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# Performance of Small Herschel-Type Venturi Tubes

P. L. WALKER, JR., R. J. FORESTI, JR., F. RUSINKO, JR., and B. W. McCORMICK
The Pennsylvania State University, State College, Pennsylvania

**The characteristics of five Herschel-type Venturi tubes with throat diameters ranging from 0.073 to 0.359 in. and throat-to-pipe—diameter ratios from 0.0882 to 0.4340 have been investigated.**

Research on the performance of Venturis with small throats and low throat-to-pipe–diameter ratios has been practically ignored to the present. Their high value of frictional resistance and the low throat pressure produced, which can cause cavitation of the liquid, have been disadvantages which have kept these Venturis from being utilized as flow-measuring devices; however, recent research by two of the authors(*6*), among others, on the possibility of using the low throat pressure of the small Venturi to induce a secondary liquid stream (that is, to use the Venturi as a proportioner) pointed to the need for information on these Venturis. In this work the exhaustive data available on throat-to-pipe—diameter ratios above 0.25 have been extended down to a value of 0.0882. In addition, the performance of the Venturi at unusual operating conditions has been investigated with the pressure recovery being varied from close to 0 to 100% (the Venturi as a metering device has a pressure recovery of roughly 80%) in an attempt to determine the effect on flow rate and Venturi throat pressure, both important parameters if the Venturi is to be used as a proportioning device.

## APPARATUS

The standard Venturi design discussed by Jorissen(*3, 4*) and presented in the British Standard Code(*1*) was adhered to in the construction. Figure 1 shows a drawing of the largest Venturi used. All Venturis utilized an entrance cone with an included angle of 21° and an exit cone with an angle of 7°. The length of the Venturi throat in all cases was nearly equal to the Venturi-throat diameter. At the middle of the throat a 1/32-in. pressure tap was located for pressure readings. A collector piezometer ring was not employed since for these small throats it was feared that too many pressure taps might severely distort the flow through the throat. The Venturis were made of Lucite acrylic resin so that flow conditions might be observed and pictures taken. The resin was very easily worked and was finished to a high polish. Sections of 17ST aluminum alloy were connected to the Venturi with hand tightening of the Venturi proper to these sections made possible by 1-in. machine threads. Extreme care was taken in machining to assure a smooth fit between these sections.

The manner in which the exit aluminum section was connected to the standard 3/4-in. (I.D. 53/64 in.) iron pipe is shown in Figure 2. The entrance section employed a similar connection. To join the Venturi unit smoothly to the pipe with no irregularities or surface roughness, a steel sleeve was soldered to both the entering and exit pipe and the internal surfaces were carefully polished. A good connection was then achieved by tightening the large hexagonal nut against a

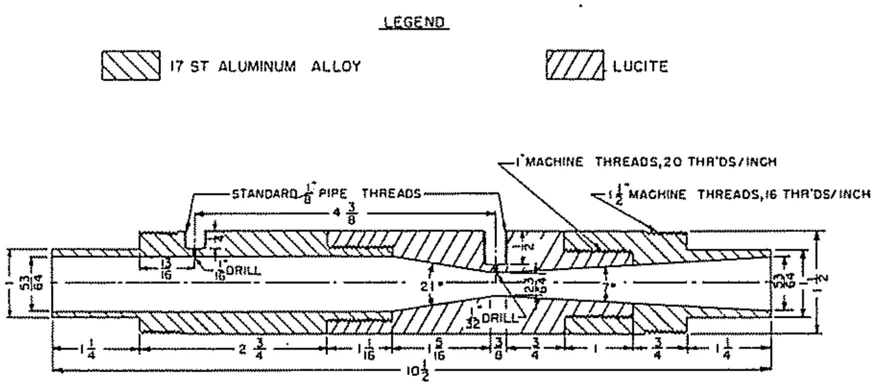


FIG. 1. VENTURI-METER DESIGN.

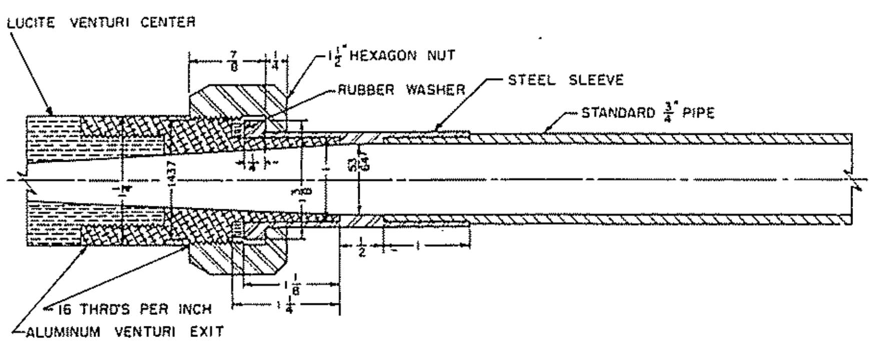


FIG. 2. CONNECTION DESIGN OF PIPING TO VENTURI METER.

R. J. Foresti, Jr., is at present with Monsanto Chemical Company, Dayton, Ohio.

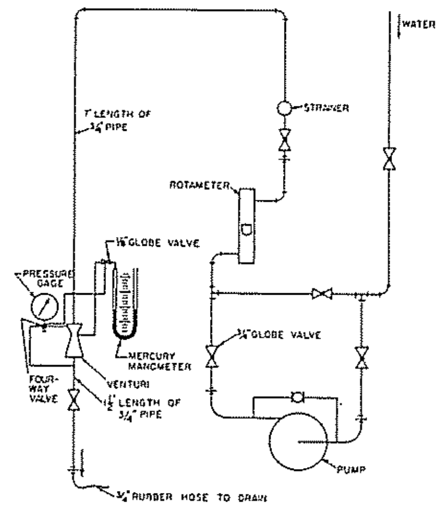


FIG. 3. VENTURI-METER FLOW SHEET.

rubber gasket. For most of the work only hand tightening was necessary.

Figure 3 shows the over-all flow sheet of the apparatus. An Aurora Pump Company pump with a capacity of 8.0 gal./min. at 275 lb./sq. in. gauge was used in the work. With the by-pass system shown, flow rates could be easily adjusted and maintained. A high-pressure rotameter with a flow-rate range of 0.5 to 11.5 gal./min. was used, on which flow rates could be read to the nearest 0.1 gal. A strainer was placed in the line to remove any foreign matter which might conceivably lodge in the Venturi throat. Preceding the Venturi, a 7-ft. vertical section (100 pipe diameters) of new galvanized-iron pipe assured a well-developed flow pattern leading into the Venturi. The upstream pressure tap was in all cases a 1/16-in. hole located 3 1/2 pipe diameters ahead of the entrance cone. The pressure drop between this tap and the entrance to the Venturi tube was negligible. The downstream pressure tap was a 1/16-in. hole drilled in the 3/4-in. iron pipe and located 7 pipe diameters after the exit cone. A 3/4-in. globe valve was located 22 pipe diameters downstream from the exit cone and was used to vary the back or exit pressure of the Venturis.

Pressure readings at the Venturi throat were made on a 32-in. Hg manometer where possible. The inlet and outlet pressures, as well as high throat pressures, were read on a 6 3/4-in. face, 300 lb./sq.in. Crosby gauge.

The water used had an average temperature of 50°F.; it contained 8 p.p.m. of oxygen, 13 p.p.m. of nitrogen, 0.3 p.p.m. of chlorine, and negligible amounts of carbon dioxide in the form of dissolved gases.

## EXPERIMENTAL PROCEDURE

The data were taken at selected values of inlet pressure, at which the outlet pressure was varied to produce different flow rates. At each condition, measurements were taken of the inlet, outlet, and throat pressure and of the flow rate. For each Venturi, data were obtained up to the maximum capacity of the pump, that being the pressure in the case of the small Venturi and flow rate for the larger Venturis. Duplicate runs gave excellent reproducibility of the data.

## ANALYSIS OF RESULTS*

**Discharge Coefficients.** At each condition of inlet and outlet pressures, discharge coefficients and pipe Reynolds numbers were calculated for the five Venturis. The discharge coefficient $C$ is defined by

$$Q = C \frac{A_t}{\sqrt{1-\beta^4}} \sqrt{\frac{2\,(p_i - p_t)}{\rho}} \qquad (1)$$

The $C$ is a measure of the deviation of throat velocity from the ideal-fluid case. In purely potential flow the value of $C$ would be unity.

The calculated variation of the discharge coefficients with the pipe Reynolds number for the different Venturis is presented in Figure 4. An appreciable decrease in the discharge coefficient for the smallest Venturi as the Reynolds number decreased below 10,000 is observed. As is evident, there is considerable scatter in the data for the larger Venturis, particularly at the lower Reynolds numbers. The reasons for this are not entirely clear as the data were reproducible; however, it was observed that for the largest Venturi the pipe was not completely filled with water at the lower flow rates. In view of the experimental scatter, the major contribution of this paper is felt to be in presenting and analyzing the data obtained on the effects of cavitation and operation at extremely low Reynolds numbers and not in presenting quantitative values of the discharge coefficient in the Venturis. It is unfortunate that the limitations of the test equipment prevented operation of the smallest Venturi in the same range of Reynolds numbers as the larger Venturis.

The effect of Reynolds number appears to be conveniently presented as a "coefficient of correction," as given by Jorissen (*3*). This coefficient is the ratio of the discharge coefficient at any Reynolds number to the discharge coefficient at a relatively high Reynolds number (greater than 300,000) above which the discharge coefficient remains constant. According to Jorissen, the coefficient of correction does not depend to any extent upon the throat-pipe–diameter ratio. This coefficient was given by Jorissen down to a pipe Reynolds number of 10,000. By use of the curve of the discharge coefficient for the small Venturi as a function of

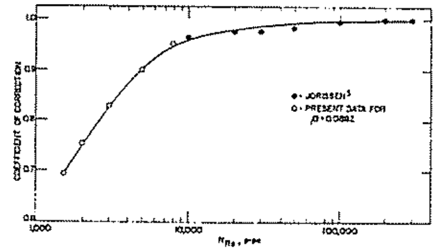

FIG. 5. COEFFICIENT OF CORRECTION AS A FUNCTION OF PIPE REYNOLDS NUMBER.

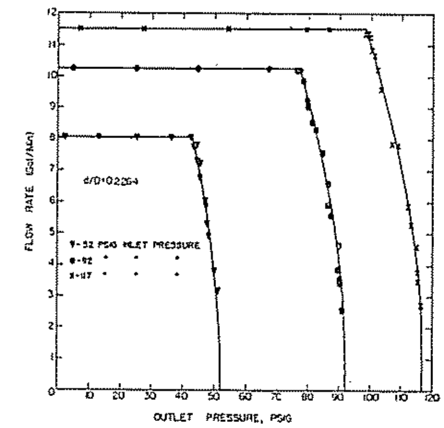


FIG. 6. EFFECT OF OUTLET PRESSURE ON WATER FLOW RATE THROUGH A VENTURI 3/16-IN.-DIAM. THROAT.

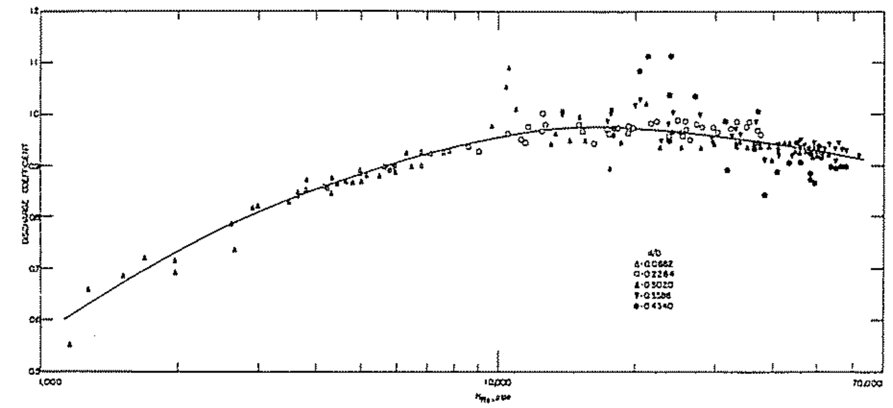

FIG. 4. DISCHARGE COEFFICIENT AS A FUNCTION OF PIPE REYNOLDS NUMBER FOR A SERIES OF VENTURIS.

*The complete data are available as document 4477 from the A.D.I. Auxiliary Publications Project, Photoduplication Service, Library of Congress, Washington 25, D.C., at \$1.25 for photoprints and \$1.25 for 35-mm. microfilm.

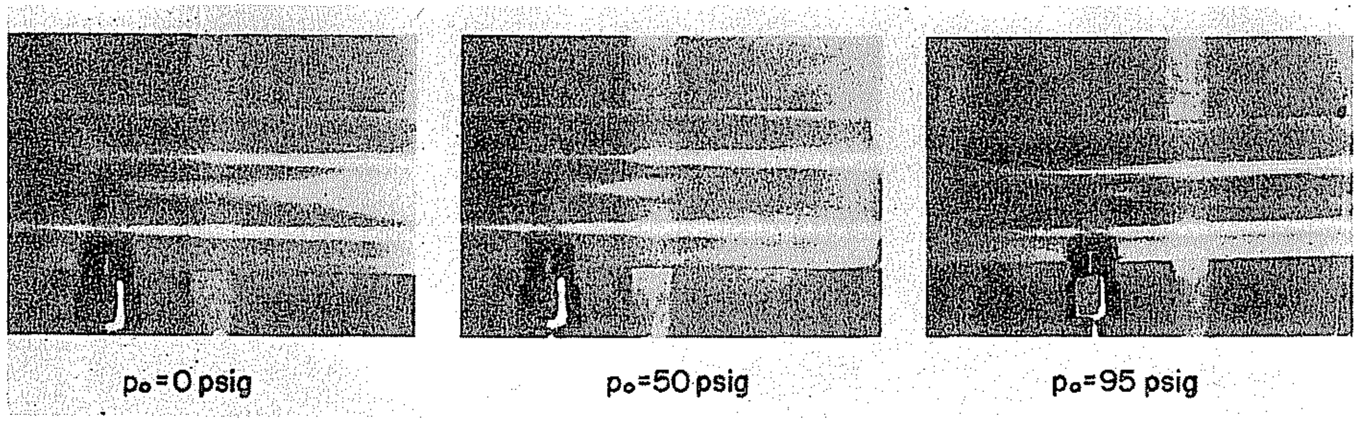


FIG. 7. EFFECT OF INCREASING OUTLET PRESSURE ON THE DEGREE OF CAVITATION IN VENTURI DIFFUSER: $\beta = 0.2264$, $p_i = 117$ LB./SQ. IN. GAUGE, $Q = 11.5$ GAL./MIN.

Reynolds number, Jorissen's coefficients of correction have been extended down to a Reynolds number of 1,500, as shown in Figure 5. The asymptotic value of the discharge coefficient was chosen so as to give the same value of the correction coefficient at a Reynolds number of 10,000 as was reported by Jorissen. It can be seen from Figure 5 that the effect of Reynolds number on the discharge coefficient becomes quite appreciable for Reynolds numbers much below 10,000. Whether or not this curve for the correction coefficient is independent of $\beta$ for relatively low Reynolds numbers unfortunately cannot be determined by the data presented here.

**Cavitation Inception.** In addition to the discharge coefficients, considerable information was obtained on the effect of cavitation on the performance of the Venturis. For a fixed-inlet pressure, as the outlet pressure was reduced, a pressure and velocity would be reached at which cavitation would begin immediately following the throat. Any further reduction in outlet pressure produced no increase in the flow rate or the measured value of the throat pressure.

The occurrence of cavitation in the Venturi is closely linked with its performance. Figure 6 shows a typical variation of flow rate with outlet pressure for different inlet pressures with a $\beta$ of 0.2264 for the Venturi. It can be seen that at a specific inlet pressure there is an outlet pressure below which the flow rate remains constant. This specific inlet pressure also represents the point of inception of cavitation; that is, as the inlet pressure is continually increased from zero to its critical value, the degree of cavitation in the diffuser section of the Venturi decreases. Upon reaching the critical outlet pressure for a constant inlet pressure, cavitation stops. Figure 7 clearly shows this phenomenon for the foregoing Venturi operating at a constant inlet pressure of 117 lb./sq. in. gauge. The three photographs from left to right represent 0, 50, and 95 lb./sq. in. gauge outlet pressures, respectively. When the "critical" outlet pressure of 98 lb./sq. in. gauge is reached, no cavitation is seen.

It is usually desirable to avoid cavitation in a Venturi for at least several reasons. First is the fact that the loss in total head across the Venturi increases considerably with cavitation present. This is obvious from Figure 6, which shows an appreciable decrease in the outlet pressure with no change in the inlet pressure and flow rate once cavitation occurs. Second, cavitation is undesirable from the standpoint of its accompanying noise and vibration. It is therefore expedient to predict when cavitation in a given Venturi will occur or, given a set of flow conditions in a pipe, to select the Venturi which will produce the largest differential pressure across the inlet nozzle without cavitating.

Cavitation is conveniently treated in terms of a parameter denoted as the cavitation index, $\sigma$, defined by (*5*):

$$\sigma = \frac{p_i - p_c}{1/2\ \rho\ (V_p)^2} \qquad (2)$$

This parameter can be defined in the general sense as an operating index; however, for a given geometrical configuration there will be one particular value of this index at which cavitation will begin. The value of $\sigma$ for the inception of cavitation is referred to as the *critical cavitation index*, $\sigma_{cr}$. If $\sigma$ is higher than $\sigma_{cr}$, there will be no cavitation present, but if $\sigma$ is less than $\sigma_{cr}$, the critical pressure will have been reached and cavitation will occur. The value of $p_c$ is usually taken to be the vapor pressure of the fluid although for extremely clear liquids it can be appreciably smaller, since it is generally accepted that nuclei of air or solid particles are needed about which the cavitation bubbles can grow.

The foregoing expression for the cavitation index can be rewritten in the form

$$\sigma = \frac{\left[\left(\frac{1}{\beta}\right)^4 - 1\right]}{C^2} \Big/ \left(1 + \frac{p_t - p_c}{p_i - p_t}\right) \qquad (3)$$

If it is assumed, as is often done in elementary books on fluid mechanics (*2*), that the minimum pressure occurs at the throat, then at the inception of cavitation $p_t = p_c$ and $\sigma_{cr}$ is given approximately by

$$\sigma_{cr} \cong \frac{\left(\frac{1}{\beta}\right)^4 - 1}{C^2} \qquad (4)$$

It was found that this approximation holds quite closely for all the $\beta$'s tested except $\beta = 0.0882$. For this case the calculated flow rates for the inception of cavitation were in error by approximately 2 to 3%, when the approximation of Equation (4) was used. From the data obtained, the term $\frac{p_t - p_c}{p_i - p_t}$ for $\beta = 0.0882$ was measured as varying from 0.03 to 0.05, depending slightly upon Reynolds number. Since that term is small compared to unity, it is suggested that the critical cavitation index be calculated for this case from

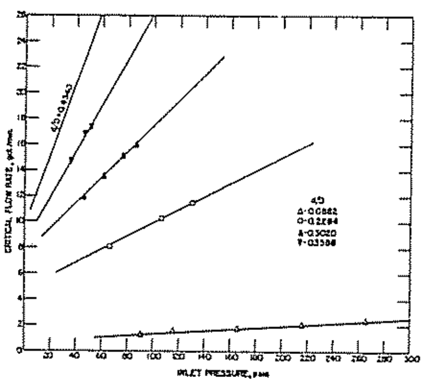

FIG. 8. COMPARISON OF PREDICTED AND EXPERIMENTAL CONDITIONS FOR INCEPTION OF CAVITATION FOR A SERIES OF VENTURIS.

$$\sigma_{cr} = 1.04 \frac{\left(\frac{1}{\beta}\right)^4 - 1}{C^2} \qquad (5)$$

Use of values of $\frac{p_i - p_c}{p_i - p_t}$ of less than 0.04 for intermediate values of $\beta$ between 0.0882 and 0.2264 should result in predicted flow rates within 2% of the correct values.

Since $C$ is a function of Reynolds number, which in turn is dependent upon the velocity to be obtained from $\sigma_{cr}$, the solution of Equation (5) for the critical velocity is an iterative one. As an example, $\beta = 0.0882$ and $p_i = 152$ lb./sq. in. gauge may be used. The problem is to find the flow rate at which cavitation will occur.

When it is assumed that $C = 1$, Equation (5) gives $\sigma_{cr} = 17{,}250$.

If $V_p$ is solved for in Equation (2) and atmospheric pressure taken as 14 lb./sq. in. gauge(*5*) and $p_v = 0.5$ lb./sq. in. gauge

$$V_p = \sqrt{\frac{p_i - p_v}{1/2\rho\sigma}}$$

$$= \sqrt{\frac{144\ (152 + 14 - 0.5)}{0.97\ (17{,}250)}}$$

$$= 1.192 \text{ ft./sec.}$$

The pipe Reynolds number will then be

$$N_{Re}, \text{ pipe} = \frac{V_p D}{\nu} = \frac{1.192(0.069)}{1.4 \times 10^{-5}} = 5{,}870$$

When the mean curve shown in Figure 4 is used, $C = 0.9$. Repeating the steps with this $C$ gives $N_{Re}$, pipe = 5,280. For this Reynolds number $C =$ 0.89, for which $N_{Re}$, pipe = 5,220. To the accuracy of Figure 4 the $C$ for this Reynolds number is again 0.89. The velocity corresponding to this $C$ is $V_p = 1.062$ ft./sec. The cross-sectional area of the pipe is 0.00373 sq. ft., and so the flow rate becomes $Q = 0.00396$ cu. ft./sec. or $Q = 1.78$ gal./min. This compares favorably with the measured flow rate of 1.80 gal./min. for the cavitating Venturi at this inlet pressure. The calculated and measured flow rates for the inception of cavitation for the five Venturis are compared in Figure 8. In these calculations, Equation (5) was used for the smallest Venturi and Equation (4) for the remainder. Good agreement is found between the calculated curves and the experimental points for the four smallest Venturis. For the largest Venturi, owing to the limitation in pump capacity, cavitation was not experimentally found; therefore, for this Venturi only the predicted variation in critical flow rate is presented.

**Pressure Recovery.** The efficiency of pressure recovery was computed for each set of data. This efficiency is defined as

$$\eta = \frac{p_o - p_t}{p_i - p_t} \qquad (6)$$

and represents approximately the percentage of the throat velocity head recovered as pressure head in the diffuser. It is desirable to maintain as high a value of this efficiency as possible in order to reduce the loss in total head attributable to the Venturi.

The variation of $\eta$ with the pipe Reynolds number for the five Venturis tested is given in Figure 9. These curves are for the noncavitating conditions only. It can be seen from this figure that there is little or no scale effect on $\eta$ for $\beta = 0.0882$ but for the larger Venturis the values of $\eta$ appear to increase slightly with increasing Reynolds numbers.

The effect of cavitation on the pressure-recovery efficiency is marked. This is readily apparent from the fact that once cavitation is reached for a fixed inlet pressure, the throat pressure remains constant for further reductions in the outlet pressure.

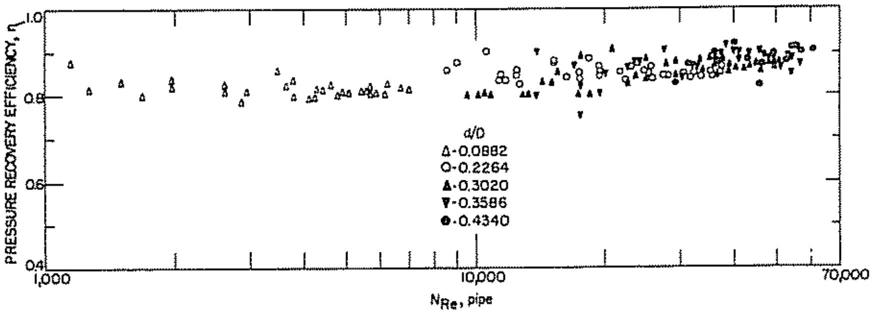


Fig. 9. Pressure recovery efficiency as a function of pipe Reynolds number for a series of Venturis.

## CONCLUSIONS

It is concluded from the analysis of the tests of five small Venturi tubes that the discharge coefficient of a Venturi tube decreases appreciably with decreasing Reynolds number below a pipe Reynolds number of approximately 10,000.

The approximation that the minimum pressure in a Venturi occurs at the throat is shown to be satisfactory for predicting the occurrence of cavitation in all but the smallest Venturi tested. For this Venturi with a throat-to-pipe-diameter ratio of 0.0882, a 4% correction to the inlet-throat pressure difference is recommended for computing the critical flow rate for the inception of cavitation.

The pressure-recovery efficiency for noncavitating conditions is found to increase slightly with Reynolds number for the Venturis with the large throat-to-pipe-diameter ratios and to be nearly independent of Reynolds number for the smallest Venturi. This efficiency for a given Reynolds number decreases with decreasing throat-to-pipe ratio; however, the rate of decrease is not prohibitive to the use of very low ratios since the efficiency of the Venturi with a throat-to-pipe-diameter ratio of 0.0882 is still approximately 81%.

It should be emphasized that these conclusions are valid only for the range of Reynolds numbers considered herein.


## ACKNOWLEDGMENT

The authors are indebted to D. C. Whitmarsh for the interest which he has shown in this work.


## NOTATION

$A_t$ = throat area, sq. ft.
$C$ = discharge coefficient
$d$ = throat diameter, in.
$D$ = pipe diameter, in.
$p_c$ = critical pressure for inception of cavitation, usually taken to be vapor pressure of fluid, lb./sq. ft.
$p_i$ = inlet pressure, lb./sq. ft.
$p_o$ = outlet pressure, lb./sq. ft.
$p_t$ = throat pressure, lb./sq. ft.
$Q$ = rate of flow, cu. ft./sec.
$V_p$ = flux average velocity in pipe, ft./sec.

### Greek Letters

$\beta$ = throat-to-pipe-diameter ratio $d/D$
$\eta$ = efficiency of pressure recovery
$\nu$ = kinematic viscosity, sq. ft./sec.
$\rho$ = fluid mass density, slugs/cu. ft.
$\sigma$ = cavitation index
$\sigma_{cr}$ = critical cavitation index, that is, at inception of cavitation

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