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Gas flow measurement by the thin orifice and the classical Venturi tube☆

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Abstract

Many applications of vacuum technology require the generation and measurement of known gas flows. For this purpose, one may use ducts with constrictions. At a constriction, the flow-dependent pressure drop occurs. The gas throughput can be obtained from the inlet and outlet pressures at the constriction, provided its characteristics are known. In the present study, the characteristics of constrictions with basic geometry, i.e. the (infinitely) thin circular orifice and the standardised (DIN 1952) classical Venturi tube were investigated. Experimental methods for measuring the gas flow through a duct are described. The characteristics of individual constrictions were carefully measured and the data quantitatively compared to theoretical calculations. The results are discussed in order to provide a better understanding of the flow phenomena and to make them applicable to thin orifices and to Venturi tubes of any size and arbitrary gas. Over the whole flow range from molecular to viscous, the thin orifice can be used. However, it causes permanent pressure loss and it only has a small aperture in relation to the duct dimensions. In the viscous range, the Venturi tube can be used successfully. Thus, it is possible to establish stable secondary flow standards if the proper constriction is chosen.

Keywords: Gas flow; Flow phenomena; Metrology; Orifice; Venturi tube

1. Introduction

The phenomenon of gas flowing through ducts can be found in many fields, e.g. vacuum science, process technology, plasma physics, mechanical engineering and space travel. In many applications, it is important to measure such gas flows accurately. The measurement by using fundamental (primary) methods is frequently tedious and therefore it is necessary to find other methods which are more practicable. In the industry, there is widespread use of so-called mass flow meters based on the heat capacity of the gas. These meters are technical instruments whose characteristics depend on the way they are manufactured. They

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have to be calibrated individually for the different gas species; in addition, they need regular recalibration due to their inherent instabilities. Furthermore, these mass flow meters are not available for very small gas flows.

An interesting alternative approach for measuring gas flow is the use of constrictions. At a point of constriction, a significant drop in pressure occurs. Based on the inlet and outlet pressures, it is possible to calculate the gas flow, provided all the characteristics of the constriction are known. The present paper investigates the characteristics of particular constrictions with simple shapes and reviews their potential regarding gas flow standards.

The (infinitely) thin circular *orifice* has the simplest shape of all constrictions (Fig. 1 top). It is investigated here with regard to the special case in which the outlet pressure is negligible compared to the inlet pressure. In many applications this condition can be fulfilled. For example, when the inlet pressure is atmospheric pressure, the outlet pressure may vary from zero to about 400 mbar, i.e. as long as choked flow is avoided (see discussion below). A set of thin orifices with different diameters operated at varying inlet pressures thus may be employed to the generation of known gas flows in a wide range of applications.

The above condition, i.e. the outlet pressure is small compared to the inlet pressure, is not easy to fulfil in certain cases, e.g. testing mechanical pumps sucking in ambient air at ambient pressure. In this application, one should preferably use a

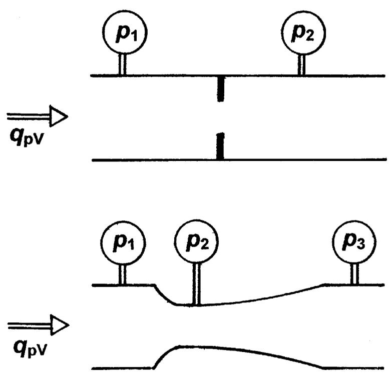


Fig. 1. Gas flow through an orifice (top) and a Venturi tube (bottom) with the positions for measuring the static pressure.

Venturi tube (Fig. 1 bottom). In the present paper the flow through Venturi tubes is investigated for the case in which the inlet pressure is nearly the same as the atmospheric pressure and the outlet pressure varies from zero to almost atmospheric pressure.

There are two essential differences between the orifice and the Venturi tube (Fig. 1). With the orifice, a sizeable permanent pressure loss from the inlet (pressure p_1) to the outlet (pressure p_2) occurs. With the Venturi tube, the flow is guided and the static pressure is also accessible at the throat. Depending on the flow, a sizeable pressure loss between inlet (pressure p_1) and throat (pressure p_2) may occur, which can be used for measuring the flow, whereas the pressure loss between inlet and outlet (pressure p_3) is much smaller.

A substantial advantage of both orifices and Venturi tubes is their inherent stability over a long period of time. Their characteristics depend exclusively on their geometrical shape. Furthermore, once the characteristics of such devices have been measured accurately, the characteristics of other individual devices of the same shape can be assumed.

The present paper starts with some basic definitions and then gives a brief description of various methods for primary measurement of gas flow. The main part of the paper gives a detailed description of gas flow through an orifice and a Venturi tube. Flow behaviour is discussed and experimental data obtained with the help of specific devices are presented. The data are regarded as exemplary, and scaling rules are given which render it possible to predict the characteristics of any orifice or Venturi tube under arbitrary conditions and for any gas species.

2. Characteristics of a constriction

Consider a duct with a constriction. Whenever there is a drop in static pressure along the duct, a gas flow develops. The gas flow itself is described by the pV value of the gas per time, i.e. the so-called throughput q_{pV}. By denoting the inlet and outlet pressures at the duct as p_1 and p_2

respectively, one arrives at the following equation [1,2]:

$$q_{pV} = C(p_1 - p_2). \quad (1)$$

Eq. (1) contains the conductance (C), which describes the characteristic properties of the constriction. In general, the conductance depends on the shape of the constriction, the inlet and outlet pressure, and properties of the gas, which in turn depend on the gas species and prevailing conditions.

3. Primary methods of measuring gas flow

The throughput is defined as pV value of the gas passing through a duct per time interval:

$$q_{pV} = \frac{\mathrm{d}(pV)}{\mathrm{d}t}. \quad (2)$$

The throughput depends on the temperature, since the pV value of a certain amount of gas (i.e. fixed mass or number of molecules) is proportional to the thermodynamic temperature. For obtaining a measurement of throughput, Eq. (2) can be rewritten by applying the product rule

$$q_{pV} = p\frac{\mathrm{d}V}{\mathrm{d}t} + V\frac{\mathrm{d}p}{\mathrm{d}t}. \quad (3)$$

In primary methods for measuring throughput, gas streams either into or out of a volume. According to Eq. (3), two basic methods can be distinguished (as shown in Fig. 2):

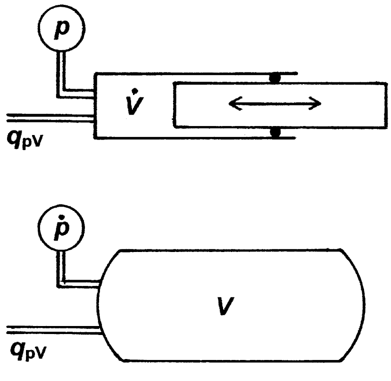


Fig. 2. Methods of measuring gas throughput. Top: constant-pressure changing-volume method. Bottom: constant-volume changing-pressure method.

- Constant pressure method, whereby the volume V is varied in such a way that the pressure p remains constant. This can be accomplished automatically by a servo loop, which reads the pressure and changes the volume accordingly. The volume may be varied by a movable plunger (piston, syringe), by distortable bellows or by a barrier liquid.
- Constant volume method, which monitors the change in pressure p in a vessel of fixed volume V due to the gas flow.

For accurate measurement of the gas flow, one thus needs three accurate physical quantities: pressure, volume and time. This will now be discussed in more detail.

The pressure p can be taken by capacitance diaphragm gauges which are commercially available with scales ranging from 0.05 mbar to more than 1000 mbar. These gauges offer high resolution and are basically accurate. Depending on the respective pressure, accuracy may be 0.5–0.1%. In the measurement of pressure changes ($\dot{p} = \mathrm{d}p/\mathrm{d}t$), the accuracy becomes poorer with regard to finite resolution, fluctuation and gauge drift.

It is relatively straightforward to measure volume change per time ($\dot{V} = \mathrm{d}V/\mathrm{d}t$) in the case of a piston. The measurement is quite simply the product of the cross-sectional area A of the piston and the velocity v of movement (i.e. $\dot{V} = Av$).

Measuring absolute vessel volume V might be more challenging. If the vessel has a simple geometrical shape, the volume may be calculated from its dimensions. If the shape is more complex, the volume actually has to be measured. In the so-called gravimetric method, the vessel is weighed when empty and then again when filled with degassed pure water at a known temperature. The volume is simply derived from the difference in weight and the known density of water. If a proper balance is used, the relative uncertainty of the vessel volume is generally below 0.1%. In the case of large vessels, the gravimetric method becomes tedious. Therefore, we applied the so-called modified gravimetric method. In this version, the large vessel is first evacuated and then filled with gas from a lightweight bottle (thin aluminium walls). The increase of pressure in the vessel caused by the filling is measured. The bottle is weighed before and after the filling process and the mass loss of weight pertaining to the bottle is taken as the mass of gas injected into the vessel. Knowing the mass of the gas and the pressure increase, the volume of the large vessel is calculated using the equation of state of the real gas.

At first glance the measurement of time appears trivial, but it is not. For example, in the constant volume method, the pressure rise per time has to be determined. There are two possible modes of operation. In the first mode, the continuous change of pressure is recorded. In typical types of apparatus, the pressure is available as a voltage from the gauge and converted into a digital number by a data acquisition system. Errors occur when the cycle times of conversion electronics and the computer clock are not correctly matched. In the second mode, the integral pressure change which occurs during the opening time of the connecting valve is measured. Here, the timing of the pressure reading is not critical, since the pressures can be read in the continuous states some time before opening and some time after closing the valve. However, the time interval during which the valve remains open has to be taken correctly. Since the reaction time of a valve is different for opening and closing, the electrical drive signal does not give the correct time interval. Instead one should employ a sensor (e.g. electro-optical) for detecting the actual valve plate position.

Furthermore, systematic disturbances of the throughput measurement have to be considered. One has to reckon with possible leaks and outgassing (desorption) of the vessel walls. The corresponding disturbances depend very much on the apparatus which is used as well as the operating conditions. Another crucial point for gases is temperature. When the ambient pressure changes, the vessel temperature also changes, and concomitantly the pressure of the gas in the vessel changes. A change in pressure pretends a gas flow.

Furthermore, thermal effects arise due to the gas flow itself. When gas flows out of the vessel, the temperature of the gas in the vessel decreases. When gas flows into the vessel, the temperature rises. Since gas only has a minimal heat capacity, it will take on the temperature in the vessel after some relaxation time due to its thermal conductivity. The time constant for equilibrium depends on pressure, vessel size and gas species; it has been carefully studied in previous work (Refs. [3–6]). For example, Fig. 3 shows the thermal relaxation of gas in a common cylindrical vessel with 100 ℓ volume. Initially, the vessel was filled with air at 1000 mbar. At time $t = 0$, the vessel was pumped out for a few seconds. As a result, the pressure in the vessel decreased due to the direct removal of the gas and the concomitant temperature decrease by the expansion of the gas in the vessel. After the pumping interval, the pressure rose due to the warming of the gas to ambient temperature. Fig. 3 shows that the thermal effects were considerable, and that the thermal relaxation time lasted several seconds.

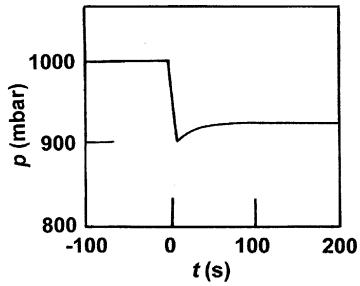


Fig. 3. Thermal relaxation process of gas in a 100 ℓ vessel. Initially, the vessel is filled with air at 1000 mbar. At $t \approx 0$, it is pumped for a few seconds, whereby the gas cools down. Thereafter, thermal relaxation takes places, causing a slight pressure rise.

4. Phenomena of gas flow through constrictions

4.1. Molecular flow

At very low inlet pressure, gas is very thin. Actually, it consists of individual molecules which hardly interact. As a consequence, each molecule travels through the duct in which the orifice is inserted without interacting with other molecules. The total gas flow is simply the sum of the independent motions of a lot of molecules. This kind of flow is called *gas-kinetic* or *molecular*.

As is well-known from the kinetic theory of gases, the conductance of an infinitely thin orifice of opening area A in the molecular flow regime takes the value [1,2]

$$C_{\mathrm{mol}} = \tfrac{1}{4}\bar{c}A, \tag{4}$$

where $\bar{c}$ is the mean thermal velocity of the gas molecules. Molecular conductance is thus independent of inlet and outlet pressure.

4.2. Gas-dynamic flow

At a high operating pressure, gas becomes dense. It may be treated as a homogeneous material and its flow through a duct can be mathematically described by continuum flow mechanics. When it is flowing, gas carries momentum and kinetic energy which are conserved if there is no interaction with the walls (no heat transfer and no friction). Think of the stationary flow of gas through a duct with a gradually decreasing cross-section (Fig. 4). When the cross-sectional area A decreases, the flow velocity v has to increase in order to guarantee the same amount of gas at each cross-section. Increased velocity means increased kinetic energy. The required energy is obtained from a reduction in both pressure p and temperature T, while the total amount of energy is conserved. This kind of flow is called *gas-dynamic*.

The gas-dynamic flow is calculated in many textbooks. The throughput of gas (taken at the inlet temperature) has the following value [2,7]:

$$q_{pV,1} = \sqrt{\frac{\pi}{4}}p_1\bar{c}_1 A_2 \Psi\left(\frac{p_2}{p_1}\right)\frac{A_1}{\sqrt{A_1^2 - A_2^2}}C_{\mathrm{d}}. \tag{5}$$

Quantities with index "1" are related to the inlet position, and quantities with index "2" to the outlet position, which is assumed to be the narrowest part ("throat"). In Eq. (5), the flow function $\Psi(p_2/p_1)$ and the discharge coefficient C_{d} have been introduced. The flow function Ψ depends on the ratios of pressures at outlet and inlet as well as on the isentropic exponent κ of the gas species. Two flow regions have to be distinguished based on the critical pressure ratio $(p_2/p_1)_{\mathrm{critical}}$ as given in Eq. (6)

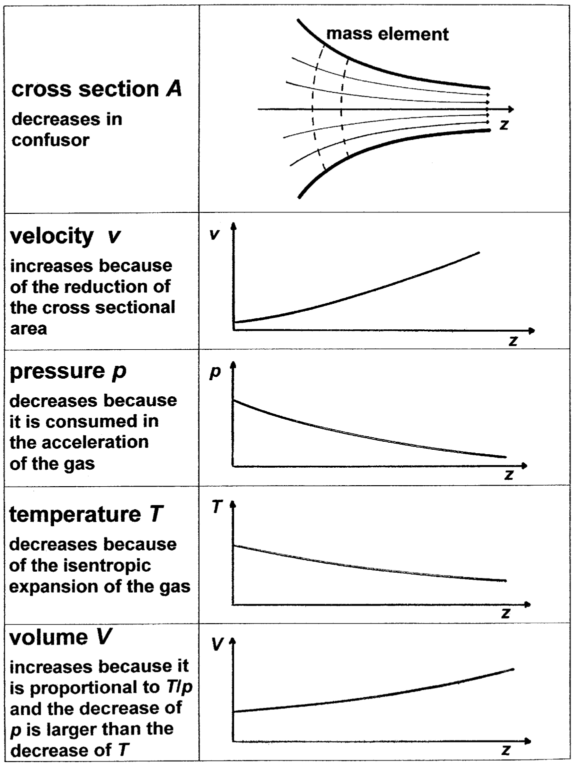


Fig. 4. Change of local properties of a gas flowing through a duct with gradually decreasing cross-sectional area A.

$$\left.\frac{p_2}{p_1}\right|_{\mathrm{critical}} = \left(\frac{2}{\kappa+1}\right)^{\kappa/(\kappa-1)}. \tag{6}$$

Here, κ denotes the isentropic coefficient of the gas. When the actual pressure ratio is larger than the critical pressure ratio, normal flow prevails and the flow function is given by

$$\Psi\left(\frac{p_2}{p_1}\right) = \sqrt{\frac{\kappa}{\kappa-1}\left[\left(\frac{p_2}{p_1}\right)^{2/\kappa} - \left(\frac{p_2}{p_1}\right)^{(\kappa+1)/\kappa}\right]}. \tag{7}$$

When the actual pressure ratio is smaller than the critical pressure ratio, choked flow develops and the flow function takes on its maximum (critical) value:

$$\Psi_{\mathrm{critical}} = \left(\frac{2}{\kappa+1}\right)^{1/(\kappa-1)}\sqrt{\frac{\kappa}{\kappa+1}}. \tag{8}$$

For easy reference, some calculated critical values are listed in Table 1, and the flow function for a number of gases is plotted in Fig. 5. The discharge coefficient C_d in Eq. (5) is dimensionless and accounts for two disturbing effects: contraction of the flow after an abrupt change of cross-section (Fig. 6 top) and friction losses at the walls, i.e. a boundary layer with low flow velocity (Fig. 6 bottom). In ideal conditions without disturbances, the discharge coefficient has the value 1. In practice, it is determined experimentally. For some constrictions with a well-defined shape and relatively large dimensions (50 mm diameter and more), comprehensive tabulations of the discharge coefficient are available in the German standard DIN 1952. Since these tabulations are only of minor use for our applications with small constrictions, additional measurements were performed as described below.

The disturbances depend on type and size of the constriction, as well as flow velocity and gas properties. They can be scaled by the Reynolds number Re, which is defined as

$$Re_d = \frac{\bar{v}_d d \rho_d}{\eta}, \tag{9}$$

where $\bar{v}$ denotes the flow velocity (average over cross-section), d the diameter at the cylindrical throat, ρ the density of the gas, and η the viscosity of the gas. The index "d" as a symbol means that the value at the throat must be used. The flow velocity in Eq. (9) is not known directly, but can be calculated from the throughput q_{pV} and cross-section $A = (\pi/4)d^2$. Thus one obtains the following useful equation:

$$Re_d = \frac{4}{\pi}\frac{q_{pV}(T_1)}{R_s T_1}\frac{1}{\eta d}, \tag{10}$$

Table 1
Parameters for gas-dynamic flow

Gas species	Isentropic exponent κ	Critical pressure ratio $(p_2/p_1)_{critical}$	Maximum value of flow function $\Psi_{critical}$
Inert gases	1.667	0.487	0.513
Air, N_2	1.400	0.528	0.484
N_2O	1.333	0.540	0.476
C_2H_6	1.200	0.564	0.459

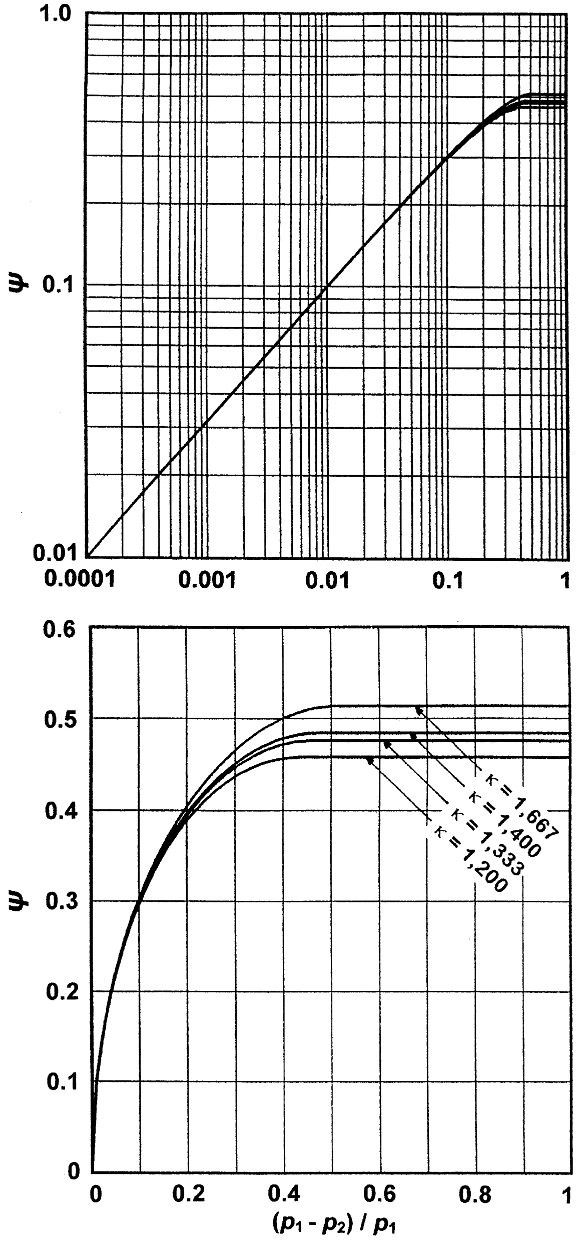
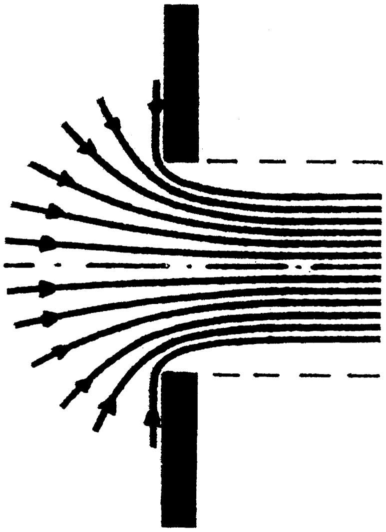


Fig. 5. Flow function (gas-dynamic) calculated according to Eqs. (7) and (8). Top: logarithmic scales at the axes. Bottom: linear scales.

where R_s denotes the specific gas constant and T_1 the thermodynamic temperature at the inlet. When the constriction is properly shaped and the throat is short, there are few disturbances. In the case of large Reynolds numbers ($Re > 10^5$), the discharge



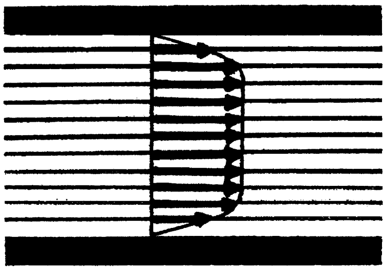


Fig. 6. Disturbances of gas flow. Top: An abrupt change of cross-section causes a contraction of the flow. Bottom: Wall friction causes reduced flow velocity close to the walls.

coefficient can be close to one, i.e. $0.99 < C_d < 1.00$ (see standard DIN 1952).

4.3. Transition flow

So far, we have treated the two limiting regimes of gas flow through a constriction, i.e. molecular (gas-kinetic) and continuum (gas-dynamic). The transition between these two regimes occurs in the so-called *transition regime*. Here, the distance which a molecule travels without colliding with another molecule, i.e. the mean free path $\bar{\ell}$, is equivalent to the transversal extension d of the constriction. The decisive parameter for the type of gas flow is the ratio of mean free path $\bar{\ell}$ to extension d, also known as the inverse Knudsen number Kn^{-1}. The mean free path $\bar{\ell}$ is reliably obtained from the viscosity η, which is normally taken at atmospheric pressure (see Refs. [2,8,9]). Thus we get the following equation:

$$\frac{1}{Kn} = \frac{d}{\bar{\ell}} = \frac{4dp}{\pi \eta \bar{c}}. \tag{11}$$

Sometimes, instead of the inverse Knudsen number, the rarefaction parameter (δ) is used. This is defined as

$$\delta = \frac{\sqrt{\pi}}{4} \frac{1}{Kn}. \tag{12}$$

Roughly speaking, in the range from $Kn^{-1} \approx \delta \approx 0.1$ to $Kn^{-1} \approx \delta \approx 100$, the flow changes from (almost) molecular to (almost) continuum.

5. Throughput measurement by a nozzle

The infinitely thin circular orifice appears to be an ideal device for studying the gas flow theoretically and experimentally as well as for generating gas flows in practical applications. Nevertheless, there are only very few investigations of gas flow through orifices over a wide range of pressures (see Ref. [10]).

In our experiment (Ref. [11–13]), we chose an orifice with a convenient size. The diameter of the orifice should be large enough to allow precise manufacturing and small enough to yield easily measurable throughputs. We used a hole with a nominal diameter of 1.2 mm, drilled in a tungsten foil of 10 μm thickness. Optical inspection of the hole revealed an excellent circular shape and an actual diameter of 1.233 mm ± 1%, whereby the uncertainty stems from light diffraction at the edges of the orifice.

When measuring the gas flow through the orifice, the inlet pressure was varied between 0.001 and 1000 mbar. Powerful pumps were used to keep the outlet pressure small in relation to the inlet pressure. In the molecular and transition regime, either a 500 l/s turbomolecular or a

$500\,m^3/h$ Roots pump (depending on pressure) was employed, keeping the pressure ratio smaller than 1:100. In the continuum flow regime, a $32\,m^3/h$ rotary vane pump was employed, keeping the pressure ratio smaller than 1:10, which is completely sufficient to warrant choked flow. Measurements were performed with the gases H_2, He, N_2, air, Ar, CO_2, C_3H_8, and Kr.

Fig. 7 shows the results which were obtained. The data of different gas species deviate markedly from each other. Although the differences are large, their origin is trivial. They stem mainly from the different thermal velocities $\bar{c}$ of the gases, which directly determines the flow (see Eqs. (4) and (5)). For better intercomparison of the data for the various gases and for a general application to arbitrary orifices, it is more appropriate to redraft the data in scaled units for abscissa and ordinate.

The ordinate shows the conductance. As shown above (Eq. (4)), the conductance (C_{mol}) of an infinitely thin orifice of opening area (A) in the molecular flow regime has a simple value. It is thus useful to plot C/C_{mol} at the ordinate.

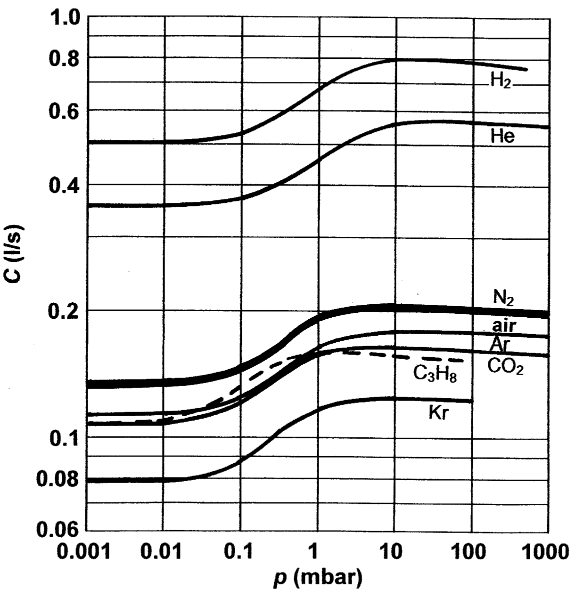


Fig. 7. Measured conductance of a thin orifice with a diameter of 1.23 mm as a function of inlet pressure p_1. The outlet pressure p_2 is much smaller than the inlet pressure p_1.

The ordinate shows the pressure. Again, for scaling purposes, it is advantageous to replot the data using either the inverse Knudsen number Kn or the rarefaction parameter δ at the ordinate. In calculating the scaling for the extension, we simply took the diameter d of the orifice. For the pressure p, we took the inlet pressure p_1 and not the average pressure of p_1 and $p_2 \approx 0$. The gas properties η and $\bar{c}$ were taken at inlet temperature, i.e. ambient temperature, and not at the temperature after expansion.

Fig. 8 shows the same data as in Fig. 7, but are plotted with scaled axes. Due to the scaling, at small inverse Knudsen numbers (small pressure), the normalised conductances of all gases are 1, independent of pressure and gas species. As the inverse Knudsen number increases, so too does the conductance. The physical reason for this behaviour is a change in the flow pattern from the motion of individual molecules to a collective directed flow, resulting in an increase of conductance. When the inverse Knudsen number becomes larger than, say, 1000, the collective flow is fully developed and gas-dynamic flow prevails.

The scaled conductance in the case of gas-dynamic flow is largest in the case of noble gases and smallest in the case of gases with complex molecules. The physical reason for this behaviour is just the molar heat capacity of the gases. Noble gases have low molar heat capacity ($c_p \approx 21\,J/mol\,K$), whereas the complex propane C_3H_8 has a high molar heat capacity ($c_p \approx 74\,J/mol\,K$). When the gas flows through the orifice, it cools by expansion. Noble gases with small heat capacity cool considerably. Thus they cover less volume and pass the orifice better than gases with complex molecules.

Theoretical calculations of the gas flow through thin orifices in the transition regime have recently become available [10,14,15]. These calculations were performed by the direct simulation Monte Carlo (DSMC) model [8], which employs statistical methods for treating the trajectories of numerous molecules. This model provides detailed information on the gas flow: local flow velocities, gas temperatures and other properties are available [14]. So far, results only exist for noble gases. Fig. 9 shows a comparison of our own experimental data, published results [16] and calculated values [10] of the conductance. The agreement is reasonably good, the deviation amounts in the worst instance to about 3%. Nevertheless, the deviation is significant. The experimental data show a more pronounced hill than theory at about $\delta = 100$. An experimental error caused by an artefact seems unlikely, since the scaled experimental data of three gas species (He, Ar, Kr) show good agreement, despite the fact that firstly the absolute conductance values differ by as much as a factor of 4.6 and secondly possible disturbances, e.g. caused by thermal effects, differ greatly for the individual gas species.

6. Throughput measurement by a Venturi tube

Venturi tubes are commonly used in the regime of higher pressures where continuum flow prevails. The basic idea of measuring flow by a Venturi tube is to obtain a significant pressure drop from inlet

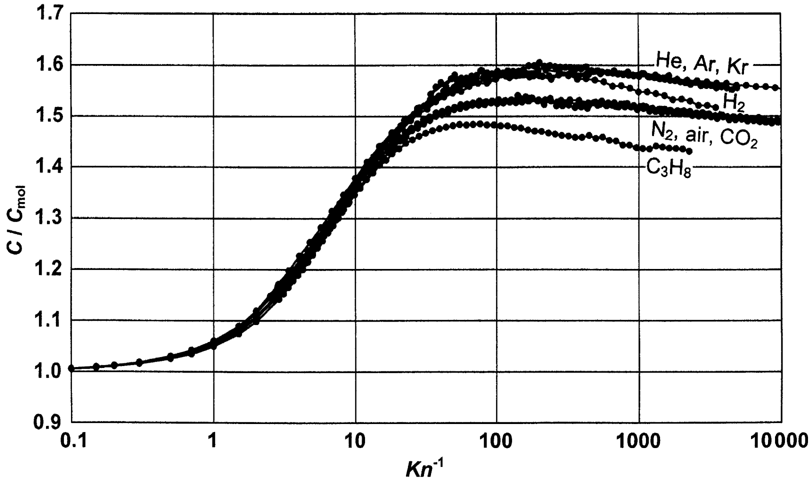
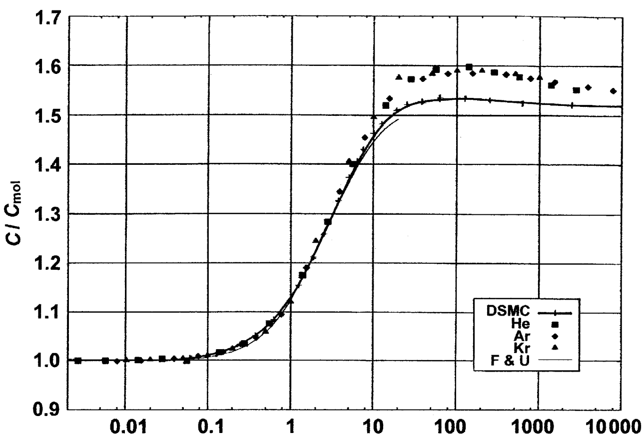


Fig. 8. Conductance of a thin orifice as a function of inlet pressure p_1. Same data as in Fig. 7, but the data at both axes were scaled.



$$\delta = \frac{\sqrt{\pi}}{4} \cdot \frac{1}{Kn}$$

Fig. 9. Scaled conductance of a thin orifice for noble gases. He, Ar, Kr: experimental data of the present work. F and U: results taken from Ref. [16]. DSMC: theoretical calculation of Ref. [10].

to throat, but to have a small pressure drop from inlet to outlet.

The Venturi tube consists of three sections (Fig. 10). In the first section (confusor), the cross-section is gradually reduced. The second section is a straight tube with one or more holes in the wall for measuring the static pressure. In the third section (diffusor), the cross-section is gradually increased. The overall shape may be optimised in order to keep disturbing effects to a minimum and to bring the discharge coefficient close to 1. In the optimisation, two conflicting demands have to be balanced. First, all sections should be as short as possible to keep friction losses at the walls small.

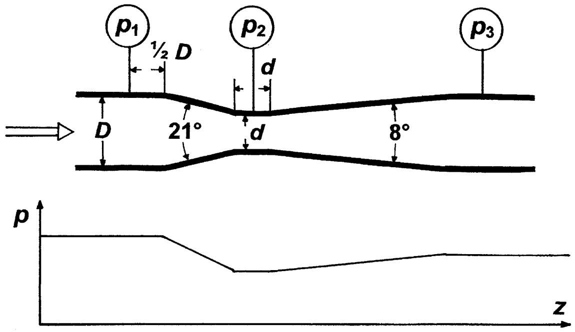


Fig. 10. Flow of gas through a classical Venturi tube. Top: geometry of the device. Bottom: static pressure along the tube.

Second, the change of the cross-section along the flow path should be sufficiently slow to avoid flow contraction at the inlet of the straight tube and to avoid tearing the flow from the walls in the diffusor.

There are a number of disadvantages associated with using an optimised Venturi tube. The optimum shape depends on the flow conditions, and it is difficult to manufacture. Therefore, Venturi tubes with a simple shape are of more use in practice, e.g. the classical Venturi tube with a 21° conical confusor and a 8° conical diffusor (compare DIN 1952, Fig. 7). In our experiments [11,13,17], we investigated those Venturi tubes with a classical shape and with throat diameters ranging from $d = 1$ to 8 mm as listed in the inset of Fig. 11. Measurements were performed by sucking ambient air—thus the inlet pressure p_1 was atmospheric pressure. The pressure difference $p_1 - p_2$ varied between 0.1 mbar (smallest difference pressure which could be reliably measured) and 300 mbar (largest difference pressure without entering the regime of choked flow). The obtained results are shown in Fig. 11.

The discharge coefficient C_d approaches the value 1 at large Reynolds numbers (in agreement with DIN 1952 and EN ISO 5167-1), which gives the result $C_d = 0.990 \pm 0.005$ (depending on wall roughness) at $Re > 2 \times 10^5$. The discharge

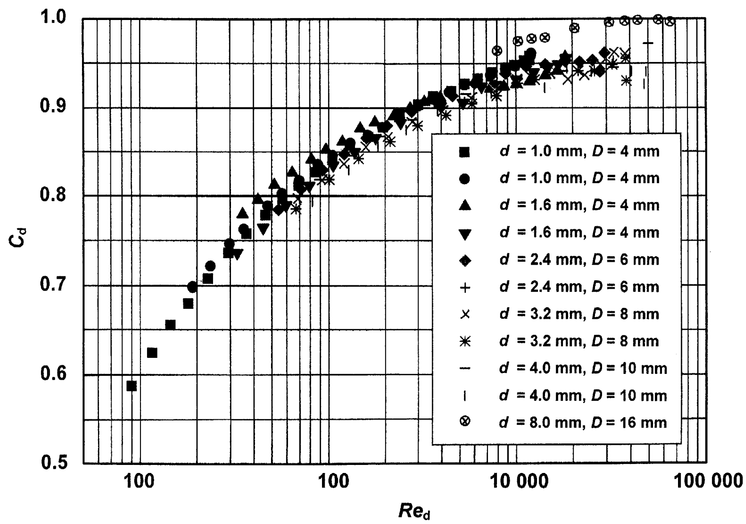


Fig. 11. Discharge coefficient of classical Venturi tubes with given throat diameters vs. Reynolds number.

coefficient becomes significantly smaller than 1 with decreasing Reynolds numbers. The reason for this behaviour is an increasing importance of friction at the walls.

The data in Fig. 11 essentially follow an universal curve with considerable scatter. A closer look at the scatter reveals that even discharge coefficients of two nozzles with the same nominal dimensions differ significantly from each other, as can be seen, for example, by comparing the upward and downward triangles symbolising the data of the nozzles with 1.6 mm throat diameter. Thus, one obvious reason for the scatter are manufacturing tolerances. The discharge coefficient is obtained by comparing the measured throughput with the throughput calculated by means of Eq. (5) (setting $C_d = 1$) and by using the nominal throat diameter, since the actual diameter could not be measured precisely. Apparently, Venturi tubes with the same nominal dimensions are not equal.

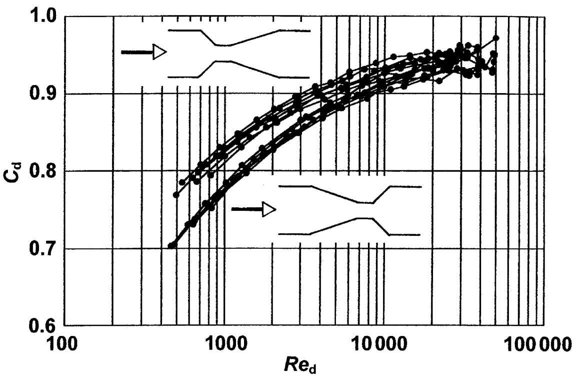


Fig. 12. Discharge coefficient of 6 Venturi tubes operated in normal direction (upper curves) and reversed direction (lower curves).

In order to get some more insight into the flow inside the Venturi tube, we also measured the throughput in the classical Venturi tube operated in the reverse direction, i.e. with 8° conical confusor and 21° conical diffusor (Fig. 12). In the reverse direction, the discharge coefficients are substantially smaller than in the normal direction, particularly at small Reynolds numbers. This behaviour is to be expected. In the reverse direction, the entrance section of the Venturi tube is considerably longer, leading to much more wall friction.

Finally, we also measured the permanent pressure loss from inlet to outlet. Fig. 13 shows the ratio of pressure loss from inlet to outlet and the pressure loss from inlet to throat. The data confirm the expected behaviour that this ratio is reasonably smaller than 1, and becomes quite small (≈ 0.1) with large Reynolds numbers.

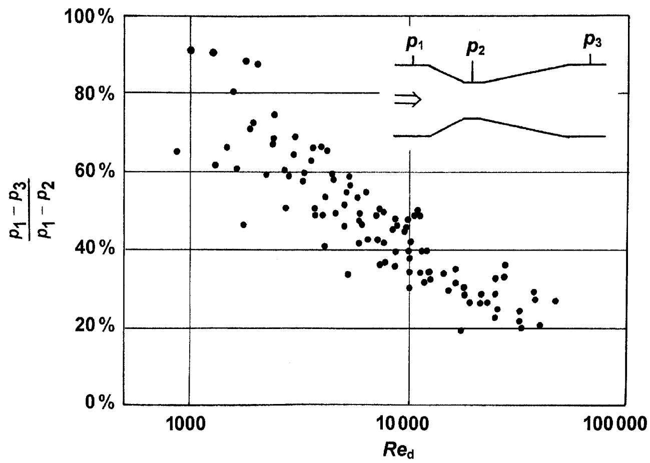


Fig. 13. Permanent pressure loss in a Venturi tube related to internal pressure drop vs. Reynolds number.

7. Summary

The present investigations have revealed conclusive information on the gas flow through thin orifices and Venturi tubes. In particular, it was possible to obtain comprehensive information on the thin circular orifice over a wide range of pressures as well as new information on flow through Venturi tubes with very small Reynolds numbers. For both devices, the flow behaviour is now well understood and the throughput can be quantitatively predicted with reasonable accuracy (uncertainty of a few per cent or less). Therefore, these constrictions can be used as flow standards in applications with modest requirements on accuracy, e.g. testing pumps (DIN 28426, 28427, and 28428 ISO 21360). In metrological applications with high demands on accuracy, a calibration of the constrictions is required. The advantages of using constrictions as flow standards is that their performance is easier to predict under different operating conditions and that their characteristics have an excellent inherent stability with time.

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