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Gas removal systems Part I: Literature survey and formulation of test program, June 1964

W. P. Isaacs

B. HerbichJ.

LEHIGH UNIVERSITY INSTITUTE OF RESEARCH

Dredge Pump Research

GAS REMOVAL SYSTEMS
PART I: LITERATURE SURVEY AND FORMULATION OF TEST PROGRAM

by

John B. Herbich
W. P. Isaacs

Fritz Engineering Laboratory Report No. 310.3

CIVIL ENGINEERING DEPARTMENT
FRITZ ENGINEERING LABORATORY

GAS REMOVAL SYSTEMS

PART I: LITERATURE SURVEY AND FORMULATION OF TEST PROGRAM

Prepared by
John B. Herbich
and
W. P. Isaacs

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ABSTRACT

A review of the literature was made on any similar or related work previously performed on gas removal systems associated with dredge pumps. This review is presented in the form of abstracts and annotations. The review interprets, to the extent practical, all such findings in relation to the dredging process.

A proposed test program is formulated and discussed for carrying out a study of gas removal from dredging suction lines in an experimental study.

P R E F A C E

The following report summarizes the studies performed between January 1, 1964 and June 30, 1964, at the Hydraulics Division of Fritz Engineering Laboratory, under terms of Contract No. DA-109-CIVENG-64-72. The progress on the study was reported in two status reports dated February 1964 and April 1964 (Fritz Laboratory Reports No. 310.1 and No. 310.2 respectively).

The project was directed by Dr. John B. Herbich who was assisted by Professor W. P. Isaacs, Dr. A. W. Brune, Mr. Hugh Murphy, Research Assistant, Mr. Brian VanWeele, Research Assistant and Mr. P. Aarne Vesilind, Research Assistant. Manuscript preparation was performed by Miss Rosalie Fischer. Professor W. J. Eney is the head of Civil Engineering Department and Fritz Laboratory. Dr. L. S. Beedle is the director of Fritz Laboratory.

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I INTRODUCTION

A. General

The modern sea-going hopper dredge is the result of progressive development which has taken place during the past century. It has found increasing importance in improving harbors and seaways both in peace time and during war. The hopper dredges in the United States are of the hydraulic suction type, equipped with special machinery which enables them to dredge the material from the ocean bed or channel bottom, to discharge it into hoppers, to transport it and to dump it at disposal sites.

The heart of the dredge is the pump. The pump is of the centrifugal radial type and must be designed to withstand heavy wear and abrasion. While in operation, it may encounter a variety of mixtures made up of liquids, solids, and gases. No particular difficulty is encountered when the mixture is composed of solids and liquids, except when the density of the mixture becomes too high. This may occur if the drag is buried too deep in mud at the channel bottom, causing the pump to choke. This condition may be remedied by lifting the drag arm.

When material containing a considerable amount of gas is encountered, the gas drawn into the suction line causes appreciable decrease in vacuum and volume of solids discharged. The remedy for this may be identical to that of choking, and for many years the difference between choking and unloading due to gas was not recognized.

In recent years, however, gas removal equipment has been installed on a number of dredges and some improvement of efficiency was noted. These systems are not totally effective and better methods must be developed to both increase the amount of gases removed and lower the cost of such an operation.

As a result, the U. S. Army Corps of Engineers entered into a contract with Hydraulics Division of Lehigh University to conduct a study to learn more about the mechanics of the problem and recommend a more efficient method of gas removal.

The study was divided into four phases:

Phase A- 1. Literature Search
2. Formulation of Test Program

Phase B- 1. Formulation of Specific Test Setups
2. Establishment of Test Schedule

Phase C- 1. Establishment of Test Setup
2. Performance of Tests with Water only

Phase D- 1. Performance of Tests with Solids-Water Mixtures
2. Tests with Physical Equipment Alterations

The current report describes the work performed under Phase A of the contract.

B. Present Gas Removal Systems

Gas samples taken from dredged material indicate that the most soluble composition of the gas may be approximately 85% methane and 15% carbon dioxide.

The present gas removal systems are all basically alike. Accumulators are installed near the pump or at the highest point on the suction line and the gas is ejected from the accumulators either by vacuum pumps or water or steam ejectors. The accumulation of the gas is possible because most of it rises to the top of the suction pipe as it is being brought up from the bottom. The accumulators are about six foot high and have diameters approximateing those of the suction lines.

II. LITERATURE SEARCH

An extensive literature search was conducted as the initial phase of this project. The results of this search are presented in two parts. The first part is concerned with application of the findings to the actual dredging process. The second part summarizes the literature which may be of value to the proposed model test facility.

A. Application to Dredging Process

1. Two Phase Flow

When the dredge is pumping mixtures composed of solids, liquids, and gases, a complex relation exists between the velocity of the mixture and the friction losses encountered. This situation would have to be analyzed as a three phase flow problem. Unfortunately, no literature was found that attempted to analyze this condition. Extensive literature, however, is available on two-phase flow, mostly that of gas and liquid. If the solid-liquid mixture being pumped can be assumed to act like a liquid, the two-phase flow analysis can be of value in determining some essential parameters such as friction factor and Net Positive Suction Head (NPSH).

Of all the various methods used in the analysis of two-phase flow, probably the most applicable is the Martinelli method[12, 13, 42]. In this method, the flow is defined as a function of two variables, X and Φ, and the pressure losses can be readily calculated. When the pressure losses are known in a drag pipe line, the NPSH can also be calculated and is of direct concern in actual dredging practice.

Other attempts have been made at analyzing two-phase flow, but these have not been accepted as widely due to the questionable accuracy and the difficulty of application. Most of these methods use the "homogeneous flow" approach, attempting to use one friction factor which will describe both the gas and liquid phases simultaneously.

2. Accumulators

The literature search did not yield much information on gas accumulators. What information was found, is not directly applicable to the dredging process but will require further investigation before their mertis can be evaluated. These possibilities will be studied in the Phase B of the project.

3. Vacuum Pumps and Ejectors

A through description of ejector performance can be found in the Heat Exchange Institute Standards for Steam Ejectors[53]. Performance data for steam ejectors is also available in paper by Fondrk[6]. Both of these references may be used to improve the performance of the steam ejectors presently in use on various dredges.

A new design has been developed for vacuum pumps which decreases vibrations and increases the efficiency[39]. On dredges using vacuum pumps for gas evacuation this new pump may be of value.

Since there also exists a problem of cleaning the gas before it is introduced into the vacuum pump, wire mesh separators are discussed in Reference[46].

B. Application to Model Test Facility

The literature search also has provided information which will be of value in the experimental work. The model facility will geometrically duplicate the prototype to 1:8 scale. All conditions experienced in actual dredging will also be duplicated as much as possible. Experiments will then be performed using various methods of gas removal.

1. Introduction Gas Bubbles

One of the existing condition in actual dredging practice is the presence of gases in the suction line. This will have to be duplicated in the laboratory. From literature, it was found that gas was injected into test sections usually in either one of two ways. The first required the use of an aspirator[22] and the gas was injected parallel to the flow and in the middle of the pipe. Other investigators found that injecting air vertically from the top of the pipe resulted in a good distribution of bubbles[13]. Both methods seem to be acceptable but the latter is probably less expensive.

2. Method of Measuring Flow Rates

The air flow in the majority of investigations was measured with a small orifice or nozzle flow meter. These meters were connected as near the point of injection as possible.

The flow rate of liquid in the majority of cases was measured with a Venturi type flow meter. In some cases, this meter was installed in the pipe line upstream from the point of gas injection. Other investigators used separators to withdraw the gas from the liquid downstream from the test section and measured the liquid flow rate at the point.

Visual observations have also been conducted on two phase flow. Most of these observations were conducted on the transparent test section and were used solely to describe the different flow patterns. These patterns are described in many reports, the initial work being performed by Alves[24]. One study, however, used visual observations to gather quantitative data[22]. In this investigation, a small amount of the sample was diverted into a transparent vertical slot and the images of the bubbles projected on a screen. This provided a bubble size distribution and also a rough check on the relative amount of gas and liquid.

3. Gas Accumulators

As was noted earlier, very little literature is available on gas accumulators. Two ideas, however, have been advanced which might be worth considering.

The first idea involves the use of the 90 degree bend in the suction line. Since liquid has a greater specific gravity than gas, it would tend to cling to the outside wall of the bend, leaving an air pocket on the inside. If the flow, however, is too rapid and turbulent, most of the gas will not be able to reach the air pocket. As a result, a proposal was made to install guide vanes inside the elbow, thus producing air pockets on the concave side of all the vanes[21]. The gas could then be drawn off by providing escape routes for the gas through the vanes. Unfortunately, this idea was not followed through, and no other reference has been found on this subject.

The second possible gas accumulation device involves the use of vortex separators[17, 7, 50, 3]. These separators were developed mainly for use in the paper manufacturing process and are used to remove both gas and grit from the wood pulp. They work on the principle of centrifugal force. The dirty pulp is pumped tangentially into a vertical cylinder. The higher density of the grit forces it to the outside. The gas will form a core in the middle of the cylinder and is drawn off by vacuum pumps.

Even though these vortex separators might not be directly applicable, the principle used might be of significant value in developing an acceptable accumulator.

4. Ejectors and Vacuum Pumps

The information available on ejectors and vacuum pumps was discussed in an earlier paragraph. This information will be used in determining a suitable evacuation system for the model facility.

III. FORMULATION OF TEST PROGRAM

A. Discussion

A testing program for the study of gas removal from the suction side of a dredging pump is formulated here. It is of general form and should be considered quite fexible as to how actual operations will be carried out as the project progresses.

Gases occuring in dredged materials occur in two distinct forms: (1) Free gas which is usually trapped due to liquid viscosity effect and which enter the pump suction line without prior opportunity to escape to the atmosphere, and, (2) Dissolved gases which are in solution in the liquid medium but which come out of solution into the "free-form" as the suction pressure is decreased below normal atmospheric pressure. Of course the higher the vacuum pressure or the lower the absolute pressure becomes in the suction line, the more of the dissolved gases that will come out of the solution.

It is envisioned that the first test series will be one to make general observations of the behavior of gas in the suction line system. Although the optimum location of a gas accumulator is to be part of this study, the initial location of the accumulator for the series of observation tests will be set as close to the suction side of the pump as possible. It is given this initial location because the vacuum pressure will be greatest near the suction side of the pump.

In this first series of tests, two types of gas injection or occurence will be observed. It is believed that the initial tests should be one of observing the system in operation handling a water mixture that has been previously saturated with gas under a hydrostatic head

(At least near saturated with gas). It is believed that dissolved gases will be removed easier with the accumulator in the previously cited initial location.

Secondly, a controlled rate of injected gas into the suction line will be observed. It is though possibly that the removal of this type of gas will be better accomplished closer to the open end of the suction line. This type might well be removed by a gas bleeder of some type. It would be advantageous to remove this form of gas at the earliest point possible in the suction system. An elbow with guide entry vanes on the inside radius might well be a possible solution for its removal since the gas should accumulate in pockets along the inside radius due to centrifugal forces.

Differences in behavior of the two distinct types of gas occurence will be noted in the initial observation tests and possibly a tr al run using both a saturated water mixture and a controlled rate of injected gas simultaneously can be observed. These observation tests should aid considerably in determining a workable model test set-up.

B. Test Set-Up.

The following initial model test set-up is presented in general form:

(1) Dredge pump, transparent

(2) Suction piping, transparent

(3) Accumulator, transparent

(4) Vacuum pump

(5) Vacuum control equipment

(6) Discharge piping, transparent

(7) Volumetric tank, with suction piping leading from and discharge

piping leading to. One side of the tank to be transparent.

(8) Measuring equipment ot measure dredge pump vacuum, pressure, rpm, velocities. Pressure also to be measured at intervals along entire length of suction piping.

(9) Gas, carbon dioxide, including injection, mixing and control equipment.

C. Test Series No. 1.

The object of this series of tests shall be to make general observations of the gas in the suction system when pumping clear water only. It should serve as a means for deducing potentially effective locations for the gas accumulator and also possible gas bleeders of a usable type. It will also provide a reference for comparison of performance of systems with and without gas removal equipment.

(1) Accumulator, vacuum pump and control equipment are not used in this series.

(2) Operate the dredge pump.

(3) Either inject gas into suction line at a controlled rate or using previously prepared water saturated with gas in volumetric tank, maintain predetermined pressures, simulating pressures at bottom of river channel.

(4) Observe rise, expansion, position and general behavior of gas in suction system.

(5) Note gage reading of control parameters, volts, amps, rpm, velocity, etc.

(6) Repeat at various rpm, holding injection gas constant.

(7) Repeat program for several values of gas content.

(8) Note amounts of gas which cause complete collapse of dredge

pump suction capability at each value of rpm, or, vice versa, rpm at which collapse occurs for each value of injection.

(9) From analysis of above data, determine several potentially effective locations of gas accumulators and gas bleeder mechanisms.

D. <u>Test Series No. 2</u>.

The object of this test series shall be to verify by test the optimum locations for the gas accumulator as indicated by observation in Test Series No. 1.

(1) Place the accumulator in the suction system at the first of several predetermined locations, and connect to the vacuum pump.

(2) Place the dredge in operation.

(3) Place the vacuum pump in operation.

(4) Exercise vacuum control so as to prevent solids or liquid carryover into vacuum pump and to regulate height of fluid in the accumulator.

(5) Repeat program of test series No. 1 but with accumulator in operation. Make similar observations and recordings.

(6) Repeat above program for other locations of the accumulator.

(7) From analysis of above data, deduce the optimum location of the gas accumulator.

E. <u>Test Series No. 3</u>.

The object of this test series shall be to determine the effect of various parameters on the performance of gas removal systems, with the accumulator located as determined from test series No. 2.

(1) Repeat the program of test series No. 2, varying all parameters

throughout their ranges, but only one at a time, to isolate the effect of each. Parameters to be studies shall include:

Gas content

Dredging depth

Density

Velocity

Discharge head

Gas removal pump vacuum

Gas accumulator fluid level

(2) Plot families of curves noting the interrelationship between parameters, as may be indicated to be worthy of observation as the tests progress, e. g., effect of gas volume at several values of density, gas removal pump vacuum, and gas accumulator level. Data recorded, in addition to the above parameters shall include dredge pump rpm, volts, amperes, vacuum, and discharge pressure.

A P P E N D I X A

A B S T R A C T S

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Baker, O. "Simultaneous Flow of Oil and Gas" The Oil and Gas Journal, July 26, 1954, p. 185.

Various flow patterns resulting from different gas-liquid ratios are discussed and a plot showing these regions is presented. The flow patterns and plot are the same as those in the paper by James and Silberman. (Reference 11, Abstract A).

Very long pipelines were fitted with meter apparatus. The gas-liquid ratios were varied and pressure measurements taken. The pressure drops along the pipeline were assumed to correspond to the following empirical relationships;

For gas only: $$\Delta P_G = \frac{Q^2 L' f S T Z}{20{,}000 D^5 \text{ Pave.}} \quad (1)$$

For liquid only: $$\Delta P_L = \frac{f L' (\text{Bbl/day})^2 (\text{lb/gal})}{181{,}916 \quad D^5} \quad (2)$$

where: Q = Gas flow rate ft^3/day x 1000

L' = Length of pipeline in feet

S = Specific Gravity of gas (air = 1)

T = $^oF + 460$

Z = Compressibility factor of gas

D = Inside Diameter

Pave= Average inlet and outlet pressures of the pipeline, psia.

f = friction factor, presented in table form vs. the Reynolds Number.

In two phase flow, the pressure drop is called ΔP_{TP}, and the following relationships are established:

$$\Delta P_{TP} = \Delta P_G \phi_G^{\ 2} \quad (3)$$

and

$$\Delta P_{TP} = \Delta P_L \; \phi_L^{\;2} \tag{4}$$

A factor X can then be defined as;

$$\phi_G = X \; \phi_L \tag{5}$$

or

$$X = \sqrt{\frac{\Delta P_G}{\Delta P_L}} \tag{6}$$

For purposes of uniformity, all tests were carried out in the turbulent flow regime ($R_N > 1000$) and for these conditions ϕ_G was defined as $\phi_{G\pi}$, or ϕ_G where both the gas and liquid are in the turbulent state.

Data was taken and plots were prepared for $\phi_{G\Pi}$ vs. X for various phases of two stage flow. From these plots, equations were obtained relating $\phi_{G\Pi}$ and X. In some of these equations the diameter of the pipe is a variable and in others, the mass velocity of the liquid phase is considered significant and is included in the equation.

The author states, however, that these equations were derived for hydrocarbon systems and should be used with caution for dissimilar cases.

In the form of a conclusion, the author presents some design criteria and economic considerations for the design of pipelines carrying both oil and gas.

Bergelin, O. D. "FLOW OF GAS-LIQUID MIXTURES" Chemical Engineering, May 1949, p. 105

The author reviews the progress in the analysis of gas-liquid mixtures. The various flow regimes are named. The Φ - X correlation of Martinelli is reviewed and it is suggested that this is the most reliable and convenient method for figuring the two-phase pressure drop.

The method is presented as follows:

(1) Find the Reynolds Number for both gas and liquid. If $R_n > 2000$ for both, the flow is fully turbulent.

(2) Select the proper coefficients. For turbulent flow, $m = n = 0.2$ and $C_L = C_g = 0.046$.

(3) Calculate X using the following equation:

$$X = \sqrt{\frac{(R_n)_G^m}{(R_n)_L^n} \; \frac{C_L}{C_G} \; \left(\frac{W_L}{W_G}\right)^2 \; \frac{\rho_G}{\rho_L}}$$

where W = mass rate of flow in lb/sec, and ρ = fluid density in lb/cu. ft.

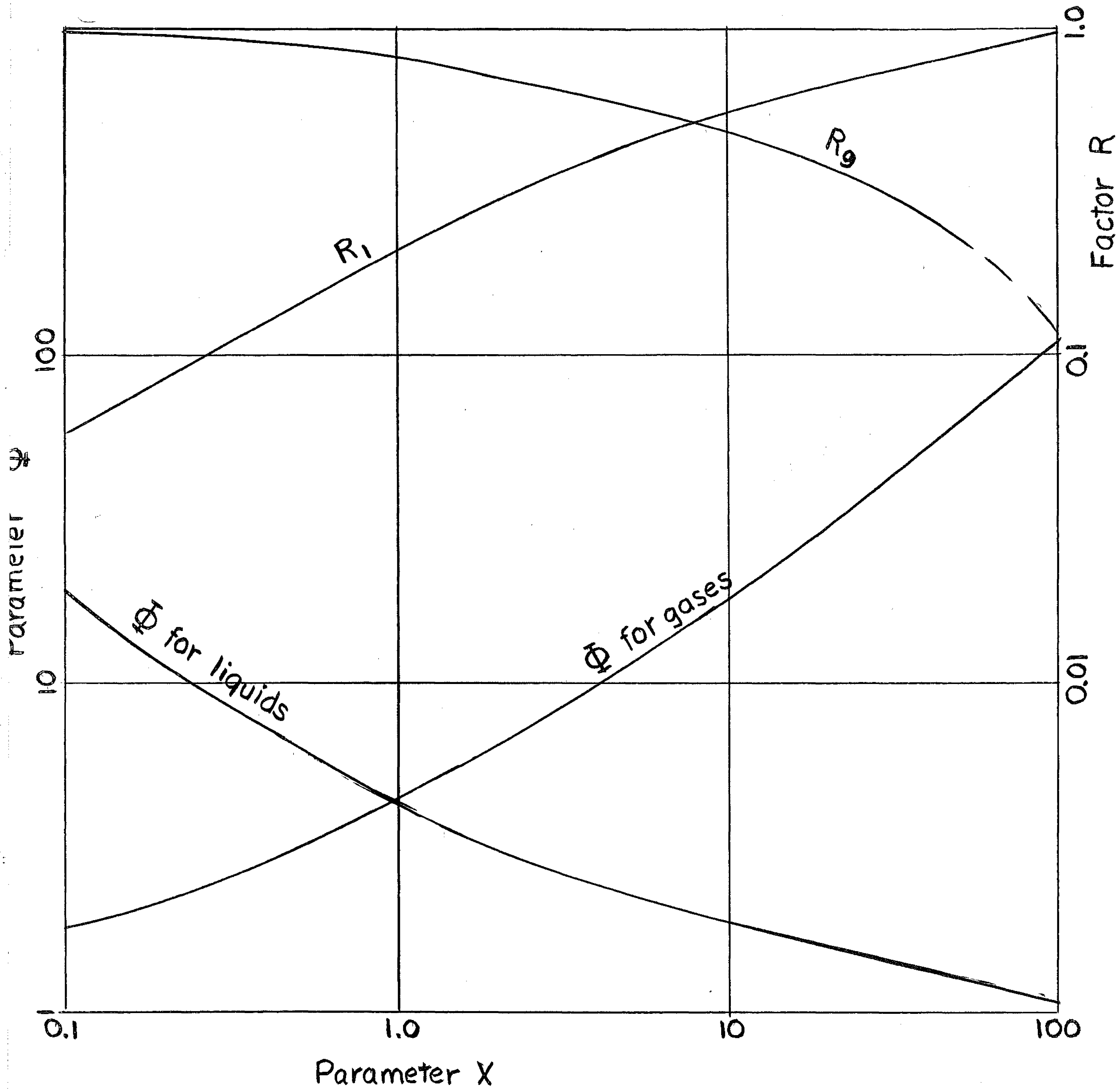
(4) From the figure presented on the next page, find Φ.

(5) Calculate the two-phase pressure drop using the following equation:

$$\Phi_{G \text{ or } L} = \sqrt{\left(\frac{\Delta P}{\Delta L}\right)_{TP} \div \left(\frac{\Delta P}{\Delta L}\right)_{G \text{ or } L}}$$

(6) From the calculated value of X find the fraction of tube filled by one phase R.

The author mentions briefly Gas Lifts, Boiler Liquids, and Condensation.



R₁
Rg
Φ for liquids
Φ for gases
Factor R
1.0
0.1
0.01
Parameter Ψ
100
10
0.1
1.0
10
100
Parameter X

Broadway, J. D. "THEORETICAL CONSIDERATIONS OF VORTEX SEPARATORS" Technical Association of the Pulp and Paper Industry 45:4 April 1962

The flow pattern of almost all vortex type separators are two dimensional. Fluid is injected tangentially and spirals downward. As the fluid finally migrates to the center it spirals up through the central outlet. The basic design parameter is the orifice ratio , a constant used for fan design, and stated as

$$0 = \frac{Q}{D^2\sqrt{\frac{2gP}{\rho}}}$$

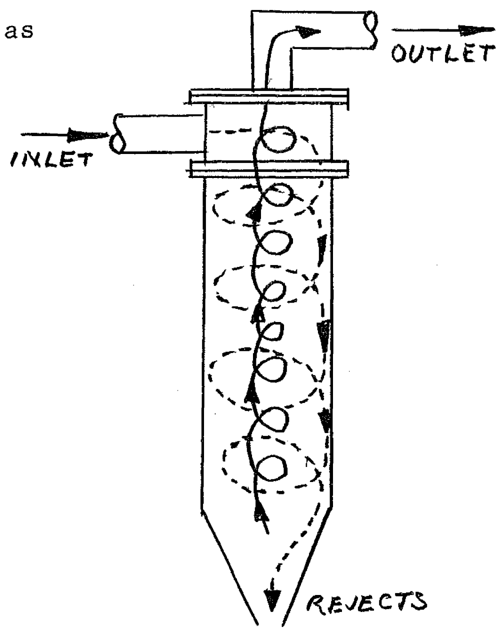
where 0 = orifice ratio

D = a standard dimension used for scaling, usually the separator barrel diameter

g = gravitational constant

ρ = fluid density

P = pressure drop across the device



Basic Vortex Separator

The parameter 0 can be varied by changing the ratio of the inlet area to the barrel area, hence changing the amount of pressure energy converted to velocity. It was found that for small values of 0, the size of the system had to be increased but the better cleaning performance more than compensated the additional cost.

If frictional losses are considered it is common to express the vortex equation as

$$v = K_g r^n$$

where

v = velocity at any radius

r = radius

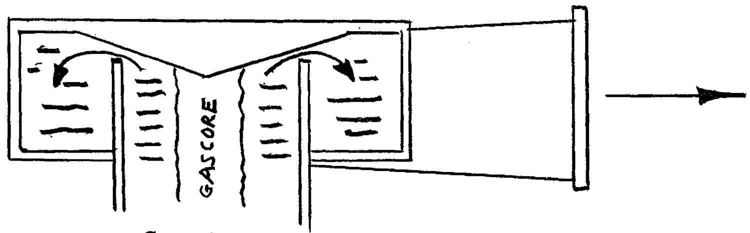
K_g = general vortex constant

n = constant between +1 and -1.

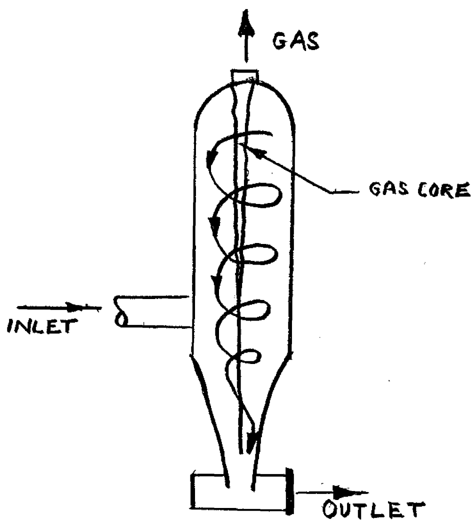
If n = -1, a condition of no frictional loss is assumed (free vortex). At n = +1, there is a complete frictional loss of energy, and a wheel type vortex results. The velocitites in the vortex are important in determining the pressure drop across it. The pressure drop increases from the wheel type vortex (n = +1) to the free vortex (n = -1). This is due to the larger fluid velocities at the center as a result of reduced energy losses.

In the center of the vortex, there usually exists a space devoid of liquid due to the high centrifugal force which flings liquid from it. The gas core extends the length of the barrel and is withdrawn at the bottom by a vacuum pump. At the top of the barrel, the fluid is withdrawn and the gas is held in with the use of a "coretrap". The vortex is conveyed through the barrel and brought against the blunt cone where it will escape out the sides while the gas core is held inside. In order that the dissolved gases might be brought into the core, the inlet velocity must be relatively high.

At the conclusion of the paper, the author presents some examples of vortex separators manufactured by his corporation (Nichols Eng. & Res. Corp.). Most of these are designed to remove grit and air. One, called the Foamtrap, is designed expressly for the removal of air. It has been patented by the author in collaboration with H. Freeman. In this device, the liquid is inserted at the top and removed at the bottom, resulting in a single vortex. The author states that this device is better for removing gases than the other, but does not give any more details.



Coretrap



Foamtrap

Chrisholm, D., and Laird, A. D. K., "TWO-PHASE FLOW IN ROUGH TUBES", Trans. ASME, 1958, p. 277

The authors attempt to build on the groundwork laid by Martinelli and co-workers in the analysis of two-phase flow. Since this earlier work involved smooth tubes only, it became necessary to include the roughness effect in the analysis. This was accomplished by re-defining the parameter X as $\bar{X}$. Since X was derived for n=0.25 only (n = power of Reynolds number in the friction factor relation $\lambda' = C' N_R^{-n}$) $\bar{X}$ now becomes applicable to all values on n.

It was found that in plotting the parameter $\bar{X}$ versus $\left(\frac{\Delta P_{TP}}{\Delta P_L} - 1\right)$ the data falls close to lines of the form

$$\frac{\Delta P_{TP}}{\Delta P_L} = 1 + \frac{C}{\bar{X}^m} \qquad (1)$$

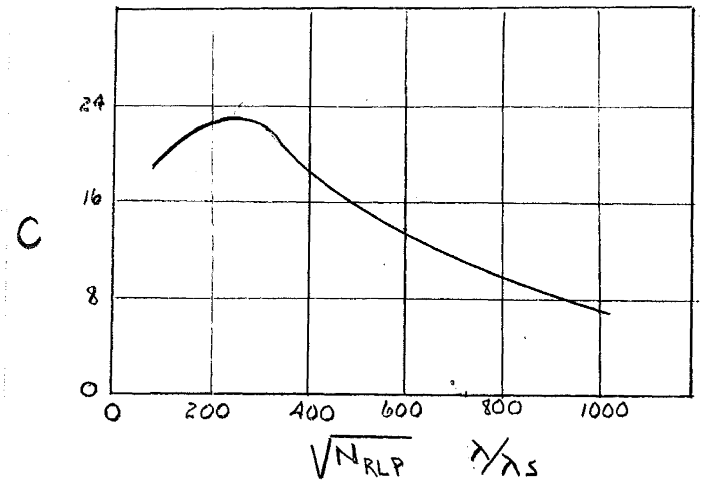
where C and m are constants for a particular liquid-flow rate and tube surface.

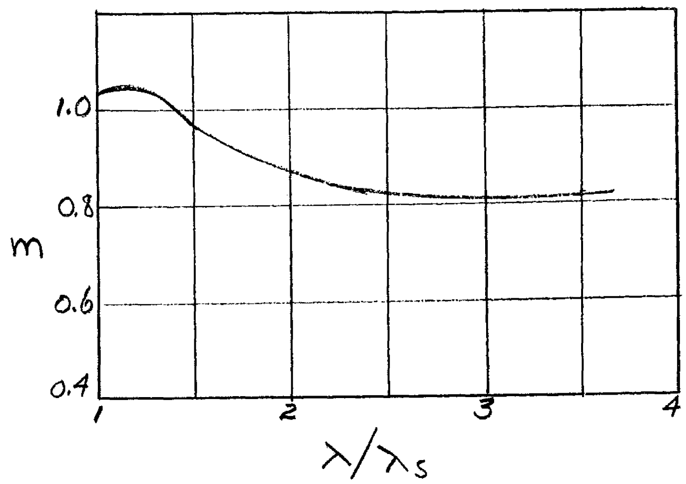
It is possible then to re-write the preceding equation as

$$\frac{\Delta P_{TP}}{\Delta P_L} = 1 + C \left(\frac{G_G}{G_L}\right)^{0.87m} \left(\frac{\mu_G}{\mu_L}\right)^{0.12m} \left(\frac{\rho_L}{\rho_G}\right)^{0.5m} \qquad (2)$$

where G = gas or liquid mass velocity based on the cross-section of the tube in lb./sec. (ft.^{2}), μ = absolute viscosity of gas or liquid in lb/sec. ft. and ρ = gas or liquid density in pcf.

Tests were conducted with smooth pipes and five different rough pipes. The resulting data was plotted as $\left(\frac{\Delta P_{TP}}{\Delta P_L}\right) - 1)$ versus $\bar{X}$ and the values of C and m were obtained. These can be conveniently plotted as shown in the following figures:





where N_{RLP} is the Reynolds Number where the liquid flows alone = $G_L D/\mu_L$; λ = pipe friction factor for rough tube; and λ_S = pipe friction factor for smooth tube.

The pressure drops for two-phase flow in rough tubes can then be estimated in the following manner:

1. Reynolds number N_{RLP} and λ for a rough tube are estimated and the friction factor for a smooth tube (λ_s) is also estimated.
2. Liquid friction pressure drop ΔP_L is evaluated on the assumption that liquid alone flows in the tube.
3. C and m are evaluated from the above figures.
4. The two-phase pressure drop is calculated by means of equation (2) above.

Using this method, the authors were able to obtain accuracies of $\pm$ 15% while the direct application of the smooth tube formula gave a maximum deviation of -46 per cent.

Donoghue, J. J. "AIR-BUBBLE GENERATOR" U. S. Navy Department, David Taylor Model Basin Report R-83 May 1943

Various methods of generating air bubbles in water are reviewed with the objective of finding some method which would produce uniform and reproducible screens of air bubbles. Two were chosen as meeting those requirements, the shear type and the dis-solution type.

The forces that act to remove the growing bubble from its attachment to an orifice determine its time of release, and therefore its size. By letting a jet of water flow past an air orifice the bubble size was controlled. It was noted that as the air flow is increased the size of the bubbles increase as long as the water velocity is zero. As the water flow is increased, with constant air flow, the size of the bubbles decreases and their number increases.

The dis-solution type bubble generator works on a principle of dissolving the air first under high pressure and then releasing it under atmospheric pressure. The physical properties of the liquid can be changed to vary the bubble size.

Three methods that did not meet the stated requirements are the simple orifice, the submerged nozzle and the porous media.

The simple orifice had many drawbacks some of which were; (a) the minimum diameter of the bubble was approximately 10 times the diameter of the orifice, (b) a small variation of relative submergence will produce non-uniform bubbles, (c) very little control over either size or number of bubbles.

The submerged venturi tube or nozzle apparatus was not acceptable because it could not produce a screen of bubbles over a rectangular area.

The porus media method had the same disadvantages as the simple orifice with even less chance of reproducibility.

The factors that determine the size of an air bubble formed in water by forcing air through a permeable surface are:

(1) the diameter of the orifice.

(2) the rate of flow of gas.

(3) the proximity of other orifices.

(4) the interfacial forces in the liquid-solid boundaries (electrolytic salt will vary the size of bubbles).

(5) the viscosity.

(6) the induction time, time of adherance to solid.

Freeman, H., and Broadway, J. D., Consolidated Paper Corp., Ltd. NEW METHODS FOR THE REMOVAL OF SOLIDS AND GASES FROM LIQUID SUSPENSIONS WITH PARTICULAR REFERENCE TO PULP STOCK CONDITIONING. Pulp and Paper Magazine of Canada, Vol. 54, No. 4, pp. 102-107, 1953.

Gas in pulp stock exits as bubbles, gas in solution, and absorbed onto fibers. To remove the bubbles, most of the total gas must be removed. A Vortrap was redesigned to remove both dirt and gas simultaneously; this was accomplished by increasing the fluid velocity within the separating tube so that a vacuum is produced and to which core the gases are displaced by centrifugal force. The gases are then removed by a vacuum pump.

The Vortrap, manufactured by Nichols Research and Engineering Corporation, has been used since 1932 as a classifier in paper mills. In 1938 H. Freeman noted a gaseous core in a glass Vortrap.

Gas in pulp stock has been determined as 3.1%, volumetric basis, with 0.25% being in the form of bubbles. The analysis of the gas is 6.5% CO_2, 25.5% O_2, and 68% unanalyzed.

At one installation a vacuum deaerator was used to remove 70% of all gases present; this included all gas in the form of bubbles. The deaerator consisted of an evacuated tank with baffle plates against which the stock was sprayed. The stock was removed by pumping from the tank, and the gases were removed by an extensive vacuum system.

A general expression for a vortex is $v\ r^n = k$, in which v is the tangential velocity at radius, r, and n, and k are constants. The extremes of n are +1 and -1. The authors assumed n as zero; thus $v = k$, a condition of constant tangential velocity at all radii.

The force acting on the wall of the tube is:

$$F = \int_{R-D}^{R} \frac{2\pi \gamma L V^2}{g} \, dr = \frac{2\pi \gamma L D V^2}{g}$$

The force acts on the area of the inside wall and is related to the pressure.

$F = 2\,\pi\,R\,L\,p.$

Consequently, $p = \dfrac{\gamma\, V^2 D}{g\,R}$, or

$$p/\gamma = \frac{2D}{R}\,\frac{V^2}{2g}$$

D = depth of liquid against the wall.

R = radius of Vortrap tube.

If the velocity head exceeds half the pressure head, a vacuum core is produced, the size of which can be predicted.

A Vortrap was redesigned to produce a sizable vacuum core, and the unit could then separate both gas and dirt. The inlet pressure and flow remained constant, but the velocity in the tube was increased and the pressure reduced.

An analysis of dirt separation was made considering particles separating in a viscous medium according to Stokes' law. The results were relatively substantiating, but the discharge of dirt was less owing to the apparent orientation of flat, clay particles.

A photograph is shown in the article of a glass Vortrap, 1-1/2 in. in diameter, with a vacuum core of 1/4 in.

In the Vortrap degasser, stock enters tangentially at the top and spirals down along the wall; most of the dirt, being adjacent to the wall, is removed at the bottom through a tangential outlet. The cleaned stock is turned upward and toward the center. Within the center of the upward rising stock a zone of reduced pressure occurs, and a vacuum develops into which the gases enter. Most of the gases are removed through a bottom center pipe connected to a vacuum pump and spray condenser; the remaining gases exit with the clean stock, are separated in a small vacuum tank and thence to the vacuum pump and condenser.

The performance of a 4-in. degassing Vortrap is as follows:

Pressure	15-35 psi
Flow	175 gpm (US)
Dirt removal, efficiency	50-85%
Percent rejects (bottom dirt)	2%
Gas removal, efficiency	80-90%

Fondrk, V. V., "THE STREAM JET EJECTOR:A VERSATILE PUMP FOR HIGH VACUUM", American Vacuum Society Symposium-Transactions 1957.

Steam Jét Ejectors have now been developed to the point where they are capable of operating at pressures from well over 50 microns of mercury absolute to well under 10 microns of mercury absolute. They can handle capacities from 0 to 250,000 cfm.

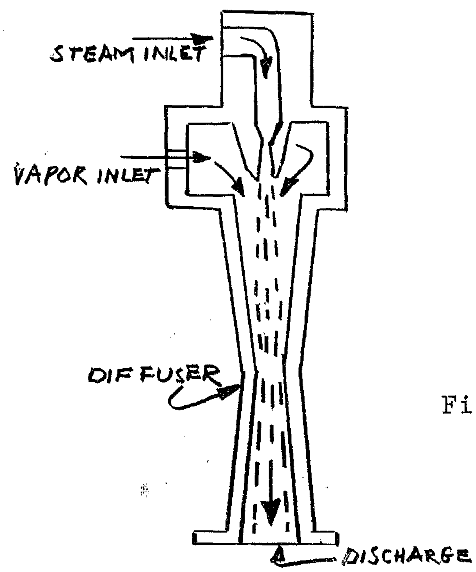


Figure 1.

Figure 1 shows a cross section of a typical steam jet ejector. Compressed steam at high pressure is allowed to expand through a nozzle thus converting its potential or pressure energy into kinetic energy. As a consequence, low pressures, or vacuums are created in the suction chamber, thus sucking the unwanted gases away.

Figure 1 shows only a one stage ejector. If very low pressures are required in the suction chamber of typical installation may have four or six stages, each separated by a compressor; or if the unwanted gas is incompressible, by a condensor and compressor both.

Figure 2 shows a typical throughput curve for an ejector used to eject air. Here chamber suction pressure is plotted against capacity in weight of air removed per hour.

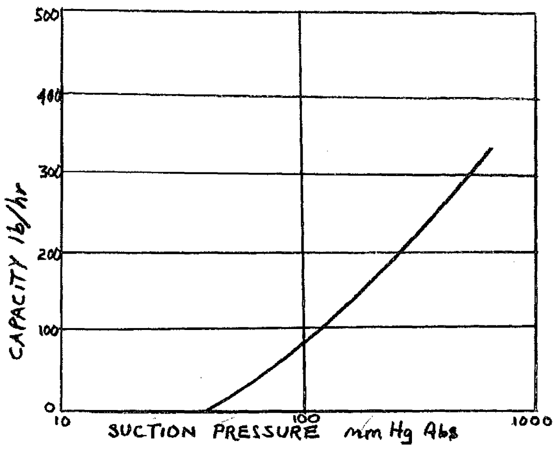


Figure 2.

Figure 3 shows the same curve where capacity is plotted in volume of air removed per hour.

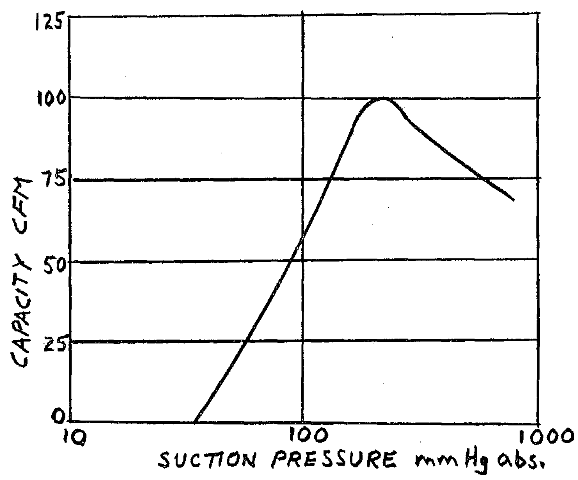


Figure 3.

Figure 2 shows that the ejector is capable of pumping increased weight rates of flow as pressure increases; whereas, figure 3 shows that the volumetric capacity reaches a peak point and then drops off as pressure increases. This peak point is usually taken as the desgin point. The ejector can be designed so that the peak point occurs at any capacity and pressure required. However, in order to prevent condensation and backflow of the expanded steam into the suction chamber when the installation is closed, the shutoff pressure should be as high as possible.

A good point to note in the design of an ejector installation is that the condenser should be located at least 34 feet above the water level in the sump in order to permit drainage by gravity even if full vacuum exists in the suction chamber. This prevents water from draining into the suction chamber (which produces some unpleasant results) even if the power system fails.

One important attribute of the steam jet ejector is its capacity to eject "dirty" gases which contain particles of dirt or liquid carryover. Any particle capable of being carried in the stream of gases being evacuated will carry through the ejector without harm to the equipment and will be washed out in the first condenser. When the load gases (unwanted gases to be ejected) are corrosive, special corrosion resistant materials can be incorporated into the design of the ejector.

Initial costs for a steam jet ejector vary with the size of the ejector. As an example, an ejector with capacity of 10,000 cfm costs from $0.75 to $1.20 per cfm, capacity of 200,000 cfm only costs $0.15 to $0.25 cfm of capacity. When steam and water are already available, installation costs approximately 40% of the first cost of the ejector.

Houghton, G., McLean, A., and Ritchie, P. "ABSORPTION OF CARBON DIOXIDE IN WATER UNDER PRESSURE USING A GAS BUBBLE COLUMN", <u>Chemical Engineering Science</u>, Vol. 7, n 1-2, 1957.

A gas bubble column is simply a hollow upright tube, or pipe in which water under pressure is made to flow downward from the top of the column while gas, after first diffusing through a perforated plate on the bottom of the column, flows upward. As the bubbles of carbon dioxide flow upward through the water they dissolve and are absorbed by water. In this manner, absorption rates as high as 90% have been obtained; almost three times the rate of the more usual "packed tower" type of water scrubber.

Any given bubble will rise with a certain terminal velocity of rise in water; and if the water is given a downward velocity equal to this terminal velocity of bubble rise, the bubble will remain stationary with respect to the column. Further increase in liquid velocity will carry the bubble in the same direction as the liquid. Hence, too great a liquid velocity will carry bubbles out of the column without being absorbed. This process is known as carryover. For the porous plates used (having porosity ranging from 72 μ to 1150 μ) it was found that the critical relative velocity between liquid and gas velocities was 0.3 ft. per sec.

It was found that for a constant liquid velocity the upward gas velocity had a great effect on the efficiency of gas absorption. At constant liquid velocity the absorption efficiency increases with rising gas velocity until a constant absorption efficiency is obtained, independent of further increase in gas velocity. For the cases tested it was found that this constant absorption efficiency is reached at a gas velocity slightly greater than 0.17 fps. The region where the absorption efficiency increases with μ_g, (gas velocity) is known as the "rising hold up" region and it was found that the absorption efficiency is proportional to $\mu_g^{0.61}$; whereas, the region where the efficiency is constant is known as the "constant hold up" region. This phenomena is explained by the fact that the bubble size is constant regardless of gas velocity. Thus, as gas velocity increases, the number, not the size, of bubbles increases, thus increasing the surface area for absorption, and thus increasing the efficiency. However, as the gas velocity increases to a point where $\mu_g > 0.17$ fps the bubbles change in size to very large, or "plug" bubbles. Thereafter the surface area remains constant, as does the absorption efficiency.

In addition to the above it was found that the absorption efficiency is proportional to $p^{-0.33}$ where p is the partial pressure of the carbon dioxide. As can be expected, absorption rate increases with temperature.

"HOW EJECTORS ARE DEVELOPING", Ed. Staff, Chemical Engineering, May 1949, p. 136

This article attempts to define the limits within which the ejector may prove to be the best all-around vacuum producer.

The advantages of ejectors are:

(1) Low first cost,

(2) Simplicity,

(3) No moving parts,

(4) Light weight, and

(5) Easy maintenance.

The disadvantages are:

(1) Usually a fixed load machine maintaining a fixed suction pressure,

(2) High water requirements,

(3) Poor steam economy at pressures below 100 psig,

(4) If operated at loads lower than design loads, instability and erratic performance results.

Some conditions which favor ejector use over other devices are:

(1) Availability of steam pressures greater than 100 psig,

(2) Availability of water,

(3) The need for suction pressures between 5 inches and 100 microns Hg.

A series of tables is presented showing ejector capacities. If the suction pressure is fixed, and the motive steam pressure is varied, the pound per hour of motive steam required to compress "n" lb/hr. of air can be measured. This measurement for various types of ejectors, with or without intercondensers and/or multiple stages, is presented in these tables. The net cfm displacement required if a mechanical pump was used to produce the same result is presented for comparison. It is seen that for low suction pressures, the ejector has a great advantage over the pump.

It is shown that by increasing the number of stages and installing intercondensers, the efficiency of the ejectors can be increased, but at the cost of higher operating expense.

The article also mentions briefly the use of ejectors as thermocompressors and boosters.

Isbin, H. S., Moen, R. H., Mosher, D. R., "TWO-PHASE PRESSURE DROP", Atomic Energy Commission Unpublished Report - 2994 November 1954.

The discussion on two-phase flow presented in this report refers to the simultaneous and co-current flow of gas-liquid mixtures. The purpose of this report is to summarize the current knowledge and limitations of the present (1954) methods of estimating two-phase pressure drops. The following topics are considered: Present status of two-phase calculation method, flow pattern, flow type, flow model, and flow stability. A extensive literature survey is included.

PRESENT STATUS OF TWO-PHASE PRESSURE DROP CALCULATION METHODS.

In many cases it is possible to show calculated two-phase pressure drop values to be within 30% of experimental values. One such example is the Martinelli method, but his covers the ranges for steady flow only. It is recognized also that total flow rate parameters are not adequately provided for in this correlation. Another deficiency in the Martinelli approach is that frictional losses, momentum changes and head effects are not fully considered. In the annular flow region, for example, gravity effects play an important role, the water layer being thicker on the bottom of the pipe than on the top.

Another method of finding pressure drops is the development of one friction factor (a combination of f_G and f_L) to cover two phase flow. One of the basic assumptions is that the gas and liquid are moving at the same velocity. This is not always true. No really good methods of "homogeneous flow" types have yet to be advanced.

FLOW PATTERNS

The mode of flow for each phase of a liquid-gas flow is determined by the shape of the confining conduit, gravitational force, interphase forces and intraphase forces. The interplay of these forces leads to a number of possible cross-sectional and longitudinal profiles for flow. The behavior of the system depends on which of these flow patterns occurs. For horizontal flow, the definitions advanced by Alves are presented. Other authors were able to duplicate these and are fairly well agreed on the definitions, but not on the most advantageous way of presenting results. In vertical flow pipes, the same definitions can be used with minor revisions.

FLOW TYPES

Flow types are usually designed on the basis of whether laminar or turbulent flow would exist if the phase under consideration were flowing alone in the pipe. Turbulent flow is said to exist at Reynolds Numbers greater than 2000; and at less than 1000, laminar flow is said to exist. Obviously there are then four possible flow types; turbulent-turbulent; turbulent-viscous; viscous-turbulent; and viscous-viscous; describing first the gas phase and the second the liquid phase.

FLOW MODELS

A variety of physical models have been used to define two-phase flow phenomena. Two of the most used are the Martinelli model and the models assuming homogeneous flow.

Martinelli Model

(This model is presented in another abstract of this report but a few notations will be repeated here).

The basic assumptions of the model are (1) the static pressure drop for the gaseous phase is equal to the liquid phase static pressure drop; (2) the volume of the pipe at any instant is equal to the sum of the volume of the gas plus that of the liquid.

As shown in the abstract of the Martinelli article, the factor β was allowed to equal unity. This resulted in the 30% error. It was later recognized that it was better to allow both α and β to vary. In the latest Martinelli paper, the correlation was extended to include condition in which the quality would vary directly with the flow rate, as in boiling water. It was discovered that the prediction overestimated the experimental tests. The difficulty was solved when the factor was reduced to the atmospheric pressure conditions. This indicated an unexpected dependency of Φ to pressure. The method is then applied to steam-water mixtures.

There have been a number of modifications and comparisons of the Martinelli Correlation, among them Levy who gave theoretical support to Martinelli's choice of parameters. Gazley's analytical development of Φ, however, is about 25% lower than the empirical function. Others have obtained experimental correlations and found Martinelli's values to be acceptable. Among these authors are Bonilla, H. A. Johnson, and VanWinger. Still others have found the method to over estimate the pressure drop, among those are Baker, Alves, Untermyer, and Schuler.

Friction Factor Models

The basis of the classification of the Friction Factor Models is the use of a single friction factor for the mixed flow. One of the more widely used methods is that of "Homogeneous Flow". The basic premise here is the assumptions of equal gas and liquid velocity and of thermodynamic equilibrium between phases (vapor-liquid equilibrium). Even though the first assumption is seldom fulfilled, many useful results have been derived with it. If this assumption does not hold, the friction factor calculated can be extremely small or even engative.

The friction factor is usually derived by using the energy balance equation, the equation of momentum, and the continuity equation. Many complex relationships have been developed from these basic equations. The experimental data verifies these equations to 50 per cent.

Other types of friction factor models have been attempted. Bergelin and Gazley observed that for both horizontal and vertical flow an increase in the liquid flow results in an increase in pressure drop. This was attributed to the "rough-wall" effect.

Huntington and co-workers developed an expression for two phase friction factor which yielded results of 17 per cent accuracy. Their friction factor was developed empirically.

Mixed Models

Mixed models are characterized by having features based both on the Martinelli and homogeneous models, even though they are contradictory. The usefulness of these models has yet to be established.

FLOW STABILITY

Two-phase flow may become unstable during transition between flow patterns which result in a large pressure fluctuation. The instability is usually associated with the transition from bubbly to stratified flow and the transition from wavy to annular flow.

LITERATURE SURVEY

As stated earlier, an extensive literature survey is included in this report. It includes almost all of the significant papers on two-phase flow up to the year 1954.

James, W., and Silberman, E., "TWO-PHASE FLOW STUDIES IN HORIZONTAL PIPES WITH SPECIAL REFERENCE TO BUBBLY MIXTURES", St. Anthony Falls Hydraulic Laboratory Technical Paper No. 26, Series B. University of Minnesota, Minneapolis, Minnesota.

Gas-liquid flows are subject to fairly complex series of flow patterns. These may be described as follows;

(1) Bubble Flow-in which separate bubbles of gas move along the pipe with approximately the same velocity as the liquid.

(2) Plug Flow-in which bubbles in the upper region agglomerate to form larger bubbles or plugs.

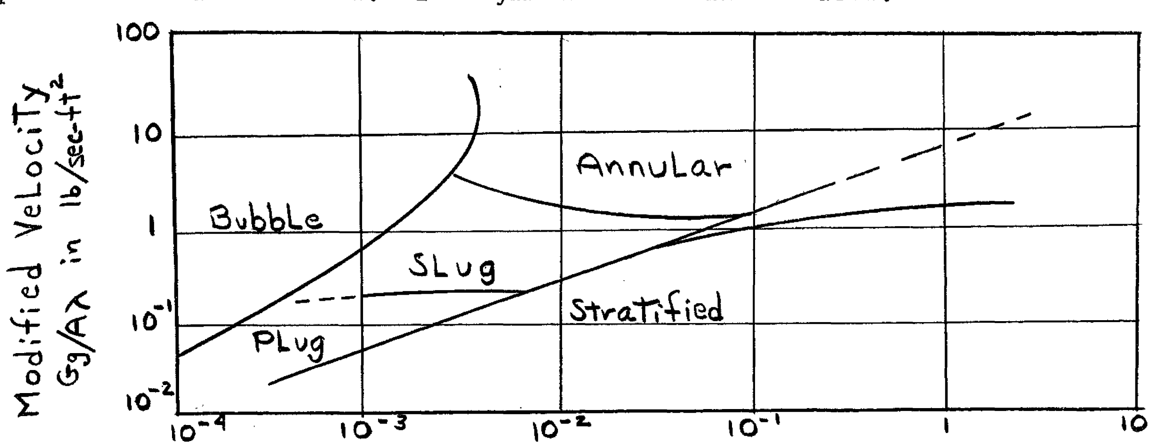
(3) Slug Flow-in which a fairly well defined interface separates the gas and liquid.

(4) Annular Flow-in which the liquid flows in a film around the pipe wall.

(5) Mist Flow-in which liquid droplets are entrained fairly uniformly throughout a gas flow.

(6) Separate Flow-in which the liquid flows along the bottom of the pipe and the gas flows above.

Baker prepared the following chart showing the relation of flow pattern to rate of flow. The symbols are defined later.



Modified Gas-Liquid Flow Ratio by Weight $\alpha/\lambda\Psi$

Figure 1.

This paper is mainly concerned with bubble flow and is divided into two main parts.

I. Study of gas-liquid flows from the mean through-flow properties. Pressure drop and friction factor are considered.

II. Study of the details of the gas-liquid flow at a given cross section of pipe. Bubble size distribution and velocity profiles are considered.

Some Preliminary Considerations

The following assumptions were made throughout this paper;

1. No absorption or evolution of gas.
2. The gas obeys the perfect gas laws.
3. Isothermal conditons exist.
4. Vapor pressure and surface tension effects are ignored.

The significant parameters and relationships are summarized below.

1. The weight rate of flow along the pipe (G) is constant, as are the liquid and gas components of weight flow (G_L) and (G_g), then $\alpha = G_g/G_L$, which is a constant for a given flow.

2. The total volume rate of flow (Q) at any position along the pipe (x) is defined by a dimensionless parameter (π) where

$$\pi_x = \left(\frac{QL}{Qg}\right)_x \tag{22}$$

3. From the specific weight of a mixture, it can be shown that

$$V_\ell = \frac{QL}{A} \tag{23}$$

where V_ℓ = a ficticious velocity of the liquid that would would exist if there were no gas flow.

4. If at any cross section (x) the liquid fraction is R_L and the gas fraction is R_g, it can be shown that

$$\left(\frac{R_g}{R_L}\right)_x = \left(\frac{A_g}{A_L}\right)_x = \left(\frac{1}{\pi} \quad \frac{V_L}{V_g}\right)_x \qquad (24)$$

5. Two additional parameters (which are empirical, used by Baker and appearing in Fig. 1 can be defined as;

$$\lambda = \frac{W_g}{0.075} \quad \frac{W_L}{62.3} \qquad (25)$$

and

$$\psi = \frac{73.3}{\sigma} \quad \left[\mu_L \left(\frac{62.3}{W_L}\right)^2 \right]^{1/3} \qquad (26)$$

where w = lb/ft.^{3} σ = dynes/cm. μ = centipoises.

It is noted that when water near room temperature is used as the liquid,

$$\psi = 1 \quad \& \quad \lambda = 28.8\sqrt{\pi\alpha} \qquad (27)$$

I. Pressure Drop and Friction Factor

By taking a free body of fluid in a circular pipe and applying Newton's first equation describing local conditions of a homogeneous gas-liquid mixture can be derived. For very small ratios of gas-to-liquid flow, however, the pressure drop in a bubbly mixture could be calculated approximately by treating the mixture as incompressible and using the corresponding incompressible friction factor. In other words it is assumed that f/f_L = 1.0. Substituting this friction factor in the free body equation, the following differential equation results;

$$\Theta = \left(\frac{\alpha}{1+\alpha}\right) \quad \left(\frac{g\ RT}{V_\ell^{\ 2}}\right) \qquad (28)$$

This equation must be integrated to determine pressure drop along a pipe carrying homogeneous gas-liquid mixture flow. Θ is defined then as the compressibility parameter.

If the Reynolds Number is defined as

$$R = \frac{DV\rho}{\mu} = \frac{DG}{g\mu A} = \frac{DG_L}{g\mu A} \quad (1 + \alpha) \tag{29}$$

and assumed constant, f_x does not vary with x either. Therefore equation (28) can be integrated, yielding;

$$\frac{f_x}{2D} = \Theta (\Pi_0 - \Pi_1) - \ln \left(\frac{\Pi_0}{\Pi_1}\right) \left(\frac{\Pi_0 + 1}{\Pi_1 + 1}\right)^{\Theta - 1} \tag{30}$$

The solution of this equation has to be obtained by trial and error. Since this is tedious, a more readily available method is desirable. By carrying out the logarithmic series expansion of Eq. (30) and assuming that $\frac{\Delta\Pi}{\Pi_0} < 0.5$ which is usually true in bubbly mixtures, one can find the pressure drop as

$$\Delta P = \frac{P_0 \quad (\Pi_0 + 1) \quad f_{xi}}{2 D \quad (\Theta \Pi_0^2 - 1)} \tag{31}$$

Experimental Work

Experiments were conducted for two purposes; to determine the limiting conditions of bubble flow and to verify the assumptions leading to equations (27) and (30).

The first apparatus used was a straight pipe section with air injection before the test section. All data needed to compute the friction factor as defined by Equation (30) were taken. Observations at a transparent viewing section confirmed bubbly flow.

The second apparatus employed a 180° turn in the test section.

The purpose of this was to determine the limits of the bubble flow regime and the effect of the disturbance.

It was found that when the flow was entirely bubble flow,

$$0.98 < f/f_L < 1.25$$

the higher values occurring at the higher gas-liquid flow ratios. The variation might be a result of the fact that $\mu > \mu_L$ and becomes increasingly so as the air-water ratio increases.

In the 180° bend apparatus, the f/f_L was approximately the same upstream and downstream of the bend as long as the flow was in the bubble regime. This was not true, however, outside the bubble regime.

If f/f_L is plotted versus the compressibility parameter (Θ) (Equation 28) the relationship

$$f/f_L \approx 1 + 0.035\sqrt{\Theta} \qquad (32)$$

will fit the data as long as the flow is in the bubbly regime. At $\Theta = 20$ the f/f_L drops to about 0.5 at $\Theta = 10^5$ and then rises again. Bubble flow always exists at $\Theta < 6$ and at water flow rates greater than 500 $lb/ft.^2$-sec.

II. Analysis at a Cross Section

The apparatus was so designed that the flow would separate into two parts, the smaller part diverted through a narrow tests section where it was possible to count the bubbles and determine their size by photography.

The dimensionless bubble diameter, d' was plotted on semi-log paper to show the % finer. This diameter was defined as

$$d' = V_{\ell} \quad d \sqrt{\frac{\rho_L}{\sigma_D}} \qquad (33)$$

where d = bubble diameter under atmospheric conditions

σ = surface tension of water

D = pipe diameter

This plot yields a straight line the equation of which is

$$d' = 7.50 - 1.49 \ln (\% \text{ larger}) \qquad (34)$$

It was found that there is a definite trend for the mean bubble diameter to increase with the square root of the diameter of the pipe and inversely as the liquid flow rate per unit area.

The variation of V_g/V_L was plotted but no correlation was found to exist between f/f_L and V_g/V_L.

The velocity profiles in the pipe were determined using Pitot tubes. It was found that the velocity distribution is materially affected by the presence of air bubbles, particulary near the top of the pipe. This non-symmetrical profile seems to indicate a secondary current upward in the center of the pipe and downward around the walls. The upper part of the pipe, where the bubbles are more concentrated, is effectively rougher than the bottom.

As a tentative conclusion, it appears that disturbances of the flow such as produced by bends, do not alter the conclusions already given above.

Lockhart, R. W., Martinelli, R. C., "PROPOSED CORRELATION OF DATA FOR ISOTHERMAL TWO-PHASE, TWO-COMPONENT FLOW IN PIPES", Chemical Engineering Progress, 45:1, p. 39-48, January 1949.

The basic postulates upon which is based the analysis of pressure drop resulting from the simultaneous flow of a liquid and gas are:

(1) Static pressure drop is equal for both the gas and liquid phase,

(2) The volume of gas plus volume of liquid must equal volume of pipe.

Working with these postulates, the authors have developed a method by which the pressure drop can be predicted for both laminar and turbulent flow. Using the first postulate and the Fanning equation for pressure drops, the authors derived the following relationships: (subscripts 1 and g refer to liquid and gas respectively; the remainder of the symbols are defined at the end of this abstract).

$$\frac{(\Delta P/\Delta L)_{TP}}{(\Delta P/\Delta L)_1} = \alpha^{\frac{n-2}{2}} \left(\frac{D_p}{D_1}\right)^{5-n/2} = \Phi_1$$

$$\frac{(\Delta P/\Delta L)_{TP}}{(\Delta P/\Delta L)_g} = \beta^{\frac{m-2}{2}} \left(\frac{D_p}{D_g}\right)^{\frac{5-m}{2}} = \Phi_g$$

The application of the second postulate resulted in the following:

$$R_g = 1 - \alpha \left(\frac{D_1}{D_p}\right)^2 \qquad R_e = 1 - \beta \left(\frac{D_g}{D_p}\right)^2$$

In all these equations, four variables have appeared, namely,

$$\frac{D_e}{D_p}, \quad \frac{D_g}{D_p}, \quad \alpha \text{ and } \beta$$

which can be expressed in terms of four experimentally determined variables;

Φ_l, Φ_g, R_1 amd R_g.

One more variable is apparent in the derivation, and this can be expressed as

$$\frac{R_{e\,g}^{\;m}}{R_{e\,l}^{\;n}} \quad \frac{C_e}{C_g} \quad \left(\frac{W_e}{W_g}\right)^2 \quad \frac{\rho_g}{\rho_e} = X^2$$

This variable can be shown to equal the ratio of liquid pressure drop to gas pressure drop, assuming each phase flows separately;

$$X^2 = \frac{(\Delta P/\Delta L)_e}{(\Delta P/\Delta L)_g}$$

It is now assumed that the four variables, Φ_g, Φ_e, R_g, and R_1 are all functions of the parameter X. In this paper, the authors propose to evaluate the correct form of X for all types of flows by the substitution of the appropriate exponents n and m and the constants of C_1 and C_g. For fully turbulent flow, ($R_e > 2000$) the following values are given:

$$n = m = 0.2 \text{ and } C_1 = C_g = 0.046.$$

All runs were made with either horizontal pipe or sloping pipes corrected for static head.

Since $\Phi_g = X\Phi_e$, for fully turbulent flow X was found to be equal to

$$\left(\frac{W_e}{W_g}\right)^{1.8} \quad \left(\frac{\rho_g}{\rho_e}\right) \quad \left(\frac{\mu_e}{\mu_g}\right)^{0.2}$$

This relationship is presented in graphical form on page (18) of this report, and these curves can be used to predict pressure drops in two-phase flow.

In a duscussion by Carl Gazley and O. P. Bergelin the curves presented by Lockhart and Martinelli are verified to ±20% and -30%.

Notation:

D = hydraulic diameter, gas or liquid

D_p = inside diameter of pipe

W = weight rate of flow, lb/sec.

ρ = weight density lb/cu. ft.

μ = absolute viscosity

P = difference in static pressure in length ΔL

C = constant in Blasius equation

m&n = exponents in Blasius equation

R_e = Reynolds number = $\frac{4 \quad W}{\pi D \mu}$

R = fraction of tube filled by gas or liquid

Subscripts:

TP = two phase

l=l = liquid

g = gas

p = pipe

Martinelli, R. C., Boelter, L. M. K., Taylor, T. H. M., Thomse, E. G., and Morrin, E. H., "ISOTHERMAL PRESSURE DROP FOR TWO-PHASE COMPONENT FLOW IN A HORIZONTAL PIPE", Transactions A. S. M. E., February, 1944, p. 139.

Tests conducted on 1 inch and 1/2 inch pipes, using air and various liquids such as water, benzine, kerosene, etc. The air was introduced into the pipe line a few feet in front of the test section. It was found that for minimum slugging, the air had to be introduced into the upper section of the pipeline and perpendicular to it.

Pressure drops in the test section were measured and plotted against the air flow rates. Some of the general trends evidenced from these plots are:

1 The static pressure drop for two-phase flow is always greater than the pressure drop for each phase flowing alone.

2 When air flow approaches zero, the pressure drop due to pure liquid is approached.

3 Flow of both air and liquid may be turbulent or laminar.

In this abstract, only the fully turbulent flow conditions are described.

There are two basic postulates on which the analysis of the results was based:

I. The static pressure drop for the liquid phase must equal the static pressure drop for the gaseous phase.

II. The volume occupied by the liquid plus the volume occupied by the gas at any instant must equal the total volume of the pipe.

These postulates lead to the following equations:

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = f_1 \frac{\gamma_1 V_1{}^2}{D_1\ 2g} \quad \text{and} \quad \left(\frac{\Delta P}{\Delta L}\right)_{TP} = f_g \frac{\gamma_g V_g{}^2}{D_g\ 2_g} \qquad (1) \text{ and } (2)$$

The subscripts g and 1 refer to the gas and liquid phase respectively. The hydraulic diameter D_1 and D_g will always be less than the pipe diameter D_p.

For the cross sectional area, the following relationships may be written:

$$A_1 = \alpha \left(\frac{\pi}{4} D^2{}_1\right) \quad \text{and} \quad A_g = \beta \left(\frac{\pi}{4} D^2{}_g\right) \qquad (3) \text{ and } (4)$$

where α and β are the ratios of the actual cross-sectional area of the flow to the area of a circle of diameter D_1 and D_g respectively. In the following analysis β is assumed to be unity and α will be unknown. (This approximation could be refined by successive trials assuming both α and β as unknown).

The friction factor may be written in the general Blasius form as

$$f_1 = \frac{\left(\frac{\pi}{4}\right)^n C_1}{\left(\frac{W_1}{\alpha D_1 \mu_1 g}\right)^n} \quad \text{and} \quad f_g = \frac{\left(\frac{\pi}{4}\right)^m C_g}{\left(\frac{W_g}{D_g \mu_g g}\right)^m} \qquad (5) \text{ and } (6)$$

For turbulent flow, $C_1 = C_g = 0.184$ and $m = n = 0.2$ (For laminar flow $C_1 = C_g = 64$ and $m = n = 1$).

Substituting the values for turbulent flow in equations (1) and (2) and with further manipulation, it can be shown that

$$D_g = \frac{D_p}{\sqrt{1 + \alpha^{1/4} \left(\frac{\gamma_G}{\gamma_e}\right)^{0.416} \left(\frac{\mu_e}{\mu_g}\right)^{0.083} \left(\frac{W_e}{W_g}\right)^{0.75}}}$$

and further that

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(\frac{\Delta P}{\Delta L}\right)_g \left(\frac{D_p}{D_g}\right)^{4.8} \qquad (8)$$

hence

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(\frac{\Delta P}{\Delta L}\right)_g \left(1 + \alpha^{1/4} \left(\frac{\mu_1}{\mu_g}\right)^{0.083} \left(\frac{\gamma_g}{\gamma_1}\right)^{0.416} \left(\frac{W_1}{W_g}\right)^{0.75}\right)^{2.4} \quad (9)$$

The preceding equation is the basic equation used in calculating the pressure drop. Only the term α remains to be found.

For simplicity, let

$$X = \left(\frac{\mu_1}{\mu_g}\right)^{0.111} \left(\frac{\gamma_g}{\gamma_1}\right)^{0.555} \left(\frac{W_1}{W_g}\right) \quad (10)$$

Plotting $\alpha^{1/4}$ vs. $X^{3/4}$ yields a smooth line with very little scatter.

Since $\alpha^{1/4}$ is now a function of X, equation (9) reveals that

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(\frac{\Delta P}{\Delta L}\right)_g \left[1 + \alpha^{1/4} X^{3/4}\right]^{2.4} \quad (11)$$

$$= \left(\frac{\Delta P}{\Delta L}\right)_g \left[1 + f(X) \quad (X)^{3/4}\right]^{2.4}$$

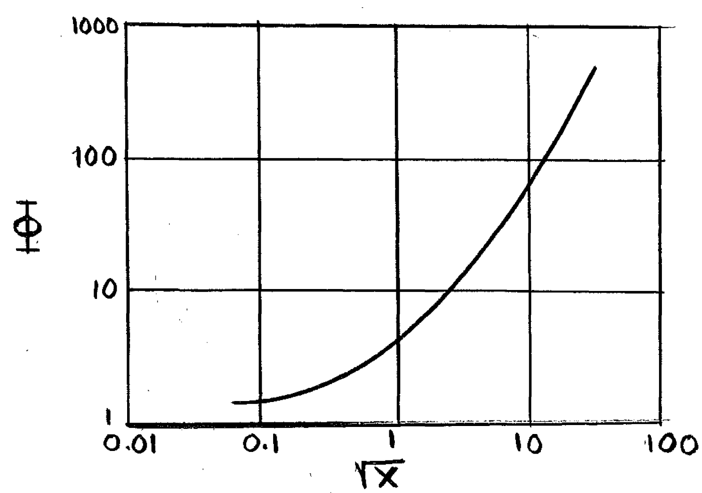
and for each maginitude of X a value of $\alpha^{1/4}$ was established. Then

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(\frac{\Delta P}{\Delta L}\right)_g \Phi^2$$

where

$$\Phi = \left[1 + \alpha^{1/4} \quad X^{3/4}\right]^{1.2} \quad (12)$$

The following graph was then plotted showing Φ as a function of two-phase flow modulus $\sqrt{X}$ for flow mechanism in which both liquid and gas are in a turbulent motion.



This figure was utilized to predict some pressure drops in the test apparatus and the preducted results were plotted against the actual. This plot showed good agreement.

EXAMPLE OF METHOD

It is desired to estimate the pressure drop in 100 ft. of 2 inch pipe in which air is flowing at a rate of 0.50 lb per sec., and water is flowing at a rate of 6 lb. per sec. The temperature of the fluids is 77° F and the average pressure in the line is 100 psia.

(a) Determination of properties and type of flow:

W_g = 0.50 lb. per sec.

γ_g = 0.502 lb. per cu. ft. at 100 psia and 77° F

μ_g = 0.388×10^{-5} lb.-sec. per sq. ft. at 100 psia and 77° F

Reynolds modulus for gas considered as a single phase = $R_g = \frac{4}{\pi} \cdot \frac{W_g}{D_p \mu_g g}$ = 295,000. Likewise, the Reynolds modulus for liquid = 73,500. Equation (9) may be utilized due to fully turbulent flow.

(b) Calculation of pressure drop due to air alone: The usual calculation reveals that for R = 295,000 the friction factor is 0.018 and the pressure drop is 1.03 psi per 100 ft. of pipe.

(c) Calculation of X:

Using equation (10), it is seen that X = 1.28

(d) Calculation of Φ :

$\sqrt{X}$ = 1.12 and hence Φ = 4.6 (from the figure)

(e) Calculation of two-phase pressure drop:

$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = (4.6)^2 \, (1.03)$ = <u>21.8 psi per 100 ft. of pipe.</u>

Murdock, J. M., "TWO-PHASE FLOW MEASUREMENT WITH ORIFICES", Transactions ASME, Dec. 1962, Series D, p. 419

This paper presents a practical method for computing two-phase flow rates through standard orifice meters to a tolerance of 1.5%. Numerous data sources were used and various liquid-gas combinations tried. By plotting $\left(\frac{\Delta P_{TP}}{\Delta P_G}\right)^{1/2}$ vs. $\left(\frac{\Delta P_L}{\Delta P_G}\right)^{1/2}$, a straight line resulted yielding the equation $\left(\frac{P_{TP}}{P_G}\right) = 1.26 \left(\frac{P_L}{P_G}\right) + 1$. Using this equation and the actual equation of flow through an orifice meter, the following relationship was derived for the weight rate of two phase flow (lb/hour):

$$w_h = \frac{359\ K_G\ Y_G\ F_a\ \ d^2 \sqrt{H_{wTP}\qquad {}_G}}{(1-y) + 1.26\ y\ \frac{K_G Y_G}{K_L}\sqrt{\frac{\gamma_G}{\gamma_L}}} \qquad (1)$$

This equation applied only when:

(1) $0.25 < d/D < 0.50$

(2) Minimum Reynolds number for gas = 10,000 and liquid = 50

(3) Minimum volume ratio of gas to liquid = 100:1

(4) Maximum liquid weight fraction = 90%

Notation:

Subscript G applies to flow of gas only

Subscript L applies to flow of liquid only

Subscript TP applies to two-phase flow

K = flow coefficient

d = orifice diameter

γ = specific weight

F = factor to account for thermal expansion

y = ratio of weight of liquid, two-phase to total

Y = expansion factor

D = inside pipe diameter

h_w = effective differential head

"UNIQUE PUMP WILL HANDLE ANY PROPORTIONS OF LIQUID AND VAPOR", Nash Engineering Co., Chemical Engineering, April 27, 1964, p. 104

This pump is unique in that it has one piece rotating member that provides liquid ring compression and conventional centrifugal action in successive stages.

The first stage involves a rotor with curved blades revolving freely in an elliptical casing which is partially filled with liquid. As the rotor turns the fluid (liquid and/or gas) is forced to the outside, but at the small diameter section of the casing the fluid is compressed and is pushed through a compartment into the eye of the second stage impeller. The noncondensible gases are separated by centrifugal action and escape through rotating ports, discharging through a casing connection.

This pump has many applications since it can operate at a large capacity and is able to remove vapors at large temperatures ranges. It is very compact and is close coupled to the drive unit. It is at present manufactured in two sizes. The largest pump has a capacity of 40 gpm of condensate alone, or 20 gpm condensate and 20 cfm saturated air. The usual construction consists of bronze rotor impeller and lobe, and cast iron volute.

Nicklin, D. J., Wiles, J. O., and Davidson, J. F., "TWO-PHASE FLOW IN VERTICAL TUBES", Institute of Chemical Engineers-Transactions, Vol. 40 n1, 1962

A number of flow patterns have been observed when gas and liquid flow together up a vertical tube, and one of the more common of these patterns is the so called "slug flow" which is characterized by very large, round nosed bubbles, or slugs of gas, which move at a velocity greater than the average velocity of the surrounding liquid.

It has been found that such slugs move at a velocity relative to the surrounding liquid which corresponds to that predicted by the Dumitrescu Theory:

$$\mu_o = 0.35\ (g\ D)^{1/2}$$

where

μ_o = relative velocity of the rising slug of gas.

g = acceleartion of gravity.

D = the tube diameter.

The absolute velocity can be taken as the sum of the Dumitrescu velocity, μ_o, plus a component due to the motion of the liquid. For upward flow of water it was found that this component is 1.2 times the average liquid velocity when the Reynolds number is greater than 8000.

thus: $$\mu_s = 1.2\ \mu_2 + 0.35\ (g\ D)^{1/2}$$

where

μ_s = absolute velocity of the gas slug

μ_2 = average velocity of the liquid

Price, R. V., "VORTRAP-VORJECT-VORVAC", Technical Association of the Pulp and Paper Industry, 42:12, December 1959

The author discusses three cyclone classifiers manufactured by Nichols Engineering and Research Corporation and attempts to clarify the differences inherent in each piece of equipment and also to specify the possibilities of application.

The first the Vortrap, is applied to removing sand, grit, and foreign material from all types of pulp and papermaking stock. It is not designed to remove gases. The second, the Vorject, is merely a modification and increases the efficiency but still does not remove air.

The Vorvac, however, is designed to remove grit and dirt as well as air. The liquid is forced through a nozzle type inlet which converts pressure energy into velocity. This nozzle enters tangentially into a cylinder, causing the liquid to follow the wall in a helical path. The dirt is carried close to the wall and is swept to the bottom of the cylinder. The cone turns back the cleaned stock which moves upward and exits at the top. The gases come out of solution as the result of the vacuum produced in the middle core. The headpiece is designed to trap the gas core so that no gas is removed off the top. A vacuum pump draws the gases through the cone.

The Vorvac is available in two sizes, the largest being 24 inches in diameter and a capacity of 1125 g. p. m. By installing units in parallel, any flow requirement can be handled.

Ripken, J. F., and Killen, J. M., GAS BUBBLES:THEIR OCCURRENCE, MEASUREMENT, AND INFLUENCE IN CAVITATION TESTING, Saint Anthony Falls Hydraulic Laboratory, Technical Paper No. 21, Series A, University of Minnesota, Minneapolis, Minnesota

As indicated by the title of this paper, gas bubbles are discussed in three parts.

Occurrence of Small Bubbles

Water which is freed of gas-water interfaces by containment in solid boundaries and the removal of large bubbles has a substantial resistnace to internal rupture, usually measureable at many hundreds of atmosphere. In natural waters, this tensile strength is almost zero.

A stable spherical gas bubble represents a balance between numerous factors such as surface tension, vapor pressure, partial pressure of the gas within the bubble, relative saturation of the gas, and external pressure. The surface tension becomes increasingly important as the bubble size decreases. This surface tension produces high internal pressures, and negative external pressures are frequently required for growth. These high internal pressures should lead to the eventual disappearnace of all bubbles, but it has been found that for some obscrue reason, this does not occur.

It was found that the gas bubbles remain very small in quiescent systems but that the introduction of mechanical agitation greatly accelerates gas transfer.

Vortex generators such as propeller tips tend to promote basic diffusion growth of bubbles as well as growth through the rapid coalescence of many small bubbles into fewer and larger bubbles. The influence of viscosity on dampening of vorticity contributes to the higher resistance to cavitation in viscous liquids.

Measurement of Gaseous Waters

The early work on measuring the gas content was performed using the assumption that the significant influences of the gas could be measured by the relative total gas content. This can be done by removing a measured portion of the sample and measuring the released gas. This method, however, does not discriminate between dissolved and free forms of gas. Moreover, it requires continual monitoring and actual removal of part of the sample.

The United States Navy uses a continuous monitoring device to measure gas content which scrubbes the sample of gas in an atmosphere of hydrogen. The gas is then measured for thermal conductivity and compared to pure hydrogen.

Many other methods have been attempted to provide an acceptable method of measuring gases. Among those are light scatter, gamma rays, and ultrasonic energy decay.

The Saint Anthony Falls Hydraulic Laboratory developed a device which was based on the velocity of propagation of an elastic pulse. Gasified mixtures were found to introduce a delay in time of propagation and this delay was translated to gas content. These measurements could be made continuously and instantaneously. This method did not correlate very well with free gas volume, however, and is at present under further study.

The Influence of Gas Bubbles in Recirculating Facilities

In studies conducted at St. Anthony Falls Laboratory, some significant conclusions can be derived. It became evident that free gas in a water tunnel is some function of at least the following three factors: (a) the total gas content, (b) the tunnel pressure, and, (c) the tunnel velocity. For example, if the tunnel velocity is zero and the pressure ishigh, only negligible free gas is present. If, however, a modest velocity is introduced, the marked evolution of free gas is seen.

It was found that water velocities as low as 10 feet per second produced vorticity sufficient to grow large gas bubbles. This indicates that prototype propellers, pumps, and turbines will normally be supplied with water which will readily cavitate. It was also found that the hysteresis in pressure controlled incipient cavitation is insignificant under stabilized free gas conditions.

Santalo, M. A., "TWO-PHASE FLOW", Applied Mechanics Review, October, 1958, pp. 523-526

The author attempts to correlate all the knowledge about two-phase flow and does not attempt to either reproduce any experimental results or to add any individual findings.

Because of the great difference in density and viscosity of the two fluids, the velocity of the gas phase is greater than the velocity of the liquid, giving rise to the so called "slip". The fraction by weight of vapor flowing through the pipe per unit time is usually denoted by the term quantity x, where

$$x = \frac{\frac{V_g A_g}{\nu_g}}{\frac{V_g A_g}{\nu_g} \quad \frac{V_L A_L}{\nu_L}} = \frac{1}{1 + \frac{V_L}{V_g}\frac{\nu_g}{\nu_L}\frac{1-Rg}{Rg}} = \frac{1}{1 + \frac{V_L}{V_g}\frac{1-Xs}{Xs}} \tag{7}$$

where the subscript g and L denote gas and liquid respectively, and V = velocity, ν = specific volume, and A = area. The static quantity x_s is the percentage of vapor by mass, or

$$x_s = \frac{\frac{A_g}{\nu_g}}{\frac{A_g}{\nu_g} + \frac{A_L}{\nu_L}} \tag{8}$$

Hence $x = x_s$ only when the velocities of the two phase are equal.

It is often convenient to use a volumetric quantity (R_g) which is related to x and x_s by equation (1), and is defined as

$$R_g = \frac{A_g}{A} \tag{9}$$

FLOW REGIME

Analysis of slip and pressure drops depends on the knowledge of the various flow regimes. These regimes have been described by Alves, Baker, and others.

The regimes vary from bubble flow (gas flow inside liquid) to spray flow (liquid inside gas). As the ratio varies, the slip velocity increases and then decreases, being zero at both ends, $x = 0$ and $x = 1$.

It is found that the most significant parameter for describing these regimes was R_g. Brancart made the following observations on air-water mixtures:

$R < 5\%$: small bubbles of air in water

$R \approx 5\%$: bullet-like plugs followed by small trailing bubbles

$R \approx 40\%$: annular flow, with bubble liquid film covering the inside of the pipe and the core of air in the middle flowing much faster than the surrounding water

$R \approx 75\%$: pure annular, no bubbles in layer of water.

PRESSURE DROP IN ADIABATIC HORIZONTAL FLOW

The author believes that the efforts made by various authors to use a single-phase friction factor have met with little success <u>when applied to a wide range of flow conditions</u>. He does concede, however, that some of these methods, including the Fanning equation presented by Baker do give fairly reliable answers.

The author believes that the most successful correlation of two-phase pressure drop was presented by Martinelli and his collaborators. It was found that the isothermal pressure drop $(\frac{\Delta P}{\Delta L})_{TP}$ should be made non-dimensional by dividing it by $(\frac{\Delta P}{\Delta L})_g$. The latter can be calculated from known friction factors. The pressure drop parameter,

$$\phi^2 = \frac{(\frac{\Delta P}{\Delta L})_{TP}}{(\frac{\Delta P}{\Delta L})_g} \tag{10}$$

as well as R_g were found to be a function of

$$X^2 = \frac{(\frac{\Delta P}{\Delta L})_L}{(\frac{\Delta P}{\Delta L})_g} \tag{11}$$

Martinelli distinguished 4 curves to cover all the flow regimes. He assumed turbulent flow when $R_N > 2000$. Others have confirmed these findings, but only for cases where $\rho_g/\rho_L \approx 0$.

PRESSURE DROP IN ADIABATIC VERTICAL FLOW

In vertical pipes, the total pressure drop is often split into two terms, a friction pressure drop and a gravity pressure drop. In the bubble flow regime, a simple expression

$$\frac{Ug - Us}{U_L} = \frac{Rg}{K - Rg} \tag{12}$$

is suggested by the author. This expression defines U_s as the apparent velocity of the homogeneous mixture. U_s is equal to the velocity of the gas phase U_g when the velocity of the liquid phase $U_L = 0$.

K is a constant found to be about 0.9.

The gravity pressure drop can be defined as

$$(\frac{\Delta P}{\Delta L})_{gr} = \frac{1}{\nu_L + x(\nu_g - \nu_L)}\rho \tag{13}$$

The true density is obtained when x is replaced by x_s as defined above. The combination of these two terms (11) and (12) can be used to find the total pressure drop.

The author also has a short summary of the analysis of pressure drop in the flow with heat transfer. This is not applicable to the current project.

Schanzlin, E. H., "HIGHER SPEEDS AND PRESSURES FOR THE HYDRAULIC PUMP", Proceedings, National Conference on Industrial Hydraulics, Vol. X, pp. 35-48, October 1956

The paper presents a pump manufacturer's view of the accelerating trend towards higher pressures and higher speeds in commercial hydraulic pump applications.

The physical phenomenon connected with the flow of oil from the reservoir to the pumping mechanism follows simple physical laws which generally are well known to the hydraulic engineer. These usually cover pressure losses and fluid velocities. To demonstrate the significance of low pressures losses and low velocities, attention must be given to the solubility of air in oil. Air dissolves in various liquids according to the physical characteristics of the liquids. Some have greater affinity for air than others, but the solubility in any given liquid is directly proportional to the absolute pressure of the air above it. Double the pressure, and the amount of air which goes into solution is doubled.

Conversely, if the pressure is halved, half the air is released from the solution. This important relationship is known as Henry's law, and according to it, the concentration of the dissolved gas in solution is directly proportional to the concentration in the free space above the liquid.

In determining air release from liquids, allowance usually must be made for the vapor pressure of the liquid. However, in case of low vapor pressure fluid such as hydraulic oil, air can be considered as the sole consideration at the normally encountered pump inlet pressures and temperatures.

As an example, air solubility in water may be observed from the way in which air bubbles begin to gather on the inner walls of a glass of tap water after it has been permitted to stand undisturbed for a few minutes. The separation of air from the water is caused by a combination of two factors:

(1) the pressure on the water has been relieved by the act of withdrawing it from the supply line, and

(2) the water warms up from contact with the higher temperature of the room.

The first factor precipitates air because there is no longer sufficient pressure on the water surface to contain all dissolved air. The second factor causes bubbles to form because the vapor pressure of the water increases and an unbalance of internal pressure develops which drives air out of solution. In further demonstration of air solubility, this same glass of water can be stirred to free the bubbles so that the glass is perfectly clear. If then, the water is heated slightly, the bubbles again will form long before the boiling point is reached.

A more familiar example of gas solubility in liquid will be recalled from the behavior of carbonated beverages. In this case, the liquid is principally water and the gas is carbon dioxide (CO_2). Carbon dioxide is considerably more soluble in water than is ordinary air, but the principle is the same. A capped bottle of such a beverage can be shaken vigorously without significant gas evolution. If any undisturbed bottle is uncapped so that the internal pressure is released gradually, little gas evolution will occur immediately unless agitation is given to the liquid. Vigorous agitation speeds up the precipitation of the bubbles so that liquid erupts violently into a mass of foam. From these examples, it can be seen that the act of relieving the confining pressure on a sample of liquid which has been in contact with air will provoke the release of some air from solution. Likewise, an increase in temperature will cause separation of dissolved air even though the pressure remains constant.

Silberman, E., "AIR BUBBLE RESORPTION", St. Anthony Falls Hydraulic Laboratory Technical Report No.1, Series B, 1949.

In the test sections of water tunnels, bubbles are sometimes formed and must be resorbed by the water before it is recirculated since the air content must be constant. An experimental apparatus was built which was able to produce these bubbles under various pressures, temperatures, and velocities and the resorption of the bubbles was measured.

The equation

$$T = \frac{L}{K_1 \beta D \left(1 - \alpha \frac{C_g}{C_G}\right) R_e{}^n} R_1$$

was proposed and verified as the basic equation governing gas bubble resorption in turbulent liquid.

In this equation T = time required for complete resorption of gas bubbles (in seconds) of initial radius R_1 (in feet), β = solubility in gas volume per unit volume of liquid at a temperature t and pressure p., α = relative saturation of the liquid (C_L/C_G), C = concentration of gas dissolved in liquid (lb/ft^3), C_G = C if liquid is saturated, C_L = actual value of C in interior of liquid, C_g = difference between C at edge of sublayer and C_L, L = linear dimension measuring scale of the turbulence, taken as the boundary dimension of R_e, R_e = Reynolds number, D = specific coefficient of diffusion (about 1.95×10^{-8} $ft^3/sec.$ for air and water). The constants K_1 and n were evaluated as 0.0162 and 1.25 respectively and $C_g/C_G = 0.18$ at high turbulence.

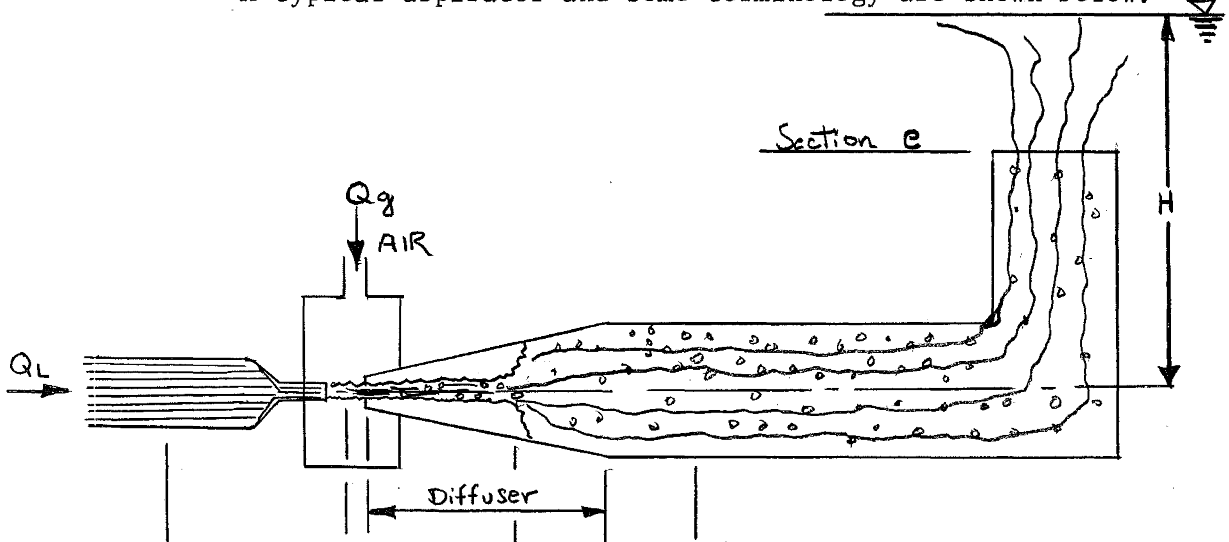
Mention was made of an idea for removing bubbles in the downstream section of the tunnel. It was suggested that bubbles be bled off by external suction applied at slots on the convex surface of the turning vanes downstream from the diffuser. Preliminary tests indicated that about 5% of the flow would have to be sucked off to withdraw a very large part (but not all) of the bubbles. This idea, however, in the authors words "was left dormant".

Silberman, E., and Ross, J. A., "GENERATION OF AIR-WATER MIXTURES IN CLOSED CONDUITS BY ASPIRATION", St. Anthony Falls Hydraulic Laboratory Report No. 43, University of Minnesota, Minneapolis, Minn.

There are several methods by which air-water mixtures can be produced in closed conduits. These are: 1. Aspirators 2. Orifices 3. Effervescence or Chemical Means. This paper is concerned only with aspirators.

Theory of the Aspirator

A typical aspirator and some terminology are shown below.



a	t	x	b	Section
Aa	Aj	Ax	Ab	Area
Pa	Pg	Pg	Pb	Pressure
0	βg	βg	βb	Gas-Liquid Ratio

The basic concept of the aspirator is the occurrence of a sudden pressure rise in the diffuser, at the point where the liquid and the water jets unite, shown as (x) on the drawing. The expansion is similar to the hydraulic jump in open channel flow and occurs for the same reason, namely, to overcome a discontinuity in pressure. Aspirators can be constructed with or without the diffuser section.

In the diffuser section, kinetic energy is converted to pressure energy and this is accompanied by turbulence. This turbulence entrains the gas bubbles.

It is assumed that the change from P_g to P_b is rapid enough to count on adiabatic process, so that;

$$\beta_b = \left(\frac{Pg}{Pb}\right)^{1/\gamma} \beta_g \qquad (14)$$

where γ = adiabatic exponent. The remainder of the terms are defined on the drawing.

Assuming no friction in the nozzle, and applying the energy equation, it can be shown that;

$$P_a = P_g + q_j \left[1 - \left(\frac{Aj}{Aa}\right)^2 \right] \qquad (15)$$

where $q = \left(\frac{\rho_L}{2}\right)\left(\frac{Q_L}{A}\right)^2$, the dynamic pressure of the liquid, the subscript indicating the location.

When no diffuser is ued on the aspirator, or if the expansion occurs beyond the end of the diffuser, it can be shown that:

$$\beta_b = \frac{Ab}{Aj} - \frac{Pb - Pg}{2q\ b} - 1 \qquad (16)$$

It is assumed that the pressure loss in the conduit is proportional to the dynamic pressure of the flowing mixture and the proportionality constant is independent of the fluid, which yields the basic equation for the conduit downstream from the aspirator. This is stated as;

$$\frac{Pb - Pg}{qb} = \frac{Pe - Pg}{qb} + N + \frac{1}{F} + \beta_b \left(N\frac{\beta g}{\beta b} - 1 + \frac{\beta e}{\beta b} - \frac{1}{F}\,\frac{\beta_z/\beta b}{1 + \beta_z}\right) \qquad (17)$$

where F = Froude Number and N = pressure loss coefficient, which includes exit losses and losses due to valves.

In a straight, uniform conduit,

$$N = fL/D \tag{18}$$

where L = length, D = diameter, f= friction factor. When valves etc. are included, N is not given by a simple expression nor is it entirely dependent on the Reynolds Number. For that reason, N is experimentally determined by pumping only liquid through the conduit ($\beta = 0$).

If $\beta = 0$, at incipient gas flow, eq. 16 reduces to

$$P_{bo} = P_e + N_{qbo} + \rho_L g H \tag{19}$$

where the subscript "o" diesignates incipient gas flow.

Combining eqs. (17) and (18), we see that;

$$\frac{P_b - P_{bo}}{q_b - q_{bo}} = N + K \tag{20}$$

which defines a new loss coefficient, N + K. This coefficient may be applied to the liquid flow only, regardless of what the gas flow might be. It can be shown, however, that the proportionality depends almost entirely on N.

The bubble size is analyzed assuming the following relationship;

$$\frac{d}{\sqrt{Aj}} = \phi\left[(q_b - q_{bo}), \frac{Ab}{Aj}, N\right] \tag{21}$$

No attempt is made to evauate the coefficient ϕ.

Experimental Work

Measurements were made on two aspirators, one with $A_b = A_a = 4$ inches, and $A_b/A_j = 3.9$; and the other with $A_b = A_a = 2\text{-}1/2$ inches and $A_b/A_j = 4.5$.

The bubbly flow was separated into two parts, one part going through a transparent viewing section.

Water flow was measured with a nozzle prior to the aspirator. Air flow was also measured with a nozzle right at the aspirator.

The bubbles were photographed through the test section and the negatives projected on a screen for study.

<u>Experimental Results</u>

<u>Aspirator Performance</u>; Equation (16) was plotted with β_b vs. $\frac{Pb - Pg}{2qb}$ (back pressure parameter). Theoretically, this should be a straight line as long as the expansion is not in the diffuser. The results verify the theory very well. It appears practicable to design all aspirators without diffusers.

<u>Effect of Downstream Conduit</u>: From Equation (20) it is seen that the proportionality $\frac{Pb - Pbo}{qb - qbo}$ is nearly equal to N. Since (qb-qbo) is proportional to $(Q_L^2 - Q_{Lo}^2)$ the data were plotted as Pb - Pe vs. Q_L^2. This gave a straight line, the slope of which was N. The data verified this relationship for aspirators with and without diffusers. The values for K were seen to be small as assumed and were neglected. This verifies the assumption that the pressure loss coefficient should be independent of the fluid used.

<u>Bubble Size</u>:

The bubble size seemed to be nearly independent of the jet diameter. A detergent was added to the water at one point and the bubble diameters were again determined. The bubble diameters were found to be smaller, showing that the bubbles are influenced by the physical properties of the liquid or gas.

Principal Conclusions

(1) Aspirator type air-water mixture generators are acceptable for laboratory work.

(2) Aspirators should be constructed without diffusers, both to save in cost and to facilitate design.

(3) Aspirators without diffusers may be designed using Equations (14), (15), and (16). These equations apply to any gas or liquid.

(4) Bubble size is not directly predictable for a given aspirator.

Soo, S. L., and Regalbuto, J. A., "CONCENTRATION DISTRIBUTION IN TWO-PHASE PIPE FLOW", Canadian Journal of Chemical Engineering, Vol. 38 n 5, Oct. 1960, pp. 160-166.

Experiments were carried out by the authors in an effort to determine the manner in which the concentration of solid particles varies across the diameter of a pipe filled with flowing gas. Thus, the two phases were the solid particles and the gas. The results obtained are only ture for fully turbulent flow and for such a small number of solid particles that gravity effects could be disregarded.

In the experimental approach to the problem, small glass beads were introduced into a 3 inch pipe carrying turbulent flowing air. Fifteen feet downstream the concentration of the solid particles along points across the diameter were measured by means of 0.062" probe which was moved along the diameter, and various electronic particle counters.

Except for the effects of gravity (which can be very large for heavier solid particles or larger amount of particles) the experimental results correspond reasonably to the results of a theoretical investigation; which, after many assumptions and mathematical operations boils down to:

$$C = C_o J_o (r \sqrt{f/D_u})$$

where

C = conc. of solid particles at any point along the diameter

C_o = conc. at the center of the pipe

J_o = the Bessel function of the first kind and of zero order

r = distance from pipe center to the point where C is required

f = acceleration due to transport of particles across the turbulent velocity field of the stream. Thus f relates the fluid drag and inertia of solids and is, in general, a function of state parameters of the system and radial coordinate, r.

D = the particle diffusivity (due to stream diffusivity)

u = mean velocity of the stream

The important parameter to be derived from the equation on the preceding page is:

fR^2/Du where R = pipe radius

The greater this parameter (e. g. - the greater u becomes) the more uniform the particle concentration.

The authors condlude that for cases where gravitational effects are very small (small particles and light loading) the concentration distribution approximates the turbulent velocity profile across the pipe cross-section. Also, the assumption of a constant concentration distribution appears to be valid for the conditions where gravity effects are small and the mean stream velocity is great.

APPENDIX B

ANNOTATIONS

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Alves, George, E., "CONCURRENT LIQUID-GAS FLOW IN A PIPE-LINE CONTACTOR", Chemical Engineering Progress, V 50, n9, Sept. 1954, p. 449-456.

This report contains the results of an experimental investigation on the isothermal flow of water-air and oil-air mixtures in a 1-in. con-current pipe-line contactor. The investigation was undertaken to provide basic data on flow pattern, pressure drop, and liquid holdup for the flow of liquid-gas mixtures in this type of equipment. Pressure-drop data for the flow of liquid-gas mixtures have been extended to include the return bends and inlet mixing tee. It is interesting to note that the flow through the pipe is finally separated by a cyclone type of separator.

Begell, W., and Hoopes, J. W., "ACCELERATION OF PRESSURE DROPS IN TWO-PHASE FLOW", Atomic Energy Commission Tech. Inf. Service CU-18-54-At-dP-Ch. E., April 1954.

Two-phase water-steam mixtures are used at high temperatures to produce nomographs for the rapid computation of the parameters used in the Lockhart-Martinelli correlation. The lowest temperature used is over 100° F and therefore the graphs are of little value in connection with the current project, for high project, for high pressure and temperature two-phase flows, however, this paper may be very valuable.

Bergelin, O. P., and Gazley, C., Jr., "CO-CURRENT GAS-LIQUID FLOW I. FLOW IN HORIZONTAL TUBES", Heat Transfer and Fluid Mechanics Institute, Proceedings of 2nd Conf. 1949.

The authors examine the applicability of the Lockhart-Martinelli correlation to stratified flow and found that the correlation gives a conservative estimation of pressure drop. No new usable method, however, is presented to correct for the discrepency.

Bergelin, O. P., Kegel, P. K., Carpenter, F. G., and Gazley, C., Jr. "CO-CURRENT GAS-LIQUID FLOW II. FLOW IN VERTICAL TUBES", Heat Transfer and Fluid Mechanics Institute Proceedings of 2nd Conf. 1949.

The Martinelli X - ϕ correlation was used to predict pressure drops in vertical flow and produced favorable results, ($\pm$30%). The authors point out that in their opinion, even though fair results were obtained, not all the important parameters are included when the correlation is applied to vertical flow.

Chenoweth, J. M., and Martin, M. W. "PRESSURE DROP OF GAS-LIQUID MIXTURES IN HORIZONTAL PIPES", <u>The Petroleum Engineer</u>, Vol. 28, n4, April 1956, p. C42-C45.

This investigation was undertaken to check the Lockhart and Martinelli correlation with pressure-drop data for two-phase flow in larger pipes and at higher pressures.

Chrisholm, D., "THE FLOW OF STEAM/WATER MIXTURES THROUGH SHARP-EDGE ORIFICES", Eng. and Boiler House Review, Vol. 73; p. 253, 1953.

Steam-water mixtures were used to find the flow rates for two-component flow through orifices.

The equation

$$\frac{W_{LN}}{W_L} - \frac{1}{C_L} = 2.1Y^{0.825}$$

where

$$Y = \frac{g}{1-g} \frac{W_{LN}}{W_{VN}}$$

g = dryness fraction

W_L = liquid weight flow rate during two-phase flow

W_{LN} = liquid weight flow rate during single phase flow

W_V = vapor weight flow rate during two-phase flow

W_{VN} = vapor weight flow rate during single phase flow

C_L = liquid contraction coefficient

has been shown to predict flow rates within ± 10% of experimental values for flow through sharp edged orifices.

Erickson, O. P., "LATEST DREDGING PRACTICE", Paper No. 3281, ASCE Transactions, Vol. 127, Part IV, 1962.

Automatic Relief Valve: There has become available to dredge contractors an automatic relief valve which reduces "choke off" from high vacuum and greatly reduces water hammer in pump and pipelines which so often proves costly in high pressure dredging. An increase in production of approximately 5% can be obtained by using this valve. The valve is also beneficial when gaseous materials are pumped, although in extremely gaseous pumping, a gas ejector should also be installed for top efficiency.

Erwin, R. W., "NEW VACUUM-TYPE DEGASSER REDUCES DEEP DRILLING HAZARDS", World Oil, Vol. 136, June 1953, p. 125, 127, and 130.

A new cascade-vacuum type degasser with a 25-inch mercury rating, controls and balances mud weight to prevent the shift from lost circulation to blow out conditions, normally a deep drilling hazard. The mud from the well is cascaded over baffles and subjected to high vacuum to draw off entrained gases. The author states that the gaseous fluffiness is eliminated from muds which enables the mud to be handled more easily by pumps.

Fried, Lawrence, "PRESSURE DROP AND HEAT TRANSFER FOR TWO-PHASE, TWO-COMPONENT FLOW", Chemical Engineering Progress Symposium Series No. 9, Vol. 50, 1954, p. 47.

When two-phase flow is in the isothermal condition, the Martinelli Φ - X correlation can be used to measure pressure drop. It was found that for non-isothermal flow in which the mixture is heated in the test section, the values of Φ are considerably higher. This is due to the kinetic energy changes. When it was corrected for these changes, the Martinelli correlation was found applicable.

Galegar, W. C., Stovall, W. B., and Huntington, R. L., "REPORT ON TWO-PHASE VERTICAL FLOW", Pipe Line Industry, Vol. 4, n2, Feb. 1956, p. 38-40.

This investigation presents experimental data on the performance of kerosene-air and water-air systems in two-phase vertical upward flow using two test sections of different size but having the same ratio of diameter to height. A tentative correlation is presented in conjunction with visual data on the flow patterns involved.

Gazley, C., Jr., "CO-CURRENT GAS-LIQUID FLOW; III. INTERFACIAL SHEAR AND STABILITY", Heat Transfer and Fluid Mechanics Institute, Proceedings of 2nd Conf. 1945.

A theoretical approach to two-phase flow is presented by using energy losses and transfers at fluid-fluid interfaces to evaluate interfacial shear and stability. It is found that the formation of interfacial waves is dependent essentially on the liquid depth and the relative velocities of each phase. It was found that a relative velocity of ten to fifteen feet per second is required for the formation of waves.

Graf, W. H., "INVESTIGATION ON A TWO-PHASE PROBLEM IN CLOSED PIPES", University of California Hydraulic Engineering Laboratory, Technical Report HEL-2-2, Berkely, California, 1962.

The two-phase flow considered in this report deals with sand-water mixtures. Two measuring devices were tested, the Loop system and the Venturi meter. The Loop system consisted of two identical vertical pipes with opposite flow directions. The summation of the head differences would determine the flow rate while their difference would determine the specific weight and thereby the concentration. The Venturi meter proved to be more awkward in application but more practical in shape. The flow rate and concentration have to be found by trial and error.

Johnson, H. A., and Abov-Sabe, A. H., "HEAT TRANSFER AND PRESSURE DROP FOR TURBULENT FLOW OF AIR-WATER MIXTURES IN A HORIZONTAL PIPE", Transactions ASME, Vol. 74, 997, 1952.

The static pressure drop and heat transfer for two-phase, two-component flow of air and water was measured on a 15 foot horizontal pipe. The Martinelli method was used in predicting the results and $\pm 30\%$ accuracy was obtained.

Kalinske, A. A., and Bliss, P. H., "REMOVAL OF AIR FROM PIPE LINES BY FLOWING WATER", Civil Engineering, Vol. 13, 1943, pp. 480-483.

Entrained air in pipelines becomes trapped at the summits and must be removed with the use of escape valves. Since these valves require periodic maintenence it would be more economical if the air could be removed without the use of a valve.

It was found that a hydraulic jump will form at the downstream end of an air pocket and the jump will carry the air off in the form of bubbles. Various slipes of pipe and discharges were studies and a dimensionless plot was drawn which could be used in the design of pipe lines in order to eliminate the escape valves.

Kalinske, A. A., and Robertson, J. M., "AIR ENTRAINMENT; CLOSED CONDUIT FLOW", Transactions ASCE, 1943, p. 1435.

When an air pocket is formed at the summit of a pipeline, the conduit flows only partially full. This air pocket can be removed either by a relief valve or by a hydraulic jump which follows the air pocket. This paper deals with the ability of the jump to carry off the air in the form of bubbles.

A long transparent tube was constructed which was so arranged that the slope could be varied from horizontal to 30 per cent. Entrance and exit structures controlled the flow so that the depth and velocity could be controlled separately. It was found that at some slope and depth, there was a critical water discharge at which the jump could no longer carry all the air being pumped in. This critical condition, for any slope of the pipe and for any relative depth of flow in the air pocket, depends on the value of the Froude number of the flow ahead of the jump. Below the critical value the flow beyond the jump will not be able to handle all the air entrained by the jump and thus the air removed will not be a function of the jump characteristics but rather on the hydraulic features of the flow beyond the jump.

In other words, the rate of air entrainment by a hydraulic jump depends largely on the water discharge and the Froude number of the flow ahead of the jump.

Lorenz, A., "NEW DESIGN OF MECHANICAL VACUUM PUMPS", American Vacuum Society Symposium-Transactions, 1958.

The most widely used vacuum pumps are oil-sealed rotary piston pumps of the sliding vane type. There are various designs for low and medium capacities. For larger capacities, pumps of the rotary plunger type are generally preferred because their inherently sturdy construction insures reliable operation even under adverse conditions.

Until recently, a serious disadvantage of rotary plunger pumps was the fact that they were frequently unbalanced, and consequently, vibrations have been a problem. Recently, however, a new design has been developed which has completely eliminated this problem. By mounting two different sizes of pistons, along with their eccentrics, and the drive gear whose mass is distributed unsymethically, all on the same drive shaft, the system consisting of the three differently distributed masses can be so arranged and adjusted so that vibrationless operation is obtained in the rotary plunger type.

Models with capacities ranging from 13 to 105 cfm have already been built according to this principle of design.

Levy, S., "THEORY OF PRESSURE DROP AND HEAT TRANSFER FOR ANNULAR STEADY STATE TWO-PHASE TWO-COMPONENT FLOW IN PIPES", Ohio State University Eng. Exp. Sta. Bulletin #149 Proc. of Second Midwest Conference on Fluid Mech., 1952.

The author presents a theoretical approach to the two-phase problem and correlates his results with those of Martinelli, obtaining fair agreement ($\pm 20\%$). He also justified the independence of Φ and R from X. (X = two-phase flow modulus, R = fraction of tube filled, % = gas or liquid, Φ = dimensionless two-phase pressure drop).

Marks, Robert H., "VACUUM DEGASIFICATION SHOWS ITS VERSATILITY" Power Engineering, Vol. 59, n8, August 1955, p. 92-95.

The types of degasifiers described are basically packed columns used to carry out mass transfer between liquid and gas under vacuum conditions. The article included applications of a vacuum degasifier in treatment of: chemical plant cooling water, water for oil field flooding operations, boiler feedwater makeup for a public utility power plant, and cosmotron cooling water.

Martinelli, R. C., Putnam, J. A., and Lockhart, R. W., "TWO-PHASE, TWO-COMPONENT FLOW IN THE VISCOUS REGION", Transactions A. I. Ch. E., Vol. 42, p. 681, 1946.

This paper is a supplement to the paper presented by Martinelli et al in the 1944 Transactions of A. S. M. E. in which the Φ - X correlation was first discussed and experimental data was presented for conditions where the flow was fully turbulent (Reynolds Number $>$ 2000 for both gas and liquid). Experimental data for fully viscous flow is now presented and the Φ - X correlation is again used.

The angle of the test section was variable and it was found that the pressure drop in fully viscous flow is now presented and the Φ - X correlation is again used.

The angle of the test section was variable and it was found that the pressure drop in fully viscous flow is independent of the angle of the tube. This conclusion, however, "should not be extrapolated to other systems until further experiments are performed".

McAdams, W. H., Woods, W. K., and Bryan, R. L., "VAPORIZATION INSIDE HORIZONTAL TUBES", Transactions ASME, 1948, p. 545.

This paper reports an investigation carried out to determine the changes in the coefficient of heat transfer for the evaporation of a liquid flowing inside a heated horizontal tube. The temperature differences used would be too great for the paper to be of any value in this project. The authors state, however, that in subsequent papers they will describe their method of obtaining the pressure drop, which would be of interest. This paper would then serve to more fully explain the subsequent papers.

Peterson, W. P., "TABLES OF FLOW PROPERTIES OF THERMALLY PERFECT CARBON DIOXIDE AND NITROGEN MIXTURES", Ames Research Center, Moffett Field, California. NASA SP-3009, 1964.

This report presents equations, tables, and figures for use in the analysis of flow of carbon dioxide and carbon dioxide and nitrogen mistures. The analysis is restricted to condition at which the gas can be assumed thermally perfect. Calculations have been made for three mixtures: 100 per cent CO_2; 50 per cent CO_2; and 90 per cent N_2. The tables which might be applicable to the problem include the gas mixture properties given as functions of temperature.

Moore, D. W., "THE RISE OF A GAS BUBBLE IN A VISCOUS LIQUID", Journal of Fluid Mechanics, Vol. 6, Part I, 1959.

The rise of a gas bubble in viscous liquids and at high Reynolds Numbers is theoretically analyzed. It is shown that the drag coefficient of a spherical bubble is 32/R where R is the Reynolds number (based on diameter) of the motion of the rising bubble. Equating the drag force to the bouyant force of the bubble; the bubble diameter and velocity can be computed.

Mathematically extending his analysis to non-spherical bubbles, the author has developed similar expressions for non-spherical bubbles. Unfortunately, the author has not been able to compare these results with direct experimental observations..

Reid, John, L., "VAPOR-LIQUID SEPARATORS", Plant Engineering, Vol. 10, n7, July 1956, p. 106-107, 182, and 184.

A wire mesh separator is used mainly for obtaining a desirable vapor rather than liquid. The principle is this: As the vapor disengages from the liquid, it carries with it fine liquid droplets. When the vapor stream passes through the fine wire mesh, the liquid droplets impinge on the wire surfaces, coalesce into large drops, and fall off. The vapor is now dry and free from entrained liquid.

Ripken, J. F., "DESIGN STUDIES FOR A CLOSED JET WATER TUNNEL", St. Anthony Falls Hydraulic Laboratory Technical Report No. 9, Series B, 1951.

Extensive studies on a model water tunnel are presented. The problem of air entrainment, however, is dismissed by the statement that "no special provision for air content control was made in the basic flow circuit except for three air collection domes to assist in pre-test bubble removal".

VanWingen, N., "PRESSURE DROP FOR OIL-GAS MIXTURES IN HORIZONTAL FLOW LINES", World Oil, October 1949, p. 156.

The author adapts the method of Martinelli and co-workers to the problems of transporting oil and gas from a well to a tank farm. Pressure drops are taken for a number of oil lines and this data is compared to the expected results using the Martinelli method. Generally the results are acceptable, even though considerable scatter is present. This scatter, however, can be attributed to the poor control of variables such as temperature, viscosity, etc.

Weiss, D. H., "PRESSURE DROP IN TWO-PHASE FLOW", Argonne National Laboratory 4916, October 1952.

Pressure drop was determined for high pressures and temperatures. The range of pressures was from 20 to 1400 psia and heat fluxes from 100,000 to 500,000 Btu/hr/lb^2. The Martinelli method of prediction of pressure drops was used and resulted in fairly close agreement. ($\pm$30%).

Wenberg, H. B., "NICOLET'S EXPERIENCE WITH THE VORVAC SYSTEM", Paper Trade Journal Reprint obtained from Nichols Engineering and Research Corp., New York.

The author is a manager of a paper manufacturing concern and relates his experiences using the Vorvac system. He claims almost complete absence of air bubbles in the manufactured paper and attributes this to the installation of the Vorvac system.

White, Philip, D., and Huntington, R. L., "HORIZONTAL CO-CURRENT TWO-PHASE FLOW OF FLUIDS IN PIPE LINES", The Petroleum Engineer, Vol. 27, August 1955, p. 40-45.

This research project was under taken to study the visually observed flow type in general and pressure drops resulting from the stable type of two-phase flow over a wide range of variables. A tentative correlation is presented that shows the type of flow that would exist under a given set of mass flow rates. An empirical correlation is presented, which allows the two-phase pressure drop to be predicted if the flow rates, physical properties, and pipe diameter are known. Also a tentative correlation is presented to predict pressure drops resulting from ripple type flow. The Martinelli method is used and found to be fairly accurate.

Zmola, P. C., Bailey, R. V., Taylor, F. M., and Planchet, R. J., "TRANSPORT OF GASES THROUGH LIQUID-GAS MIXTURES", Oak Ridge National Laboratory and Tulane University Publication, December 1955.

A vertical column of liquid was used and air was bubbled through the liquid. The flow rate of air and the detention time in the liquid was measured. It was found that even in still liquids the gas would first be in bubble form and as the gas flow is increased, would form slugs. This same phenomenon is also noticed in horizontal tubes.

STANDARDS FOR STEAM JET EJECTORS, 3rd EDITION 1956, HEAT EXCHANGE INSTITUTE, New York.

Describes in detail the different types of ejectors along with methods and materials of their construction. Essential design information is given in the way of graphs, data, and examples. Standard methods of operation and basic specifications for performance tests are explained.

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Department of Civil Engineering
FRITZ ENGINEERING LABORATORY

HYDRAULICS DIVISION

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