal energy. A turbine designed to operate with a two-phase inlet condition supplied from a geothermal source may also be found in the patent literature.[41]

VI. SCALING

The ability to scale fluid mechanical systems is very valuable in research and engineering. Scaling parameters have been proposed by numerous authors to account for various effects.[31][42]

Ishii[42] obtains nine dimensionless groups by non-dimensionalizing the continuity, momentum, and energy equations obtained in his two-fluid model. This non-dimensionalization assumes only one characteristic length is present in the system. This length may not be well defined in some systems. Ishii classifies these dimensionless groups as kinematic, dynamic, and energy transfer parameters. For similarity to exist between two systems each of these parameters must be included.

Crowe and Lee [31] in their study of scaling laws for gas-particle flow through venturis discussed other parameters which must be considered in scaling.

The number of parameters necessary to scale a two-phase flow appear to rule out scaling of the "general case". It is possible that in special cases the influence of some parameters may be small, allowing the flow to be scaled using only a few parameters.[42]

The ability to scale valves for two-phase or three-phase flow appears to be exceedingly limited at the present time, since single-phase scaling is limited to about a factor of two.[1]

VII. INSTRUMENTATION

This section deals with the state of art in flow measurements and the corresponding instruments.

The measurement of flow can be quite critical in operating plants.A ccurate real time knowledge of flow could, in many circumstances, give sufficient warning of the state of flow, allowing ameliorative measures to be taken. The operator could also halt the process ahead of a failure and hence save on shut down time and equipment replacement expenses.

In the event of a failure the time history available in the form of measurements would provide a better insight into the situation prior to failure. Hence further repetition of such failures can be prevented.

The above mentioned are however the common needs for instrumentation in complex plants. In addition to these the nature of the fluid handled in coal plants necessitates special instrumentation.

From the engineering point of view slurry flow requires rugged instrumentation that can handle the slurry over considerable periods of time without resulting in loss of reliability or accuracy Hence designs which have intrusive probes are rather impractical.

From a theoretical point of view of the measurement technology a slurry generally deviates considerably from possessing such nice properties as isotropy and homogeneity. Hence the measurements which should ideally made locally in time and space are averaged instead.

VII.1 MEASUREMENT OF FLOW

The measurement of flow could be made in one of the two general methods [43].

VII.1.1 The Direct Methods

As the word *direct* implies these methods involve direct measurement of fluid velocity. As such they are the most attractive method. However the implementation of such measurements are often laborious and/or they do not give good accuracies. Thus these are generally used to calibrate the *indirect* instruments.

Some examples of direct measuring techniques are:

VII.1.1.1 Visualization (quantitative only)

VII.1.1.2 Isokinetic probes

VII.1.1.3 Sampling devices such as,'Quick closing valves'

VII.1.2 The Indirect Methods

These measure the physical properties most amenable to measurements and then use these properties to deduce the flow. If the property measured is l ,say, then;

$$l = l(\mathbf{x}, j_i)$$

Where

$\mathbf{x}$ is the state vector of other properties influencing l
j_i volumetric flux of the i^{th} component.

If $\mathbf{x}$ is a constant or the dependance of l on $\mathbf{x}$ is weak , then,

$$l = l(j_i)$$

Hence this is an inherent problem. Precise studies will include appropriate correction terms corresponding to $\mathbf{x}$. One common element in $\mathbf{x}$ is the temperature.

Almost all the instruments used in flow measurements use these indirect methods. The list of some of these common instruments given below will include both volume and mass flows.

This discussion will concentrate on two phase flows. However a crude extension could be made to multiphase flows in the following manner.

Let each of the indirect measurement measure a particular property $l^{(k)}$ which could be linearly approximated with the component quantity a_i, then;

$$l^{(k)} = \sum_{i=1}^{N} a_i . l_i^{(k)}$$

Where

N is the number of components in a general poly phase flow.

Hence by measuring N properties such that the resulting N component equations are linearly independent of one another, all the a_i can be evaluated.

However all $l^{(m)}$ ($m = 1$ to N) should be measured at the same cross section at the same time without any interference effects.

VII.1.2.1 Beam attenuation [43]

This method works on the degree of attenuation of x,γ rays and β particles. These beams attenuate as they travel through the medium. The degree of attenuation depends on the composition of the medium. Hence inversely by measuring the attenuation the medium composition can be estimated.

From Lambert's law;

$$\frac{dI}{I} = -\mu . dx$$

Where

I the local intensity of the beam
x the distance travelled by the beam

μ the linear absorption coefficient

Hence the beam is attenuated as it travels through the medium. Thus by measuring the beam intensities at the entrance and exit points, μ can be estimated. Knowing the properties of the components an equation can be obtained for the composition. For two phase flow this equation itself is sufficient to get the component composition.

This method would give valid results provided,

1. The two phase flow is homogeneous

2. Beam is well collimated so that all photons move in the same direction.

3. The beam is monoenergetic

4. The component distribution is normal to the incident beam.

But these conditions are seldom conformed to by the fluids used in the industry. However fluids coming close to these requirements may be used at a laboratory level.

VII.1.2.2 The acoustic flowmeter [44],[45]

One type of acoustic flowmeter utilizes attenuation in the manner of the previous section. The illustration below is based on a gas-solid particle flow. The basic relationship is, $\alpha_t = \alpha_g + \alpha_s + \alpha_v + \alpha_k$

Where

α_t The total acoustic attenuation
α_g The absorption component due to the gas
α_s The scattering component due to the particles
α_v The viscous component due to the relative motion between the gas and the solid
α_k The thermal component due to the irreversible heat transfer between the gas and the particles

A schematic of this is illustrated in Fig.VII.1.2.2

However in a bubbly flow situation there is acoustics generated due to the presence of bubbles which may have to be taken into consideration.

VII.1.2.3 The capacitive density measurement [44],[45]

This method works uses the dependance of the capacitance of the medium on its composition. Hence by measuring the capacitance of the medium the composition may be deduced. A schematic diagram of this is shown in Fig.VII.1.2.3.a. For flows in the range *dry* to *slurried* to *spray dried* the capacity measurement is more suited for moisture content measurement as opposed to density measurement.

The fractional change in density due to a volume increase of dielectric e_2 is given by: $dK = dC / [C_0(e_2 - e_1)]$

Where

C measured capacitance
C_0 vacuum capacity
K packing
$\varepsilon_{1,2}$ dielectric constants of materials 1,2

However capacitors are not suitable for the combination of high pressure and high temperature slurries.

The electrode configuration is shown in Fig.VII.1.2.3.b

VII.1.2.4 The optical probe [46]

Light beams may reflect or refract at a glass/fluid interface depending on the index of refraction of the fluid medium. This property is used in the optical probe. The uses of the optical probe are almost unlimited within the fluids commonly used in engineering practice. This is useful when the electrical methods cannot be used due to the components having similar electrical properties. A typical probe unit is illustrated in Figs.VII.1.2.4. a & b.

For a particular choice of geometry and transparent material the light will be reflected or refracted depending upon how the index of refraction, N , compares with the critical number, N^*. Hence if N^* is such that;

$N_g < N^* < N_l$ then, the gas phase will reflect the light whereas the liquid phase will refract it.

Where $N_{g,l}$ are indices of refraction of gas and liquid respectively.

Theoretically the output signal will be a non-uniform string of square waves. In practice, the waveform is not square. Hence one has to set up a voltage threshold and trigger constant voltages accordingly.

However extreme care must be taken as regards the selection of the threshold because the accuracy of the results heavily depend upon this. With well chosen thresholds and careful experimentation accuracies in the order of 2% may be expected.

VII.1.2.5 The acoustic Doppler flowmeter [47]

This flowmeter is essentially made up of a sonic wave transmitter which transmits sound waves into the flow and the receiver that picks up the scattered waves, Figs.VII.1.2.5. a & b. The picked up wave is shifted in frequency relative to the emitted wave. The overall frequency shift is given by;

$$\Delta F = F.(V/c).(\cos\Theta_1+\cos\Theta_2)$$

Where

c speed of sound
F frequency of transmitted signal
V flow velocity
ΔF frequency shift
Θ_1 angle between the flow and incident sound beam

Θ_2 angle between the flow and scattered sound beam

This technique is quite ideal for slurry flow measurements. Some of its features are:

1. Non intrusive flow measurement

2. $\Delta F \propto V$

3. Direct frequency output suitable for telemetry.

4. High precision obtainable provided response speed can be waived.

5. Simple electronics.

The received signal however is not a pure tone. Hence the bandwidth is characterized by the statistical distribution of the period of Doppler signal.

This gives the integration time T, required for any desired precision E, given the mean Doppler frequency F, and the relative standard deviation of the frequency $S = \sigma . \sqrt{F}$

$$T = S^2 / E^2 . F$$

A table containing some physical values is given in Table 1 (Fig.VII.1.2.5.c) The standard deviations are shown in Figs.VII.1.2.5. d & e.

VII.2 MEASUREMENT OF VISCOSITY

In the synfuel industry the measurement and monitoring of viscosity is quite important. This need arises from the fact that the viscosity changes by orders of magnitude for small changes in the composition of the slurry. These large changes directly affect the flows and pressure drops throughout the system and may result in plant breakdown.

VII.2.1 Acoustic Slurry Viscometer [48]

The acoustic attenuation of fluid medium is found to be a function of the fluid viscosity. Hence the acoustic attenuation could also be used to deduce the viscosity. The general layout of this measurement system is shown in Fig.VII.2.1

In some theoretical work done by Stokes it has been shown that the acoustic attenuation in a fluid medium is given by;

$$\alpha = \frac{2.\omega^2.\eta}{3.\rho.c^3}$$

Where

ω wave frequency in rad./s.
η shear viscosity
ρ density
c velocity of sound in the medium and is given by

$$c = \sqrt{\frac{\gamma.K}{\rho}}$$

Where

γ ratio of specific heats
K the bulk modulus of elasticity.

However the attenuation expression is valid only for cases where the product of the relaxation time and frequency is very much smaller than unity. That is;

$$\omega.\tau << 1$$

Where

ω angular frequency
τ the relaxation time

τ is given by;

$$\tau = \frac{4.\eta}{3.\rho.c^2}$$

In most slurry situations a combination of high ω and high η is quite rare. Hence the theory holds well.

However the readings are quite sensitive to particulate loading and the back-scattering measurement is used to calculate the particle fraction and hence a correction can be made to the viscosity.

VII.2.2 Capillary Tube Viscometer [49]

This method is one which can be used to deduce shear stress and shear rate. Hence it can be used either for a Newtonian or for a non-Newtonian fluid.

The principle of this viscometer is the measurement of pressure drop across a long smooth tube of known diameter as the fluid flows through the tube in laminar flow at a known rate.

From Rabinowitsch - Mooney analysis

$$\tau_\omega = \frac{D.\Delta p}{4.L}$$

Where

D internal diameter of capillary tube.
Δp pressure drop across the whole tube
L Length of the capillary tube.

And the shear rate $\dot{S}_\omega$ is given by:

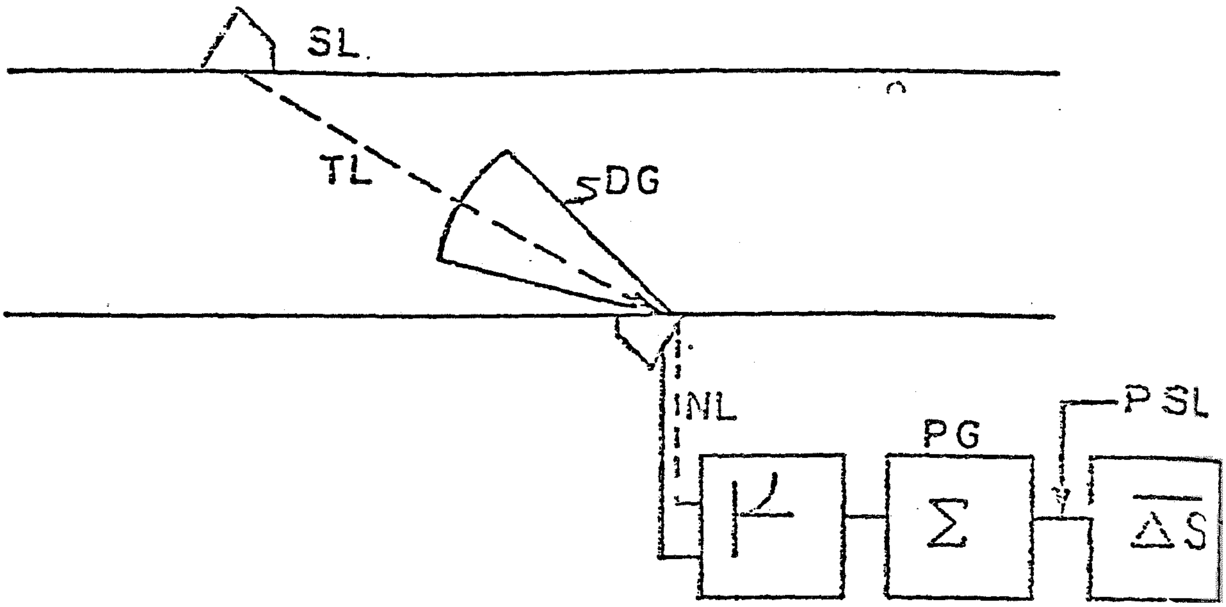
$$\dot{S}_\omega = \left[\frac{8.V}{D}\right]\left[\frac{3.\eta'+1}{4.\eta'}\right]$$

Where

V superficial velocity of the fluid
η' local slope given by $\eta = \dfrac{\partial \tau_\omega}{\partial\left[\dfrac{8.V}{D}\right]}$
and all other parameters as defined before.

In order to measure η' a flow curve which is a logarithmic plot of τ_ω Vs. $\dfrac{8.V}{D}$ is constructed.

However the pressure measurements in such situations are quite inaccurate.



SL, Source level at inner wall

TL, Transmission loss through fluid

DG, Receiving transducer directional gain

NL, Background noise level

PG, Processing gain

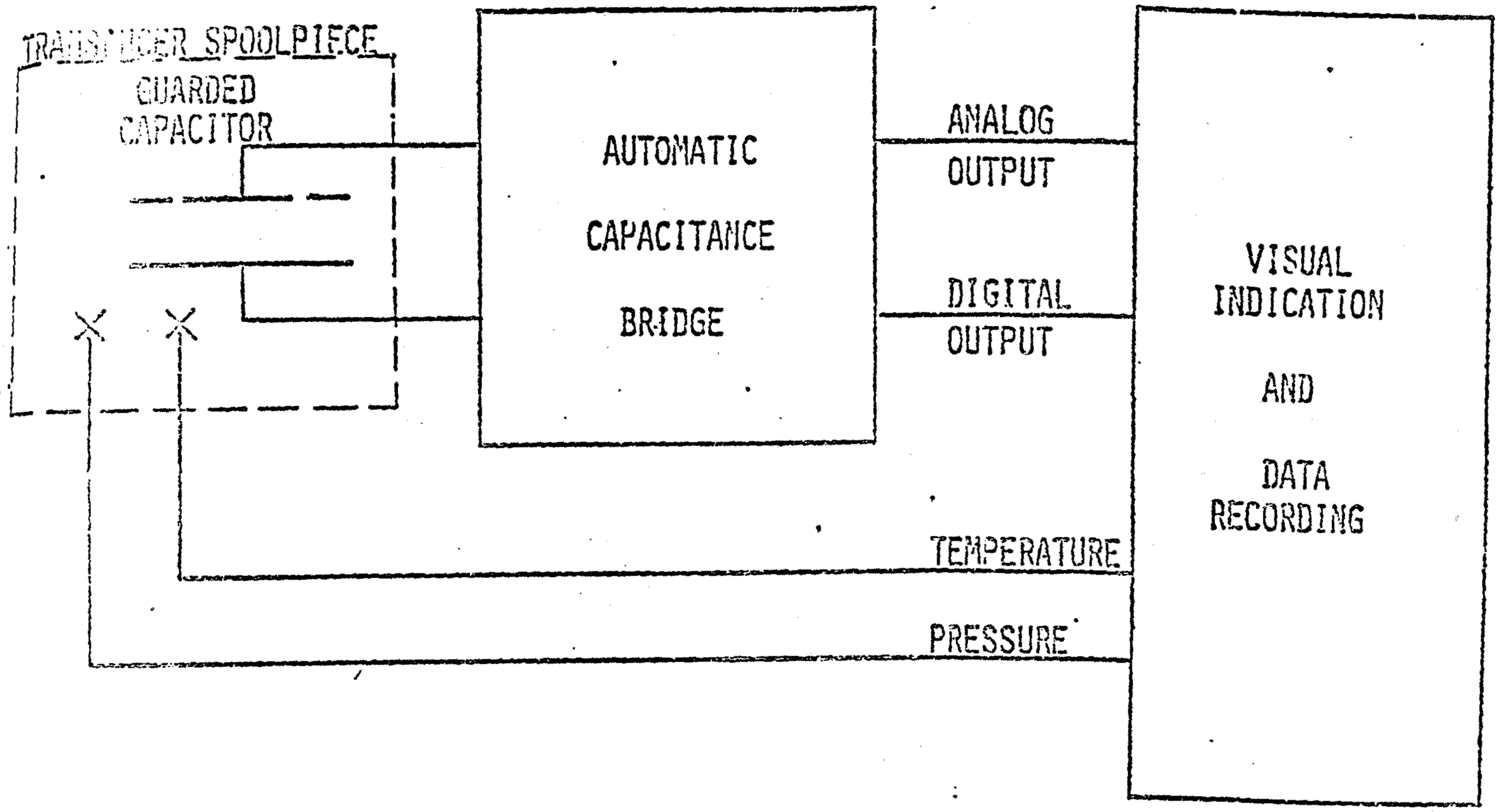
PSL, Processed signal level

ΔS, Averaged measurement parameter
(Δf, $\Delta\phi$, Δt, etc)

SLD, Square law detector

*, Summation

Fig VII.1.2.2 [45]



Capacitive Density Measurement System

Fig VII.1.2.3 a [45]

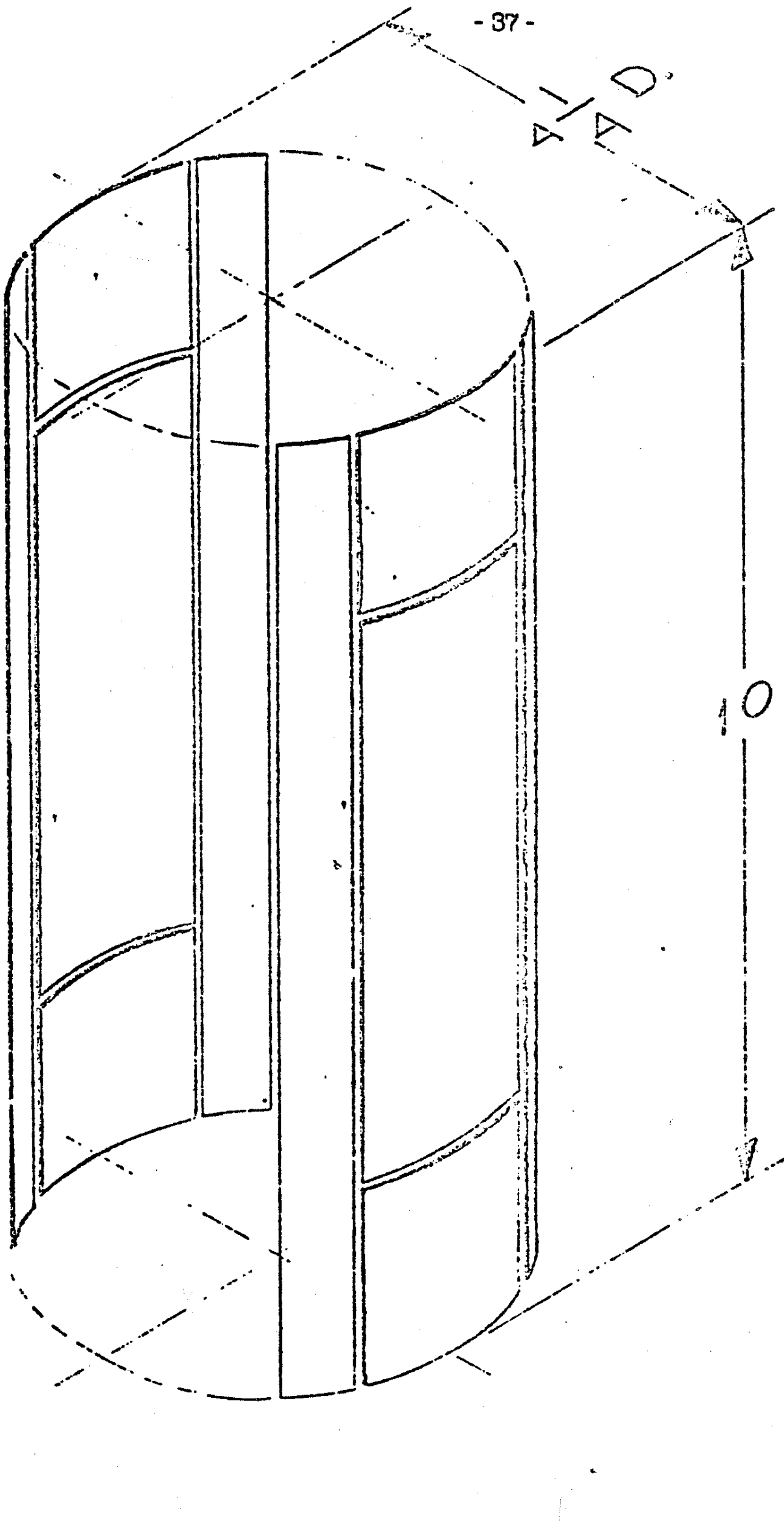


Fig VII.1.2.3 b [45]

Cylindrical Capacitor Electrode Concept

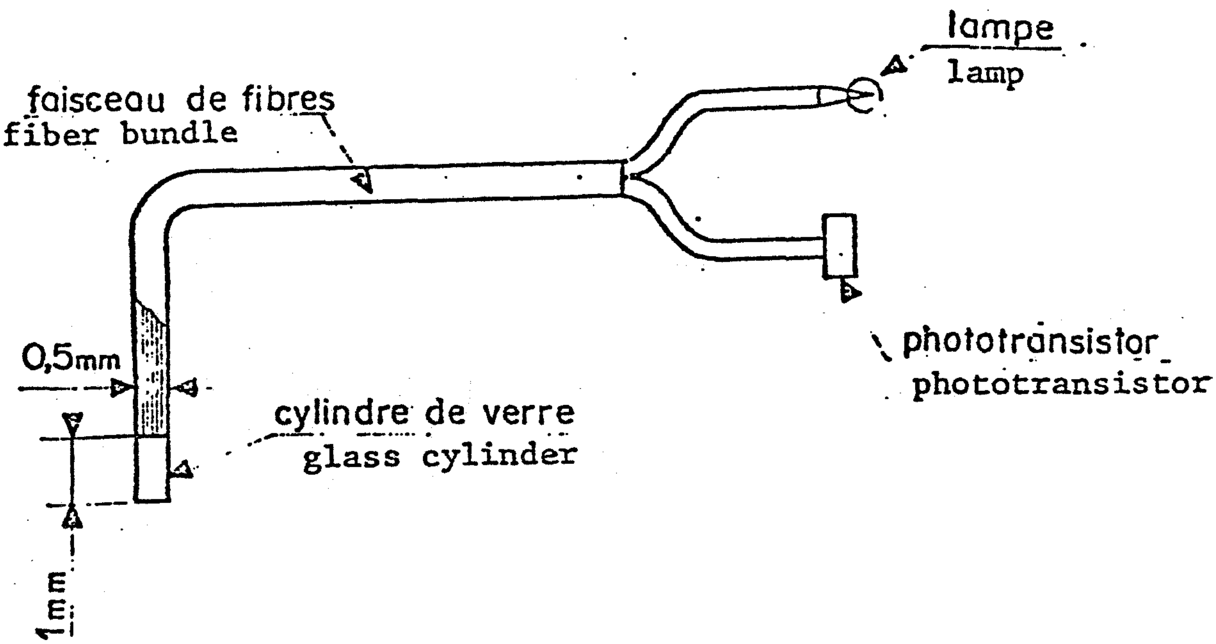


Fig VII.1.2.4 a
The fiber bundle system (Hinata). [46]
Le système à faisceau de fibres (HINATA)

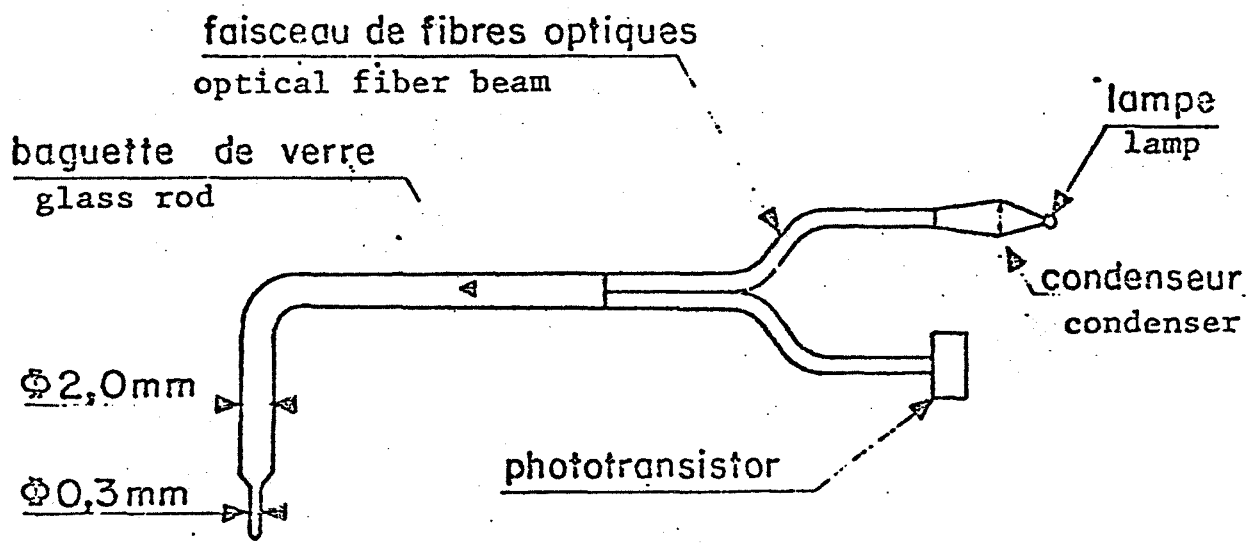


Fig VII.1.2.4 b
Le système à tige de verre (MILLER et MITCHIE)
): The glass rod system (Miller and Mitchie). [46]

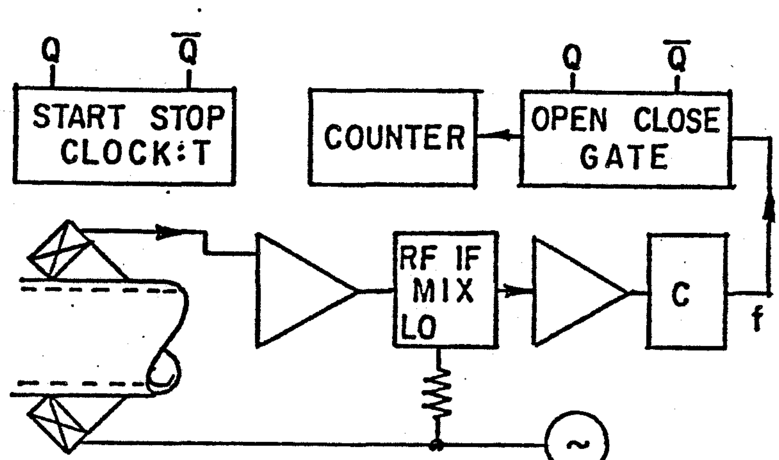


Fig VII.1.25 a Block Diagram for the Doppler Flowmeter [47]

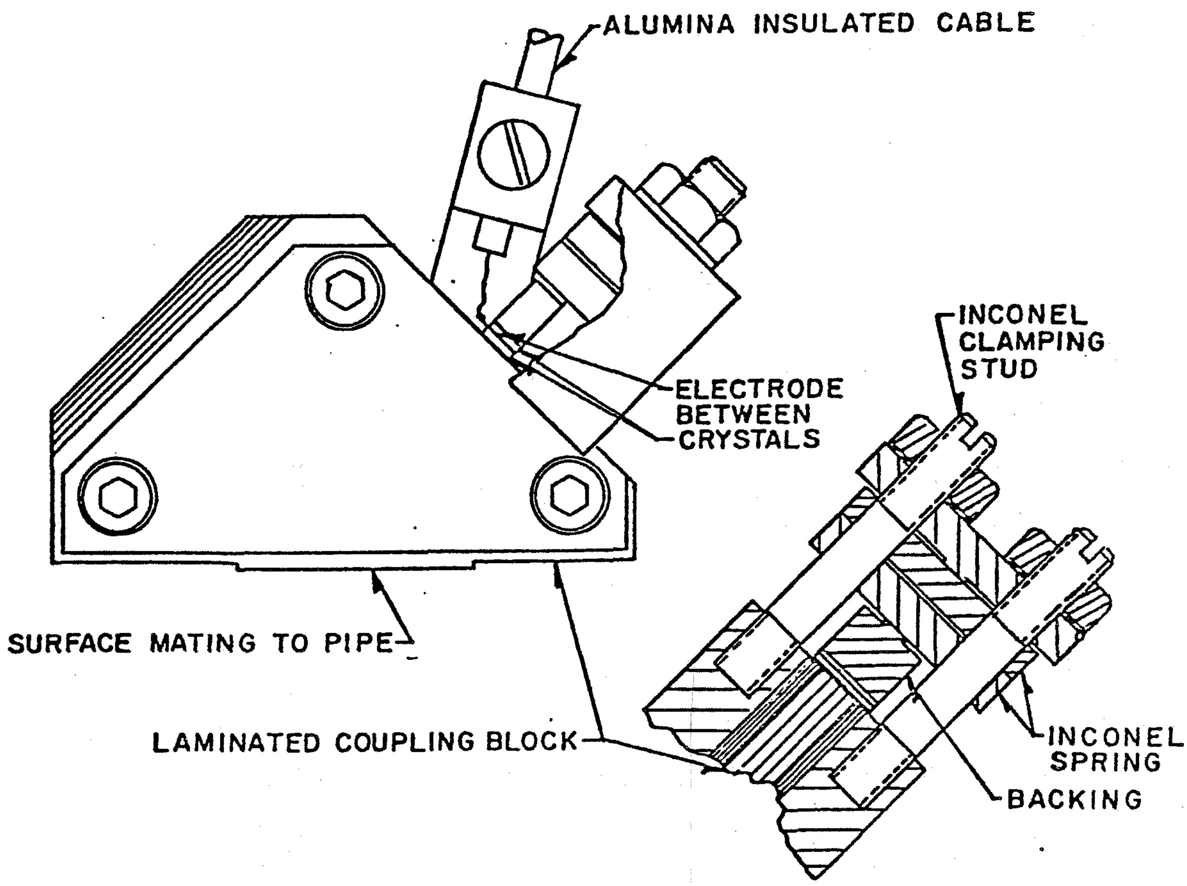


Fig VII.1.25 b Ultrasonic Transducer on a Laminated Coupling Block [47]

TABLE 1

$T = S_1^2/E^2F \qquad S_1^2 = 0.1$

E %	COUNT RATE, F		
	1000Hz	200Hz	100Hz
10	.01	.05	0.1
1	1	5	10
.1	100	500	1000

Integration Time, Seconds

Fig VII.1.2.5 c

[47]

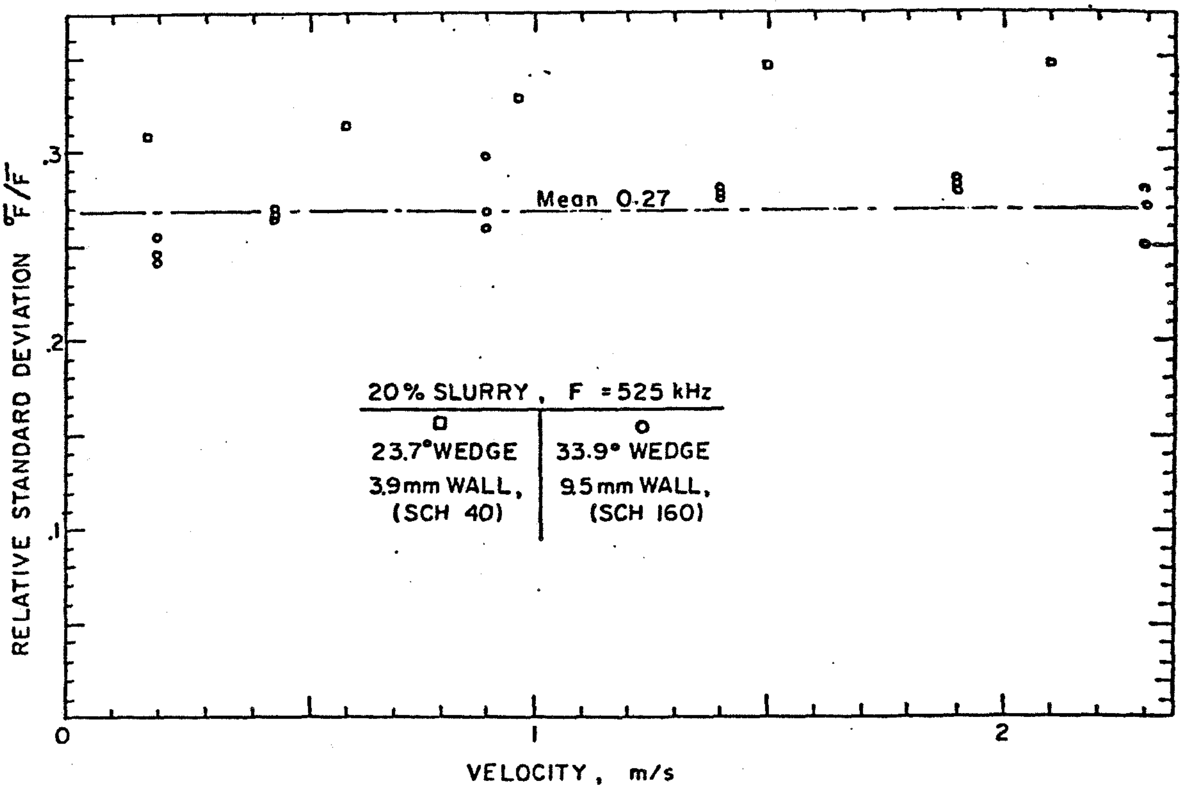


Fig VII.1.2.5 d Relative Standard Deviation of Doppler Frequency [47]

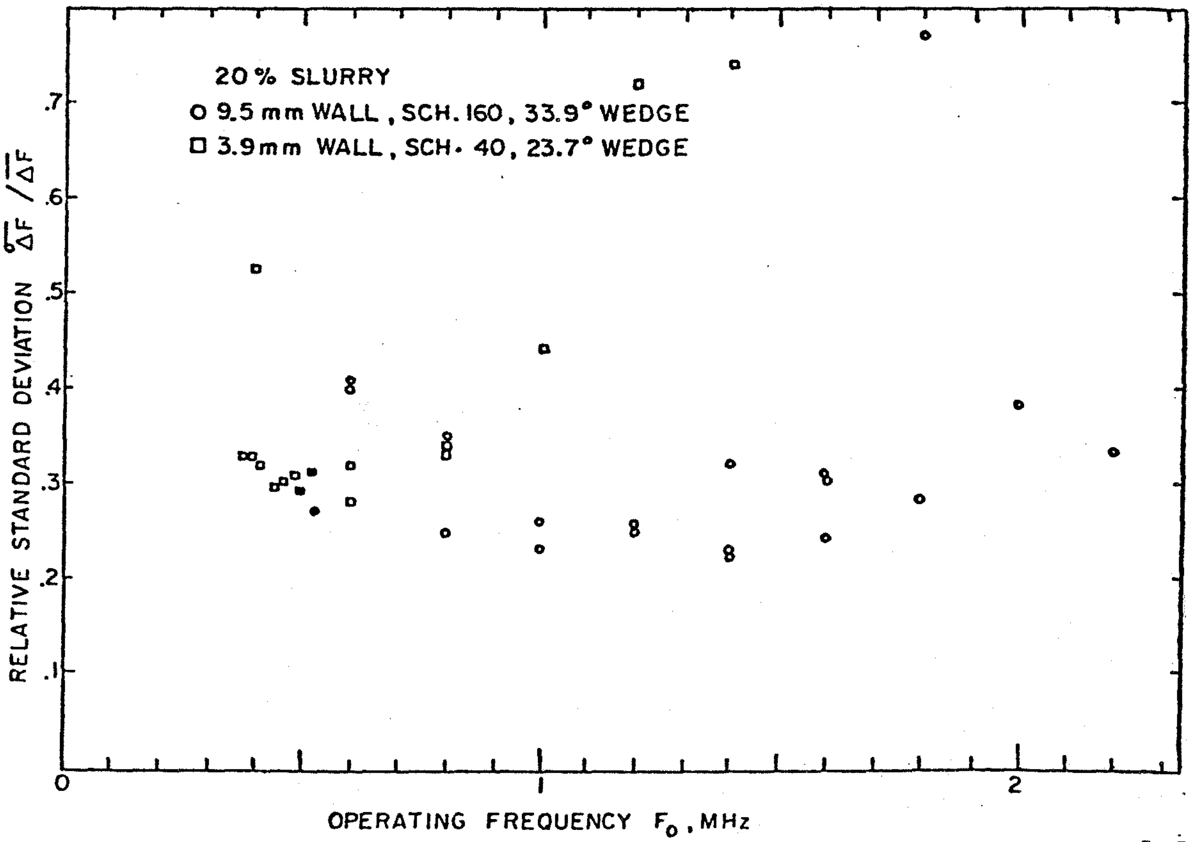


Fig VII.1.2.5 e Relative Standard Deviation of Doppler Frequency [47]

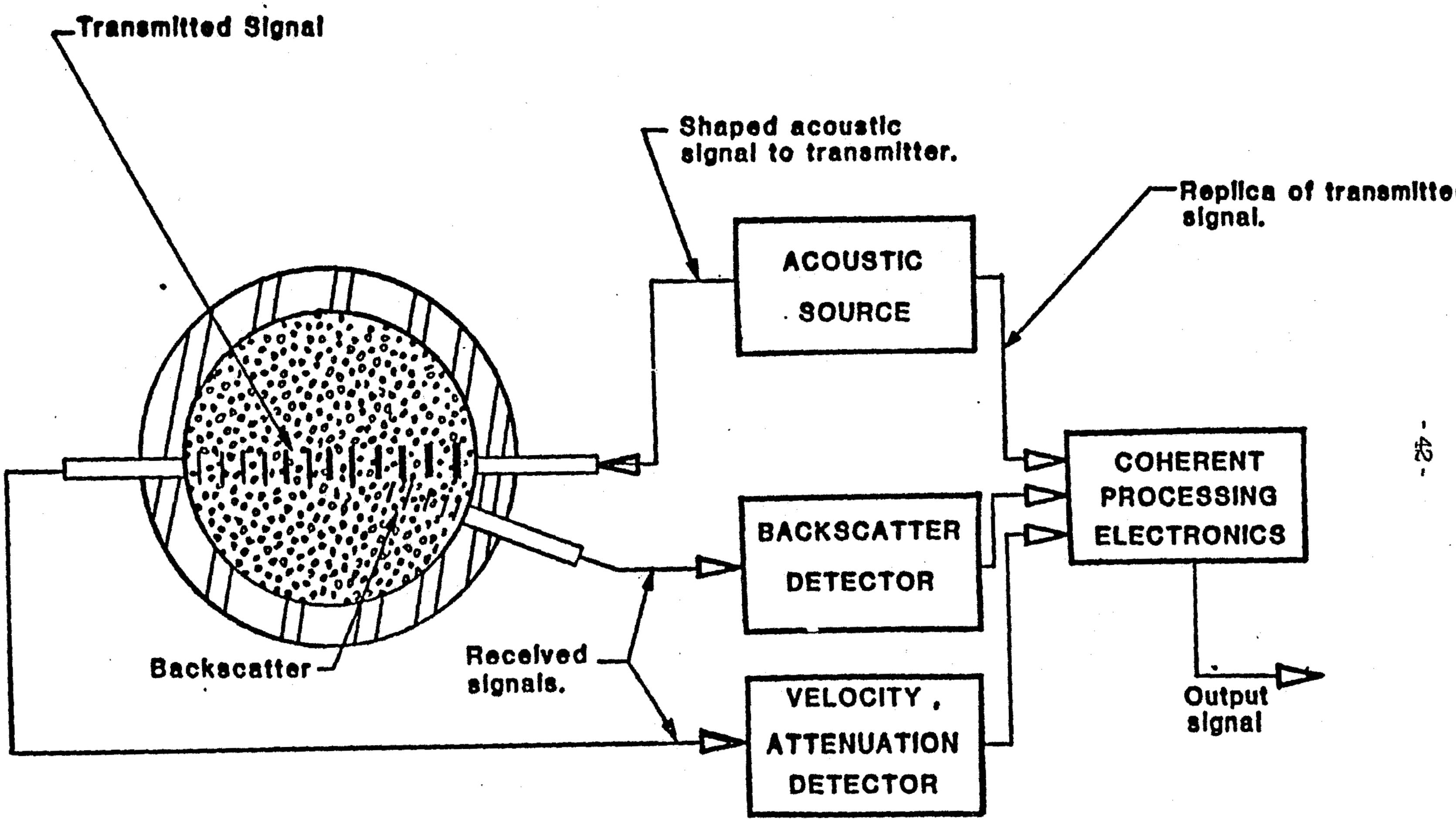


Fig VII.2.1 SLURRY VISCOMETER CONCEPT [48]

VIII. SUMMARY AND CONCLUSIONS

High-pressure letdown of coal/ash-solvent slurry required in the coal conversion process has proved to be a formidable problem. The severity of the conditions involved in these letdown process is unequaled elsewhere in the process industry.[5]

Experience in the pilot plants indicate erosion damage is a very serious problem. Attempts to operate in the highly erosive environment have been based on one of two strategies; providing sufficient material to allow sacrificial loss, or the use of hard materials, and streamlining to prevent erosion. To date none of these attempts have been completely successful.

An examination of the letdown devices used in the existing pilot plants indicated that most valves in use are modified versions of commercially available equipment. The most common types being the angle valve, and the oil field choke valve. No "active" pressure letdown systems, capable of recovering energy from the letdown stream, were found to be in use. Furthermore, surveys of the process industry indicate no equipment is presently available which would allow such energy recovery.

Upon examination of the various valves used in the process industry, it was found that these devices could be classed into generic groups describing their principle method of energy dissipation. Valves used in the process industry for high pressure letdown were found to occupy five generic groups, while the valves used in the coal conversion pilot plants may be classed in only two of these groups; jet formation devices, and choked flow devices. Valves operating on the principles of vortex formation, skin friction, or tortuous path have not been employed in the pilot plants. Valves using fluidic control principles, such as the vortex valve, should certainly be investigated for applications in the coal conversion processes.

Solutions of these problems will require novel and inovative approaches to both the material and hydraulic aspects of letdown valve design. Greater attention should be paid to the utilization of both vortex control and ckoking in passive devices. Though active devices could be developed for some processes in the long run, the most immediate need is for careful study of the design of passive devices. However is is possible that some of the "discarded" energy could be utilized to effect at least part of the phase separation process which usually follows the letdown process. Valves which not only allow pressure reduction but also effect seperation of at least one of the phases should be explored.

An examination of the control valve sizing methods available to the process industry, indicated the sizing techniques are based on an elementary analogy between a control valve and an orifice. Sizing techniques seem to be adequate when operation is in the range where the fluid is single-phase and the flow is fully turbulent. Techniques for conditions other than these generally rely on use of a 'homogeneous" flow model. The validity of these 'homogeneous" models is not universal, and often tends to be inaccurate due to high relative velocities between the phases when high pressure drops are encountered.

The concept of two-phase "critical" flow has received little attention in the valve industry. This concept is essential to the understanding of phenomena such as "cavitation" and "choking" in valves. Substantial work in this area has been done in the nuclear industry in connection with reactor blowdown during a loss-of-coolant-accident and could be used in the valve industry.

The lack of ability to scale two- and three-phase fluid dynamic systems cannot be overemphasized. The determination of proper scaling methods is vital to the design of industrial components if "trial and error" methods are to be avoided. The development of these scaling methods is made particularly difficult, though, by the fundamental lack of knowledge of the hydrodymanics of two- and three-phase flows.

Development of multi-phase flow instrumentation is necessary for both basic studies of fluid flow and process control. Effective system control requires the knowledge of component flow rates and properties. Several methods are available to measure component flow rates in multi-phase flow although, none seem to be suited for making measurements in slurries. The ruggedness required and other complicating factors such as low signal-to-noise ratios are limiting factors in available equipment. Equipment necessary for on-line viscosity measurement in coal/ash-solvent slurries is not presently available, hence development of new technology is needed in this area.

In conclusion, the problems associated with high-pressure slurry letdown in the coal conversion processes must, for the present, remain at least partially unresolved. The reliability of these letdown systems may be improved by research into harder materials which resist erosion, but the ultimate solution of this problem will require a better understanding of the flow phenomena involved. The knowledge acquired in a comprehensive investigation of these flow phenomena should also provide the basis for predicting system performance and scaling systems.

Table VIII.1 contains a listing of references used in the preparation of this report classified by topic. References are listed in three groups; Primary References, References, and Additional References. The articles listed under "Additional References" are, in many cases, of a more fundamental nature. Although much of the information in these articles is not directed at the construction of letdown devices, consideration of the fundamental behavior of these flows is an essential ingredient in the design of a successful letdown device.

TABLE VIII.1

REFERENCES BY TOPIC

Operating Conditions/ Pilot Plant Experience	
Primary References	[1] [3] [7] [12]
References	[2] [4] [6] [9] [10] {3} {4} {5} {12}
Additional References	[36]
State-Of-The-Art Letdown Devices	
Primary Refereneces	[3]
References	[1] [5] [8] [13] [15] [16] [17] [50] {3} {4}
Additional References	[9] [36]
Control Valve Sizing Methods	
Primary Refereneces	[11] [17] [20] [25]
References	[14] [18] [21] [22] [29] [31] [32] [33] {1} {6} {7} {13}
Additional References	[19] [23] [24] [26] [27] [28] [30] [34] [35] [42]
Active Letdown Devices	
Primary Refereneces	[9] [37]
References	[39] [40] [41] {3} {4}
Additional References	[36] [38]
Scaling	
Primary Refereneces	[42]
References	{8} {10} {11}
Additional References	{13}
Instrumentation	
Primary Refereneces	[44] [45]
References	[46] [47] [48] [49] {2}
Additional References	[43]

[ ] -- Cited References { } -- Bibliographical References

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Addenda

This is an addenda to Report N.E2005 entitled "Synfuel Systems: A Review of Letdown Devices and other Multiphase-Flow Problems" by A.L. Charles, K. Chelvakumar, C.E. Brennen and A.J. Acosta, California Institute of Technology, Pasadena, California, 91125.

The essence of this report has been to highlight that there are problems associated with letdown devices and other multiphase flow phenomena in synfuel systems and that there is a lack of the fundamental knowledge which could be utilized to solve these problems. It seems clear that this difficult combination necessitates the instigation of a program of technological development and basic research. It seems to us that the Department of Energy in cooperation with universities and research laboratories should develop a program directed toward providing basic research information which could be used in the technological development. Some of the components of such a program are identified in this report. It is our judgement that the existing state of knowledge of two and three phase flows particularly in letdown devices is substantially inadequate for future needs. Therefore a program of <u>at least</u> $5 million per annum would be necessary to provide any really applicable information in the time frame required by the developers.