PERFORMANCE OF A VENTURI METER WITH SEPARABLE DIFFUSER

by Thomas J. Dudzinski, Robert C. Johnson, and Lloyd N. Krause

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

ABSTRACT

The effects on venturi meter efficiency and discharge coefficient of a radial, out-ward step at the transition from throat to diffuser are reported over a Reynolds number range of 1×10^4 to 5×10^5 and over a Mach number range of 0.2 to 1.0. Step size was varied from 0 to 12.5 percent of the throat radius. Diffuser efficiency was dependent on Reynolds number, Mach number, and step size greater than 2 percent. The discharge coefficient was independent of Mach number, step size, and back-pressure variation for critical flow.

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SUMMARY

The effects on venturi efficiency and venturi discharge coefficient of a radial, out-ward step at the transition from throat to diffuser were determined. A 1/2-inch- ($1\frac{1}{4}$-cm-) throat-diameter venturi with an ASME long-radius nozzle and an 8° included angle diffuser section was used. The step between the throat and the diffuser was sys-tematically varied from 0 to 12.5 percent of the nozzle-exit radius.

Data were obtained in air at near ambient temperature for a nozzle Reynolds num-ber range of 1×10^4 to 5×10^5 and a Mach number range of 0.2 to 1.0. Results indicate that a step size of up to 2 percent can be tolerated with no significant reduction in ven-turi efficiency. The efficiency increases with increasing Reynolds number, decreases with increasing Mach number, and decreases for step sizes greater than 2 percent.

Within the accuracy of the experiment, the discharge coefficient was independent of Mach number and step size, and was also independent of back pressure in the case of critical flow.

The use of a separable conical diffuser in a venturi meter appears feasible and of-fers a savings in cost and installation effort.

INTRODUCTION

In situations requiring low pressure loss, the venturi meter is one of the most com-monly used devices for the measurement of flow rate. Considerable work has been done on optimum meter design (refs. 1 and 2) and on the effects of piping configurations (refs. 3 and 4). Discharge coefficients have been analytically and experimentally determined for various venturi geometries (refs. 5 to 8). In addition, the performance of the meter has been studied in critical flow (refs. 9 to 11).

One of the shortcomings of the venturi has been the high cost of its fabrication. For maximum efficiency (maximum pressure recovery), the divergent half-angle of the dif-fuser is usually about 3° or 4° (refs. 12 to 14). If the venturi is fabricated from a single

piece of material, it is difficult to machine this small divergent angle so that the beginning of the diffuser is at the desired axial location. If the diffuser is fabricated separately from the entry section, it is difficult to match the diameter of the diffuser entrance with the nozzle-exit diameter.

A two-piece construction in which the conical diffuser is separate from the nozzle would offer considerable advantages in construction and assembly cost, if the discontinuity (step) in the junction between the throat and the diffuser due to manufacturing tolerances could be permitted. The primary purpose of this work was to study experimentally the effect of such a radial step at the nozzle exit on the discharge coefficient and on the diffuser efficiency. To ensure maintenance of the throat area at its desired value, only positive steps were investigated, that is, those steps in which the diffuser entrance was larger than the throat.

The variation of pressure recovery and discharge coefficient with Reynolds number, Mach number, and step size was determined over the Reynolds number range $1 \times 10^4 < R_{d,v} < 5 \times 10^5$; the Mach number range $0.2 \leq M_v \leq 1$; and the step-size range $0 \leq \Delta r/r_v \leq 0.125$. The discharge-coefficient measurements also provided information on the invariance of the discharge coefficient of ASME flow nozzles under critical flow conditions with varying supercritical pressure ratios.

SYMBOLS

A   area

C_d venturi discharge coefficient

D   inside diameter of pipe

d_o orifice diameter

d_v venturi-throat diameter

K   orifice flow coefficient

M_v venturi-throat Mach number

$\dot{m}$ actual mass flow rate

$\dot{m}_i$ ideal mass flow rate

p   pressure

R_d Reynolds number based on orifice or throat diameter and ideal mass flow rate

r   inside radius of venturi diffuser entrance

r_v initial inside radius of venturi diffuser entrance, equal to nozzle-throat radius

Δr diffuser entrance step, $r - r_v$

T   temperature

Y   orifice expansion factor

β ratio of throat or orifice diameter to pipe diameter

γ ratio of specific heats

η venturi efficiency

ρ density

Subscripts:

o   orifice

v   venturi

1   upstream location

2   downstream location for orifice case or throat location for venturi case

3   location downstream of venturi

TESTS AND APPARATUS

Description of System

A schematic diagram of the nominal 2-inch (5-cm) flow system used in the experiment is shown in figure 1. The system was designed and constructed in accordance with ASME recommendations (ref. 15) and consisted of two main portions: an upstream ori-
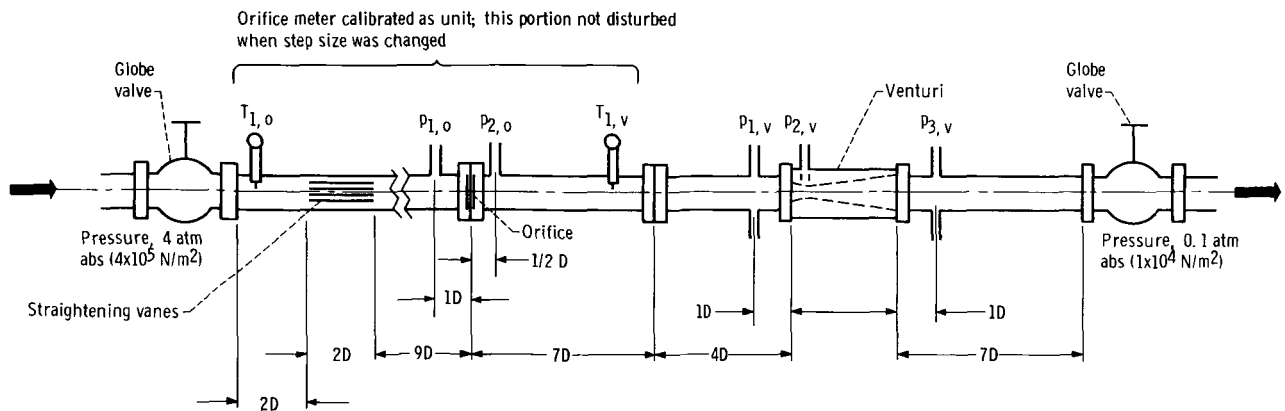

Figure 1. - Schematic diagram of flow system. Inside diameter of pipe, D, is 2.072 inches. (5.263 cm).

fice serving as a working standard, and the venturi being tested. To be able to vary the venturi-throat Mach number and Reynolds number independently, the upstream pressure was varied by means of a throttling globe valve. The test gas was air at near ambient temperature.

Working Standard

The mass flow rate was determined from the pressure drop across a standard ASME thin-plate concentric orifice with 1D and $\frac{1}{2}$D pressure taps. The stainless-steel orifice plate had a diameter d_o of 0.9985 inch (2.536 cm) and a diameter ratio β_o of 0.482.

The orifice, together with the upstream and downstream pipe sections, was calibrated through the required Reynolds number range in the Lewis water calibration facility. Figure 2, which is a plot of flow coefficient against Reynolds number (based on orifice diameter and ideal mass flow rate), shows that this calibration agrees with the ASME published values (ref. 15) to within about 1 percent. Inspection of the orifice showed that the systematic difference could not be accounted for by inadequate sharpness of the upstream edge of the orifice.
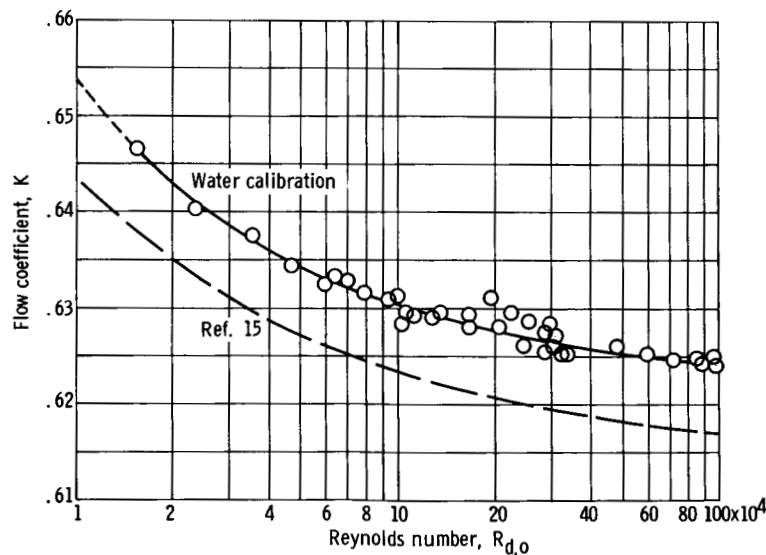

Figure 2. - Flow coefficient as function of Reynolds number for working standard orifice. Diameter of pipe, 2.072 inches (5.263 cm); orifice diameter, 0.9985 inch (2.536 cm); ratio of orifice diameter to pipe diameter, 0.482.
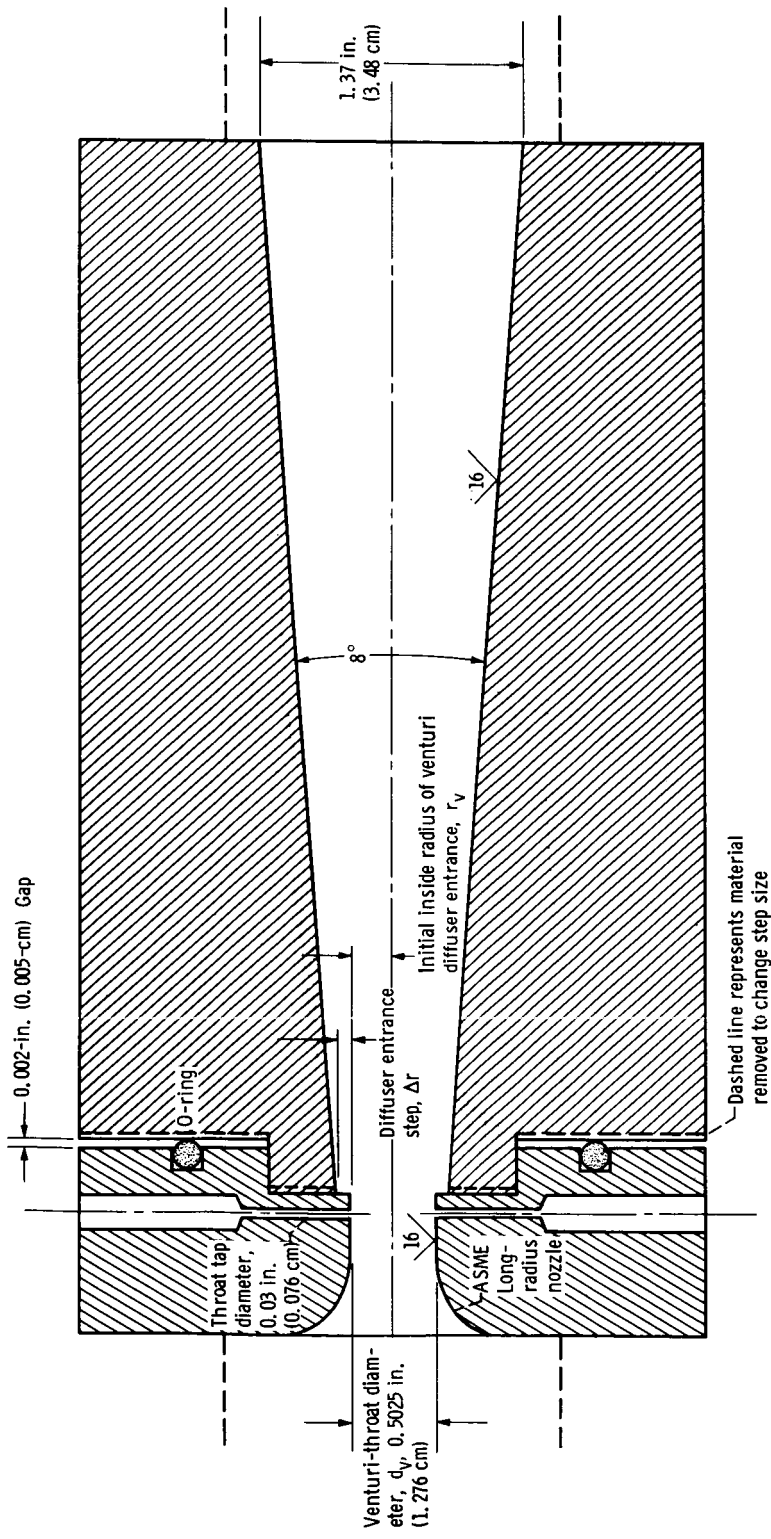
Figure 3. – Cross section of venturi. Ratio of venturi-throat diameter to pipe diameter, 0.242.

Venturi Meter

A cross section of the brass venturi meter tested is shown in figure 3. The inlet section is a standard ASME long-radius nozzle with a throat diameter d_v of 0.5025 inch (1.276 cm) and a diameter ratio β_v of 0.242. Two throat static taps 0.03 inch (0.076 cm) in diameter and 180° apart are located at a distance of $1.5 \, d_v$, or 0.75 inch (1.90 cm), from the upstream face of the nozzle. The diffuser section has a half-angle of 4° and an exit diameter of 1.37 inches (3.48 cm). The meter was initially constructed with a smooth transition between the convergent and divergent sections, that is, a step size $\Delta r / r_v = 0$.

Procedure

The initial tests were performed with the venturi step size $\Delta r / r_v = 0$. Data were obtained at each of five venturi Mach numbers in the subsonic Mach number range from 0.2 to 0.9, while the upstream static pressure was varied to cover the Reynolds number range. In addition, a range of critical flow conditions, with upstream- to-downstream-pressure ratios from 1:1 to 5:1, was covered. The venturi Reynolds number ranged from 1×10^4 to 5×10^5. The following pressures were measured:

$p_{1,o}$	static pressure upstream of orifice
$p_{1,v}$	static pressure upstream of venturi
$p_{3,v}$	static pressure downstream of venturi (in critical flow condition only)
$p_{1,v} - p_{2,v}$	differential pressure across orifice
$p_{1,o} - p_{2,v}$	differential pressure across nozzle portion of venturi
$p_{1,v} - p_{3,v}$	differential pressure across venturi

The upstream static pressures were measured with precision bourdon-tube gages having direct-reading 8-inch (20-cm) dials. The differential pressures were measured with precision quartz bourdon-tube gages with servo followers. Downstream static pressures were computed from the difference between upstream static pressure and the appropriate differential pressure, except that, when the venturi was operated in critical flow, a moderate-accuracy industrial bourdon-tube gage was used to monitor downstream static pressure directly. Gas temperatures were measured upstream and downstream of the orifice with bare-wire thermocouples.

Results of these initial tests established the diffuser efficiency and discharge coefficient for the venturi in its ideal condition.

A radial, backward-facing step was then introduced by the removal of material from

the diffuser at the inlet end. The magnitude of this step is represented by $\Delta r/r_v$ (fig. 3). The step size was systematically increased in seven steps up to 12.5 percent; the largest step was therefore about 0.03 inch (0.079 cm). At each step size, data were obtained at the flow conditions mentioned in the Procedure section.

CALCULATION PROCEDURE

The two quantities calculated were the venturi efficiency and the venturi discharge coefficient. The venturi efficiency η is defined in terms of directly measured differential pressures and is given by

$$\eta = 1 - \frac{p_{1,v} - p_{3,v}}{p_{1,v} - p_{2,v}} \tag{1}$$

The discharge coefficient C_d requires calculations of the ideal mass flow rate $\dot{m}_i$ and the actual mass flow rate $\dot{m}$ through the venturi. The ideal mass-flow-rate calculation assumes the flow to be one-dimensional and isentropic from the pressure measuring station upstream of the venturi to the pressure measuring station at the venturi throat. From reference 16,

$$\dot{m}_i = A_v \left[\frac{2 p_{1,v} \rho_{1,v} r^{2/\gamma} (1 - r^{\gamma - 1/\gamma})}{1 - \beta_v r^{2/\gamma}} \frac{\gamma}{\gamma - 1} \right]^{1/2} \tag{2}$$

where $r = p_{2,v}^* / p_{1,v}$ and $p_{2,v}^*$, the pressure in the throat of the venturi corrected for the pressure tap diameter (ref. 17), can be estimated for the case of the venturi used herein by

$$\frac{p_{2,v}^* - p_{2,v}}{p_{1,v} - p_{2,v}} = 0.0112 - 0.0076 \frac{p_{2,v}}{p_{1,v}} \tag{3}$$

The actual mass flow rate $\dot{m}$ is calculated from the thin-plate orifice calibration. From reference 16,

$$\dot{m} = KAY \left[2 \rho_{1,o} (p_{1,o} - p_{2,o}) \right]^{1/2} \tag{4}$$

where K is the flow coefficient determined by the water calibration. The expansion factor Y is defined as (ref. 16)

$$Y = 1 - \left(0.41 + 0.35 \, \beta_o^4\right) \frac{p_{1,o} - p_{2,o}}{\gamma p_{1,o}} \tag{5}$$

The venturi discharge coefficient is then given by

$$C_d = \frac{\dot{m}}{\dot{m}_i} \tag{6}$$

ACCURACY

The probable error in the determination of the venturi efficiency is about 1/4 percent over the complete Reynolds number range.

The probable error in the determination of discharge coefficient varies from about 1/2 percent at the low end of the Reynolds number range to about 1/4 percent at the high end of the Reynolds number range. The limit of error of 95 percent of the data was twice the probable error.

RESULTS AND DISCUSSION

The performance of a venturi can be described in terms of two parameters: the diffuser efficiency and the discharge coefficient. The efficiency of a diffuser is important because it is a measure of pressure loss. The discharge coefficient is important because it directly affects the accuracy of flow-rate measurement.

Diffuser Efficiency

The angle and the length of the diffuser shown in figure 3 were chosen to give maximum efficiency (refs. 12 and 13). The results obtained with such a diffuser over a range of Reynolds numbers and Mach numbers are shown in figure 4. Results are presented for three step sizes as representative of all the data taken. Figure 4(a) for zero step size shows that the efficiency increases with increasing Reynolds number. This result is expected because of the decrease in boundary layer thickness with increasing
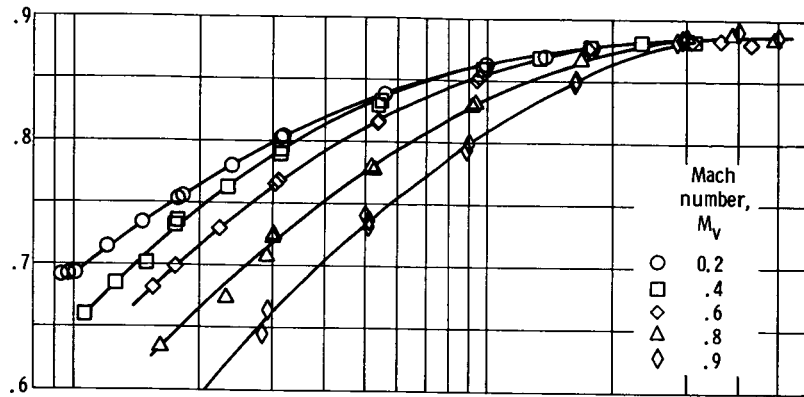
(a) Step size, 0.
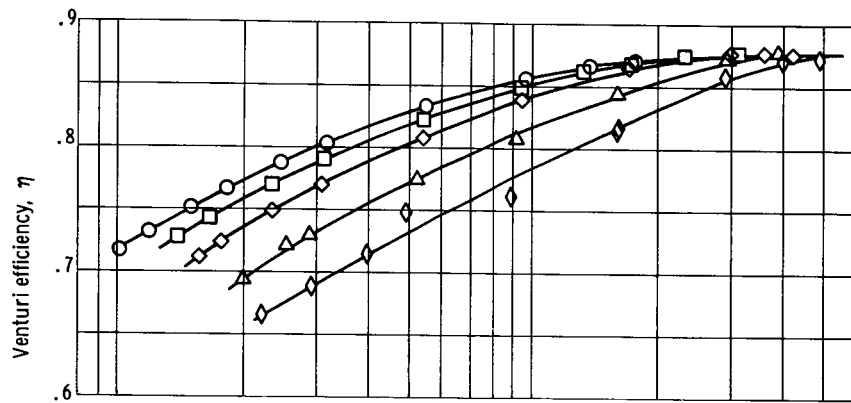
(b) Step size, 0.02.
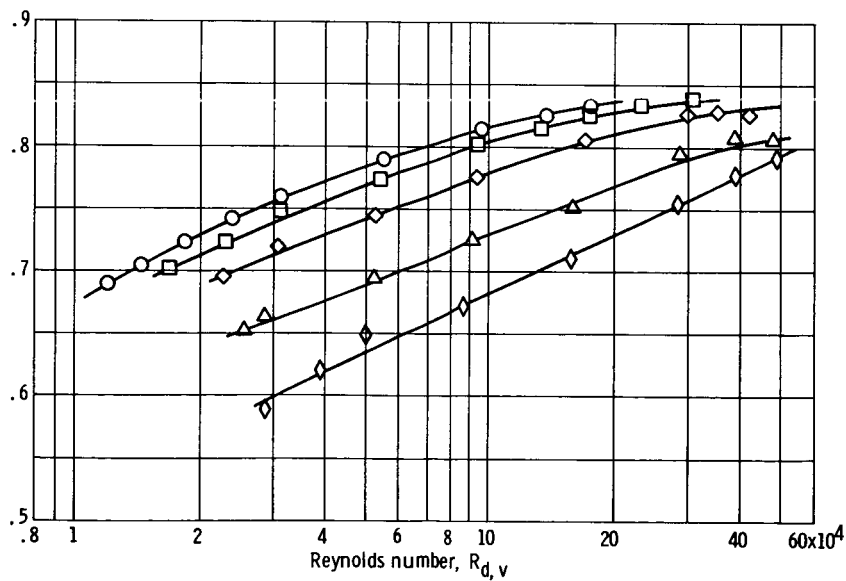
(c) Step size, 0.08.

Figure 4. – Venturi efficiency as function of nozzle Reynolds number.

Reynolds number. The maximum value of slightly less than 90 percent is in agreement with other reported results (refs. 12 to 14). Figure 4(a) also shows a decrease in efficiency with increasing throat Mach number. This decrease is the result of the severe adverse pressure gradient imposed on the boundary layer at the inlet portion of the diffuser at the higher Mach numbers. This Mach number effect diminishes with increasing Reynolds number, but becomes more pronounced as the step size increases, as shown in figures 4(b) and (c).

The effect of step size on efficiency is more clearly illustrated in figure 5, which is
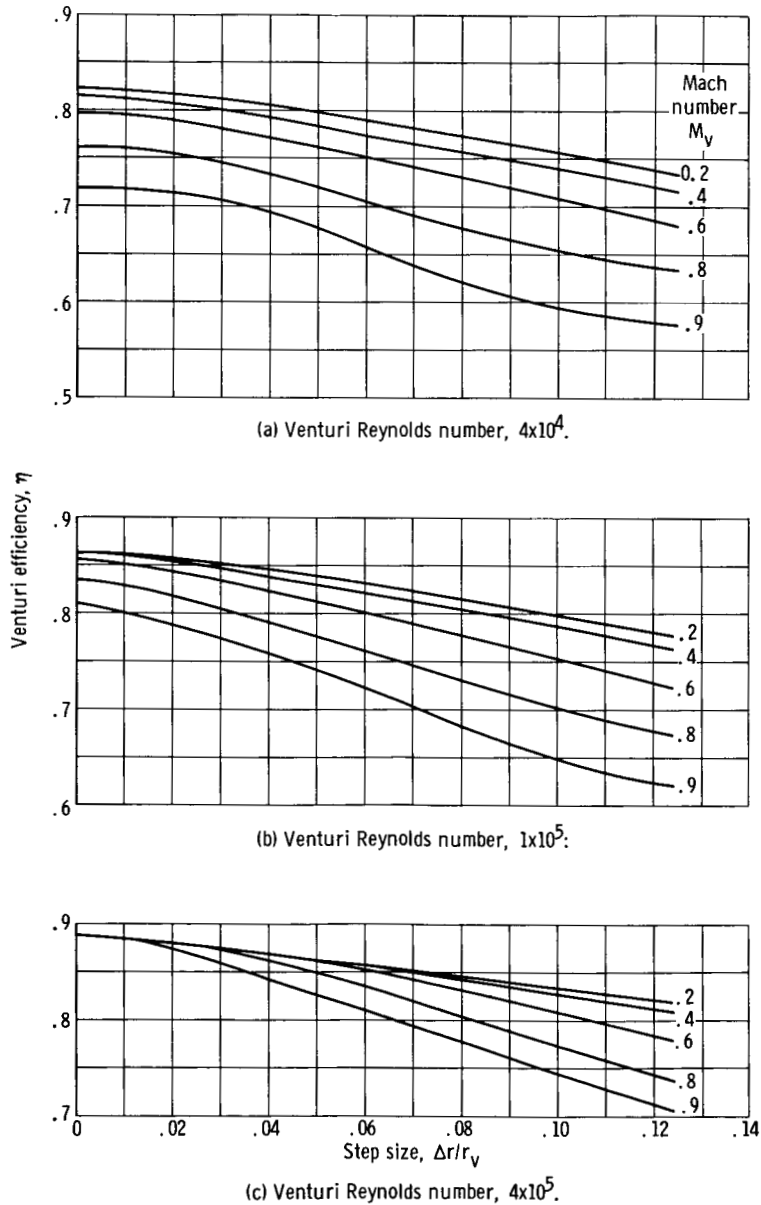

(a) Venturi Reynolds number, 4×10^4.

(b) Venturi Reynolds number, 1×10^5:

(c) Venturi Reynolds number, 4×10^5.

Figure 5. - Venturi efficiency as function of step size.

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a cross plot of figure 4 at Reynolds numbers of 4×10^4, 1×10^5, and 4×10^5. This figure shows that there is no significant change in meter efficiency as the step size is increased from the ideal value of 0 to a value of 2 percent. As step size increases beyond 2 percent, the efficiency of the meter decreases over the entire Reynolds number range and also becomes more strongly dependent on Mach number. These results indicate that, for the meter described in this report, the inlet diameter of the diffuser section can be 2 percent larger than the nozzle-exit diameter without a noticeable increase in pressure loss.

Discharge Coefficient

Venturi discharge coefficients are reported herein down to only a Reynolds number of 2×10^4. This lower limit is imposed by the corresponding lower limit of Reynolds number, 1×10^4, at which the flow coefficient of the working standard orifice is known accurately.

Since the step on the diffuser section of the venturi is located downstream of the throat static-pressure tap, no effect on the discharge coefficient is to be expected as the step size is increased. The absence of a systematic effect is verified in figure 6, which
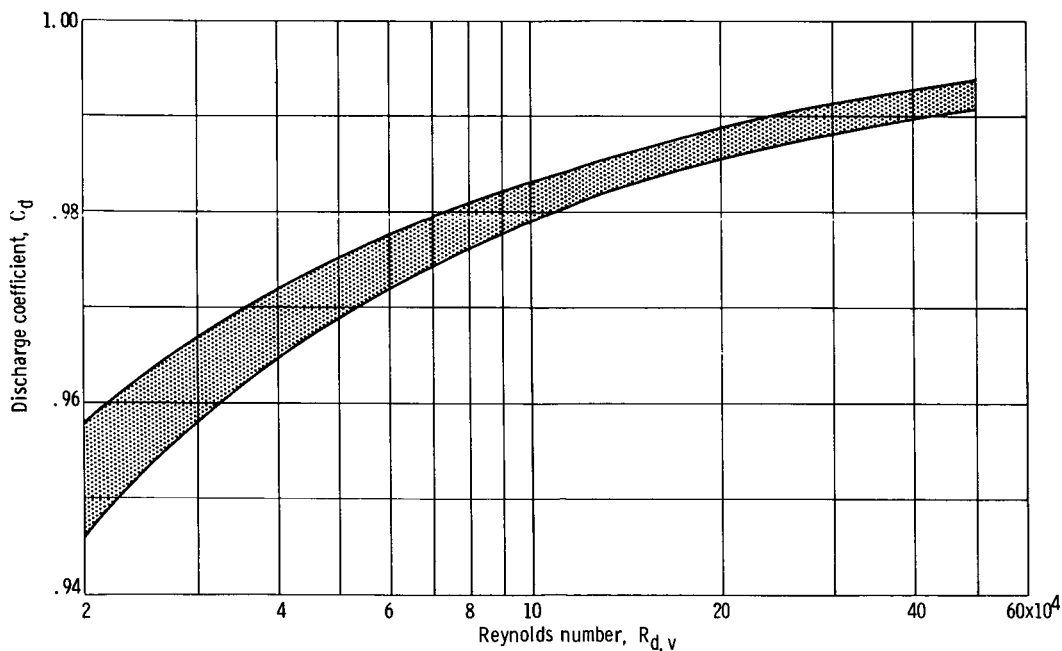
Figure 6. - Envelope of curves of discharge coefficient as function of Reynolds number for all step sizes. Inside diameter of pipe, 2.072 inches (5.263 cm); venturi-throat diameter, 0.5025 inch (1.276 cm); ratio of throat diameter to pipe diameter, 0.242.

shows the envelope of all curves of discharge coefficient against Reynolds number for all step sizes used. The half-width of the envelope is comparable with the probable error of the discharge-coefficient measurement.

All the data obtained at $\Delta r/r_v = 0$ are shown in figure 7. The mean curve drawn through these data is compared with an analytical calculation of the discharge coefficient for a long-radius ASME nozzle (ref. 5). The results agree within 0.4 percent. Comparisons between calculated and measured discharge coefficients, previously reported in the literature, show about the same agreement (refs. 5 and 8).
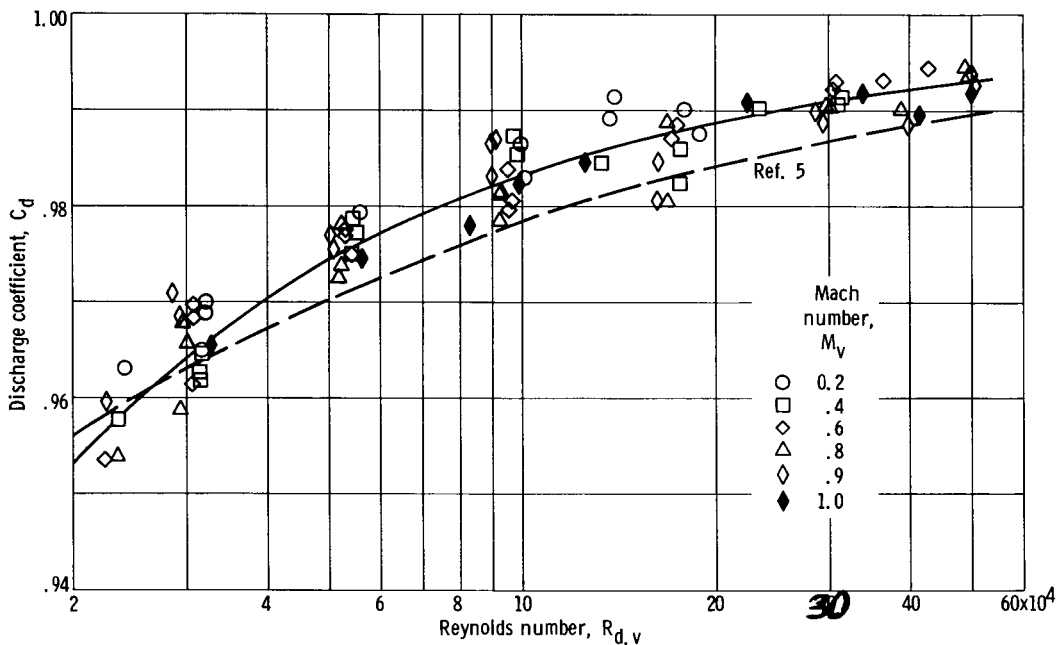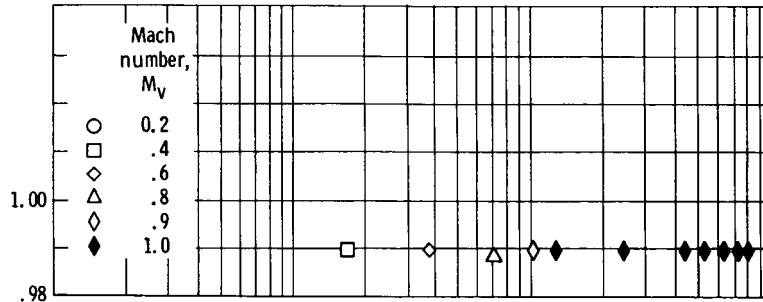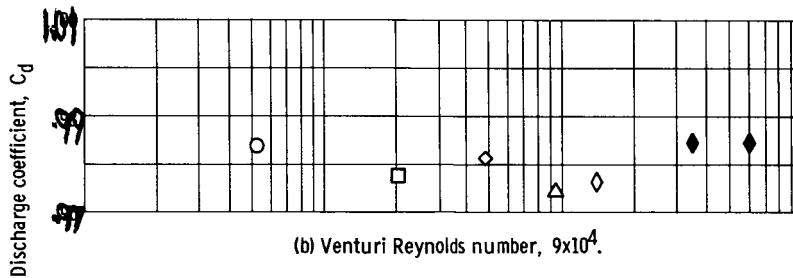

Figure 7. - Discharge coefficient as function of nozzle Reynolds number. Step size, 0; inside diameter of pipe, 2.072 inches (5.263 cm); venturi-throat diameter, 0.5025 inch (1.276 cm); ratio of throat diameter to pipe diameter, 0.242.

Figure 8 shows the insensitivity of the discharge coefficient to Mach number. Here, the discharge coefficient is plotted against the venturi pressure-ratio function $(p_{1,v} - p_{3,v})/p_{1,v}$ for several values of Reynolds number and for a step size of 8 percent. A value of 1 on the abscissa represents a pressure $p_{3,v}$ of 0 downstream of the venturi. This figure clearly demonstrates that variation in subsonic Mach number at constant Reynolds number and variation in pressure ratio at sonic throat velocity have a negligible effect on the discharge coefficient of an ASME flow nozzle. The negligible effect of Mach number is also illustrated in figure 7 for the case of zero step size. These results are in agreement with previously reported work (ref. 9). The increase in scatter
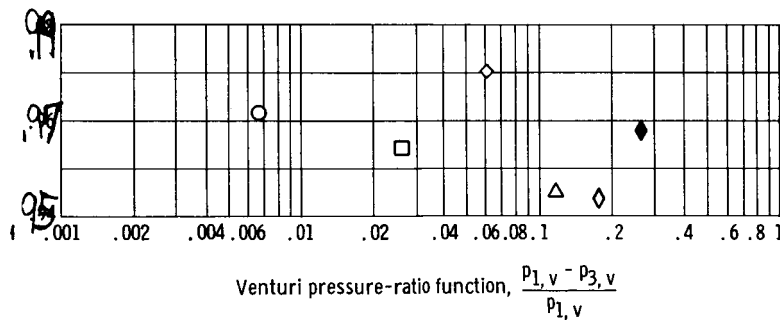
Figure 8. - Discharge coefficient as function of venturi pressure-ratio function.
Step size, 0.08; inside diameter of pipe, 2.072 inches (5.263 cm); venturi-throat
diameter, 0.5025 inch (1.276 cm); ratio of throat diameter to pipe diameter, 0.242.

of the data of figure 8 as the Reynolds number decreases is due to decreased accuracy in pressure measurements.

CONCLUDING REMARKS

Results of this investigation indicate that a step size $\Delta r/r_v$ up to 2 percent can be tolerated between the inlet section and the diffuser section of a venturi with no adverse effects on meter efficiency. The construction of two-piece venturi meters within this tolerance is feasible and would improve the ease of fabrication and handling.

Although the geometry of venturis may differ slightly in inlet contour and diffuser divergence, a similar relation between step size and efficiency may be expected for other related geometries.

The efficiency of the venturi is dependent on both Reynolds number and Mach number, the Mach number dependency decreasing with increasing Reynolds number. The efficiency increases with increasing Reynolds number, decreases with increasing Mach number, and decreases as step size increases beyond 2 percent.

The discharge coefficient of the meter is solely a function of Reynolds number, independent of Mach number and step size. Results indicate that, within the accuracy of the experiment, the discharge coefficient at constant Reynolds number is the same for both subsonic and critical flow. In addition, increasing the back-pressure ratio at critical flow has no effect.

The results for the discharge coefficient and the meter efficiency should not be extrapolated to higher Reynolds numbers because the upper limit of Reynolds number for these experiments approaches the transition from laminar to turbulent flow for the nozzle boundary layer.

Lewis Research Center,
 National Aeronautics and Space Administration,
  Cleveland, Ohio, January 24, 1968,
   128-31-06-77-22.

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ERRATA

NASA Technical Memorandum X-1570

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April 1968

Page 1:   In the first line of the Summary, "radially" should be "radial."

Page 6, paragraph 2, line 5: The ratio 1:1 should be 1.1:1.

Page 6, paragraph 2, line 11: $p_{1,o} - p_{2,v}$ should be $p_{1,v} - p_{2,v}$.

Page 12, figure 7: For the cycle at the right the abscissa scale should read 10, 20, 30, 40, 60×10^4.

Page 13, figure 8(b): The ordinate scale should read .97, .99, 1.01.

Page 13, figure 8(c): The ordinate scale should read .95, .97, .99.