THE DESIGN, CONSTRUCTION AND CALIBRATION OF
A LOW VELOCITY ANEMOMETER
CALIBRATOR

by

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CHAPTER I

INTRODUCTION

One unique feature of man's perception of his environment is his desire to quantify what he observes. This desire has led naturally and inevitably to the development of number systems and units of weights and measures. As science and technology have evolved, measurement techniques and equipment have been devised and refined, enabling man to discover and understand the laws of nature.

The flow of fluids is a commonly occurring phenomenon which the scientist and engineer have been challenged to predict, control and measure. The two most common fluids, water and air, have been puzzled over for centuries and have been studied in detail for many years.

This thesis addresses the problem of measuring airflow, specifically at low velocities, in the range of 20 to 2,000 feet per minute. Several techniques and many versions of hardware are available to measure air velocities in the field and in the laboratory. (See *FLUID FLOW AND MEASUREMENT*, in *Appendix D*, page 65). Selecting the most appropriate meter and using it correctly depends partly on the degree of accuracy required, the purpose of the airflow (environmental control, conveying, process control, and so forth), and degree of access to the airstream.

The heating, cooling and ventilation industry, for example, makes wide use of airflow to achieve comfortable environments, to control products and processes, and to remove contaminants from the air. Our constantly improving standard of living, desire for increased productivity, and growing concern for industrial hygiene have created tremendous interest in accurately measuring air velocity in conduits (ductwork) and in free space. For instance, laminar flow clean rooms, required by space technology, demand large quantities of very clean air flowing at low velocities. Conversely, industrial exhaust hoods are designed to draw in ambient air at sufficiently high velocities to capture the gaseous and particulate contaminants which must be removed. In either case, the need for making field measurements of velocity to verify design criteria is obvious.

In 1971, the Williams-Steiger Act (Public Law #91-596) went into effect. This law, better known as the Occupational Safety and Health Act (OSHA), established many conditions which must be met in industrial and commercial buildings, including maximum and minimum rates of airflow surrounding the occupants. Because of this law, plant engineers, design consultants, government inspectors, installing contractors and union personnel all have need for air velocity meters to check ambient conditions.

The read-out of the meter thus becomes critical in settling disputes over performance of air moving systems and conditions in the work spaces. Obviously, accurate calibration of the meter is vital, and a method is needed for checking the calibration, especially at velocities below 100 feet per minute.

This thesis presents the design, construction, and calibration of an instrument of known characteristics to accurately and easily calibrate several types of commercial air velocity meter currently on the market. The instrument, called a calibrator, is constructed of readily available materials and equipment, and produces accurately known outlet velocities.

CHAPTER II

DESIGN CONSIDERATIONS

Background

Measurement of air velocity can be accomplished several ways (see Appendix D, FLUID FLOW AND MEASUREMENT, page 65). The airflow generally will be occurring either in closed conduits (ductwork) - typical of heating, cooling, ventilation, air cleaning, and exhaust systems - or under ambient indoor or outdoor conditions. Table 1, page 5 shows the range of velocities and temperatures, with typical values, for several applications. Since outdoor conditions are generally the province of meteorologists and air pollution engineers, this thesis confines its attention to indoor applications.

The range of velocities, from no noticeable velocity up to 6,000 ft/min (which is about 68 miles per hour), has led to the development of several types of meter and various calibration techniques. The most troublesome calibration procedure seems to be in the range below 100 ft/min, where accurately known velocities are hard to achieve.

Present Methods

Most manufacturers of commercial air meters have developed their own calibration procedures and devices usually in compliance with standards established by the American Society of Mechanical Engineers (ASME) [3]*,

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*Numbers in brackets refer to References on page 53.

TABLE 1

AIR VELOCITIES AND TEMPERATURES
FOR VARIOUS APPLICATIONS

Application	Range of Velocities	Typical Value	Range of Temperatures	Typical Value
Outdoors (USA)				
Summer	0-100 mph	7 mph	45° to 130°F	80°F
Winter	0-60	15	-30° to 85°	40°
Indoors				
Sedentary (summer)	0-40 ft/min	25 ft/min	72° to 80°	75°
(winter)	0-30 "	20 "	68° to 78°	75°
Factory (summer)	0-1000 "	100 "	70° to 100°	85°
(winter)	0-250 "	50 "	55° to 80°	68°
Low velocity duct system (heating, cooling, ventilating)	300-2000 ft/min	1200 "	40° to 170°	125°(H)* 55°(C) 75°(V)
High velocity duct system (heating, cooling, ventilating)	1800-5000 ft/min	3000	40° to 150°	100°(H) 45°(C) 75°(V)
Cold storage	20-2000 ft/min	Varies with product	-40° to 65°	Varies with product
Industrial exhaust				
Hood	50-200 ft/min	150	65° to 400°	85°
Duct	2000-6500	4000	"	"

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* (H) - heating; (C) - cooling; (V) - ventilating

where nozzles, venturis, or orifice plates of presumably known performance characteristics - that is, the relationship between flow rate, fluid density, and pressure drop across the device - are used for calibration.

Published performance data are taken as the calibration standard, and discharge velocities are calculated from the inferred flow rate through a known discharge area. If the discharge velocity is sufficiently high, say 450 ft./min. or greater, the velocity pressure can be determined by a pitot-static probe traverse, from which flow rate and velocity can be calculated without having to resort to nozzles, orifices, or other restrictions in the flow.

Other methods have also been tried, such as dragging the meter probe at known velocity around in a closed chamber of "still" air, with questionable success.

One of the more sophisticated, delicate, expensive instruments, the DISA low velocity anemometer, is calibrated by sweeping out a known volume at a measured rate through a fixed area for which the velocity profile can be determined [7]. The heated vibrating wire sensing probe is so small compared with the area that velocity "at a point" is being sensed. However, this type of instrument is beyond the budget and required accuracy of most commercial work, and has an upper range of only 200 ft./min.

Design Objectives

The goal of this work is to design and construct a calibrator for commercial meters which meets the following criteria:

1. Have provision to measure pressure drop across one or more nozzles for determining the volume flow rate.

2. Produce a range of flow rates which result in discharge velocities from 10 ft./min. to 2,000 ft./min.
3. Conform as nearly as possible to ASME Power Test Code criteria.
4. Include a holding device near the discharge to position accurately the probe of the meter being calibrated.
5. Have provision for future addition of heating or cooling sections to heat the inlet air to 170°F or cool it to 40°F.
6. Include viewing panels so that entering and leaving sides of the nozzles are visible.
7. Weigh less than 75 pounds, be portable, and be small enough to fit on a desktop.
8. Be constructed of readily available materials.
9. Operate on 120 volt single phase 60 Hz electrical power.

Theory of Operation

A brief description of fluid meter theory of operation appears in Appendix D, FLUID FLOW AND MEASUREMENT, page 63 . Based on that theory, this calibrator was designed to consist of a fan section, a large plenum divided into two sections by a partition containing four ASME long-radius nozzles of different diameter mounted in parallel, and a 4" diameter discharge duct terminating in a 2" diameter ASME long-radius nozzle. Figure 1, page 9 , shows the schematic arrangement of the various parts. A detailed discussion of each part follows.

Fan Section

The selection of a fan for a particular duty requires knowledge of the volume flow rate, total pressure and static pressure required, operating temperature, capacity control requirements, and the operating characteristics of the various types of fan.

The maximum and minimum flow rates were calculated from the relationship between the discharge area and the desired discharge velocities. For purpose of fan selection, it is sufficiently accurate to write the relationship as follows:

$$q = A \times V \qquad [1]$$

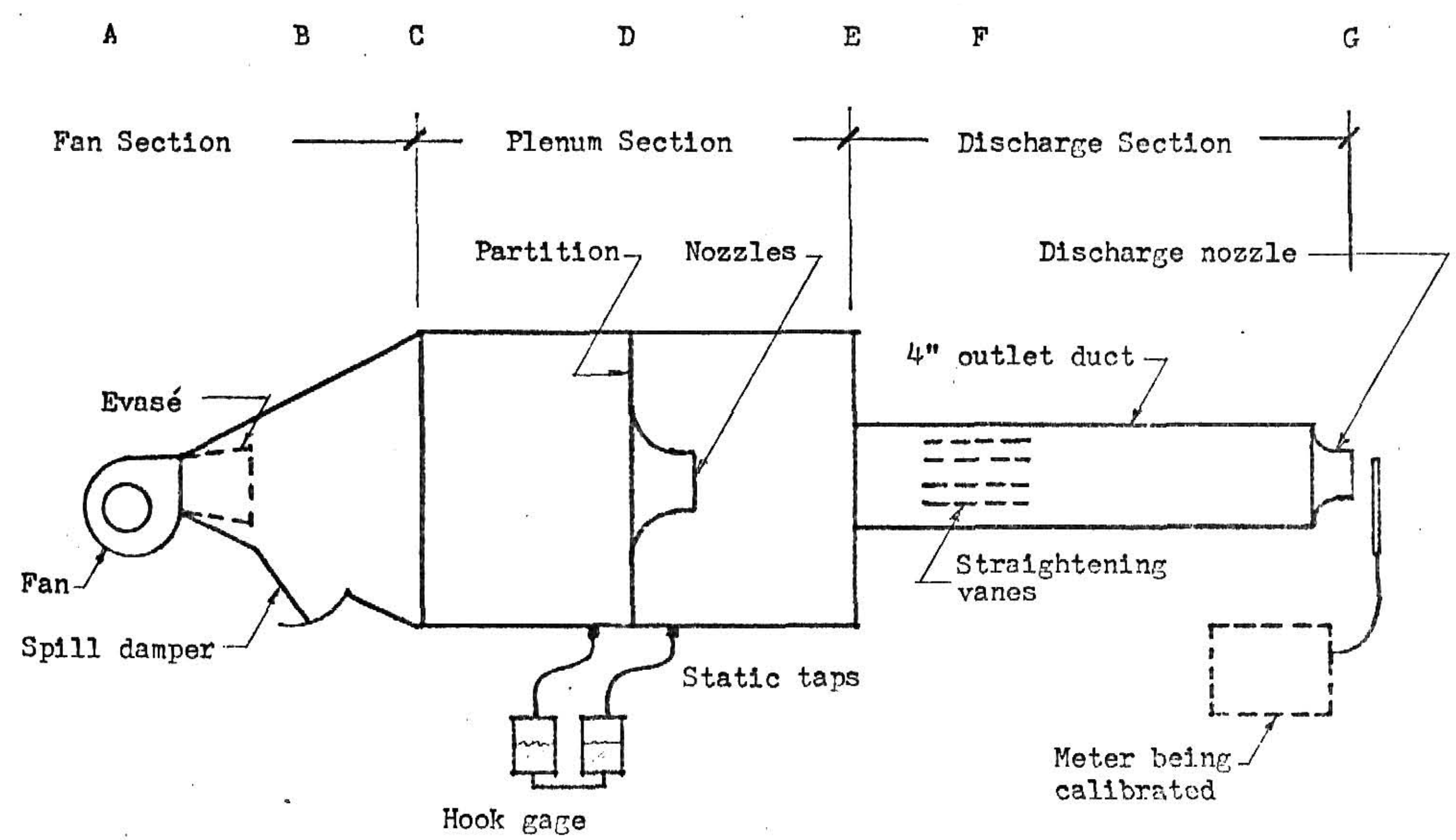
where q is volume flow rate in ft^3/minute

A is cross-sectional area in ft^2

V is outlet velocity in ft/minute

A long-radius ASME nozzle was selected as the discharge port (location G, Fig. 1) for this calibrator to achieve an outlet velocity profile as flat as possible [14]. Using a 2" diameter nozzle, 2,000 ft/min. outlet velocity required 45 ft^3/min. flow rate, and 10 ft/min. requires 0.22 ft^3/min. flow rate.

The total head depends on the configuration of the calibrator, the head loss through the calibrating nozzles, and the velocity head at discharge. At the maximum flow rate, the discharge velocity head is 0.25" water (see equation [20] in Appendix D) and it will be shown later that the head loss through the nozzles is about 0.28". For estimating purposes, if we allow another 0.25" to account for friction and dynamic losses in the system, then the fan should deliver 43 ft^3/min. at 0.78" total pressure. By selecting the fan to produce 0.78" static pressure (rather than total pressure) a safety factor is included.



SCHEMATIC ARRANGEMENT OF COMPONENTS

Figure 1

The combined requirements of relatively low flow at relatively high head, compactness, quiet operation, economy and availability led to the selection of a single inlet, single width, direct drive centrifugal fan having a forward curved wheel. Figure 2 shows the performance curve of this fan plotted from the manufacturer's published data. More detailed information on the fan appears in the Materials and Equipment List, page 54.

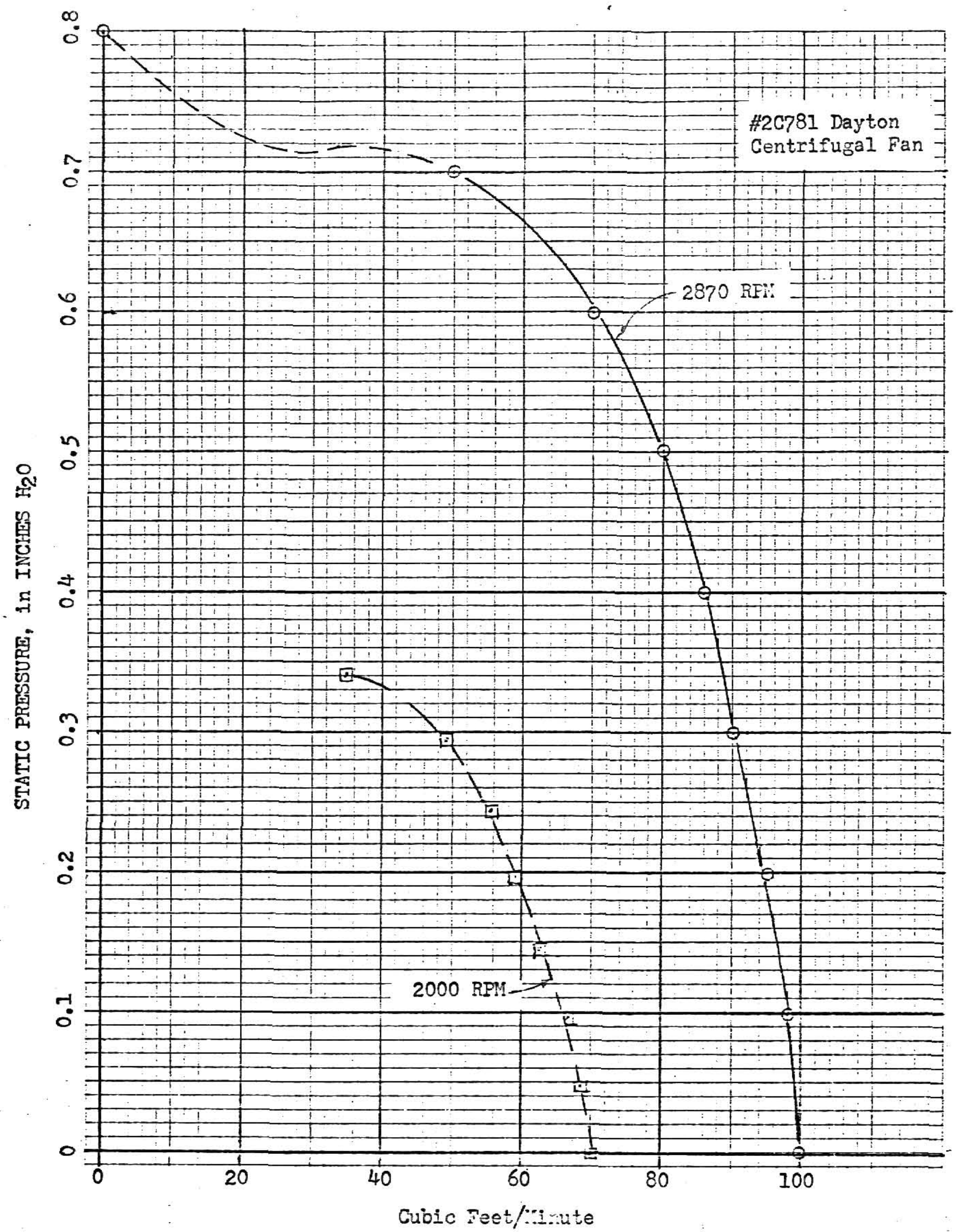
A forward-curved wheel has a dip in the volume vs. head curve, so most manufacturers do not publish performance data to the left of the peak of the curve, hoping to minimize the likelihood a user will select a fan to operate in the unstable area. In Figure 2, the unpublished portion of the curve is shown dotted.

There are two possibilities for the relative positions of the fan section and the plenum section, as shown in Figure 3. Each arrangement has several advantages and disadvantages.

1. Drawthrough - two possible configurations are shown. The solid lines show the fan located outside the plenum. The dotted lines represent a longer plenum with the fan mounted inside. Either drawthrough arrangement has the following advantages:

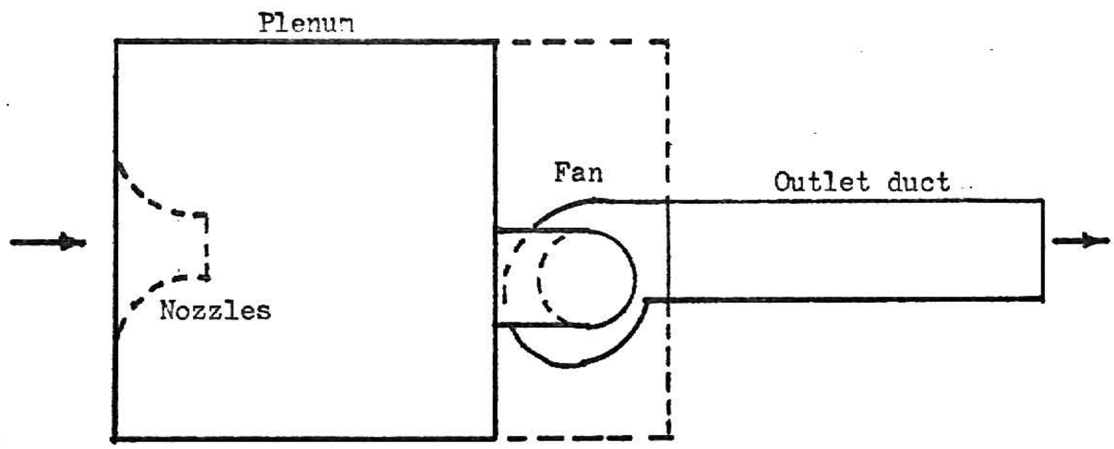
  a. Smooth airflow approaching the nozzles.
  b. Easy access to the nozzle inlets, to change which nozzles are in use.
  c. No inlet plenum required.
  d. Small expansion from the fan outlet to the discharge duct.

The fan-inside arrangement has the additional advantage that no ductwork is required to the fan inlet, since the fan can draw its air directly from the

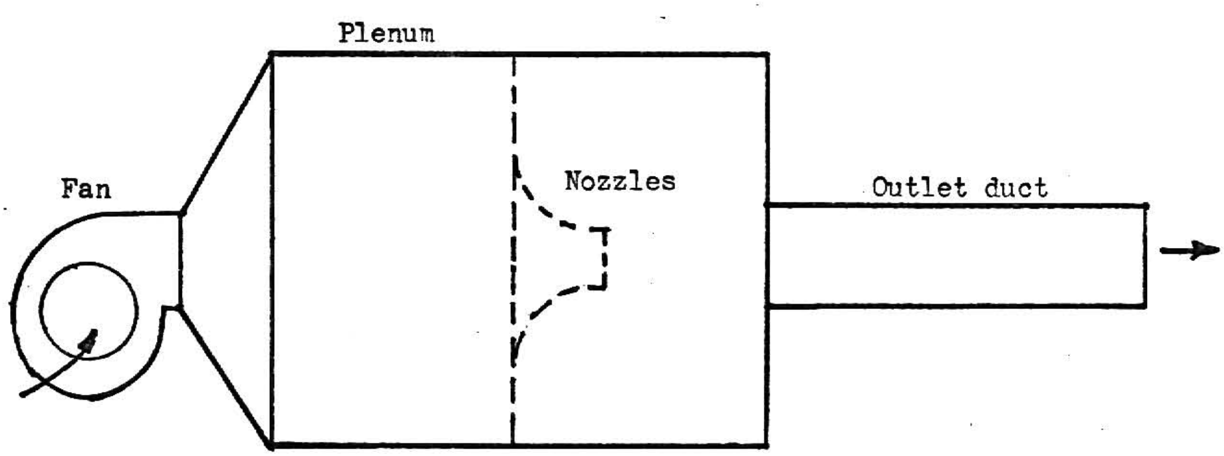


FAN PERFORMANCE CURVE

Figure 2



Drawthrough Arrangement



Blowthrough Arrangement

FAN-PLENUM RELATIONSHIP

Figure 3

plenum. This arrangement has two serious drawbacks, however. The plenum must have an access door for fan motor maintenance, which poses a sealing problem, and the heat given off by the motor enters the airstream.

These two problems are avoided with the fan-outside scheme, but two other disadvantages appear. Ductwork is required from the plenum to the fan housing, because of the negative pressure at the fan inlet. Any air entering or leaving the system downstream of the nozzles destroys the accuracy of measuring the flow rate.

2. Blowthrough - if the fan is located upstream of the plenum, these advantages accrue:

a. The heat given off by the motor has negligible effect on the airstream.

b. Air leakage around the fan shaft occurs upstream of the nozzles, before measurement takes place.

c. The motor is readily accessible for maintenance.

d. The fan section may be made removable for inserting a heating or cooling section between the fan and the plenum (Location C, Fig. 1).

e. The plenum is positively pressurized, rather than negatively, so any small leaks are more readily detectable.

Probably the most serious threat posed by this arrangement is the possibility of uneven or swirling airflow approaching the nozzles. Although the inlet plenum is very large compared to the largest nozzle, the relatively high discharge velocity of the fan might create uneven flow.

To minimize this problem, an evasé (diffuser) with an expanding included angle of 14° was designed for the fan discharge to enhance the conversion of discharge velocity pressure to static pressure. The relative velocities in the plane of the outlet of the fan section (location C, Fig. 1) were measured with and without the evasé to study its effect. The results of this study appear in Chapter IV, CALIBRATION PROCEDURES.

There are several ways to achieve variable flow rates (capacity control) with a centrifugal fan, such as speed control, variable inlet pre-rotation vanes, inlet or outlet dampers, or varying the flow vs. resistance characteristics of the system [4]. Speed control and pre-rotation vanes both affect the rotational speed of the fan wheel relative to the incoming air, which changes the fan performance curve. Figure 2 shows calculated values of flow vs. head for 2,000 rpm wheel speed, based on the normal speed of 2870 rpm. Prerotation vanes have a somewhat similar effect on the curve. However, speed control or inlet vanes for a system this small are not economical and would not cover the entire range of capacities.

Increasing the system resistance or using fan inlet or outlet dampers shifts the operating point upward along the fan curve toward the no-flow or "blocked off" point. At very low flow rates the fan would be operating in the unstable "dip" portion of the curve, which is undesirable.

To avoid these problems, the fan section is designed to include a spill damper downstream of the fan but upstream of the plenum (location B, Fig. 1). Thus the fan can deliver a relatively constant volume of air, but at low flow rates through the nozzles most of the air spills out of the fan section, bypassing the plenum and discharge sections. This method of capacity control would not be acceptable on the drawthrough system, because it becomes a leak after nozzle flow has been measured. However, leakage into or out of the system upstream of nozzle flow measurement is of no consequence. In this design, the spill damper function is performed by the access door to the inlet plenum. Even with the door closed, slight fan section leakage is unimportant.

Plenum Section

The ASME Power Test Code [3] specified dimensional tolerances for the installation of orifices and nozzles under various configurations of piping. For a nozzle discharging into a large enclosure or being supplied from a large enclosure, the Code stipulates the nearest wall parallel to flow shall be at least five nozzle diameters away, and the nearest wall perpendicular to flow shall be at least ten diameters away. These criteria established the plenum dimensions and the arrangement of the nozzles in the plenum partition, as will be shown later.

The selection of nozzles (instead of orifices) and nozzle sizes were based on the range of flow rates, on minimizing the total head requirements of the fan, and on the availability of components. For either a nozzle or an orifice, the loss of head across the device is a function of the beta ratio, defined as the ratio of the nozzle diameter (or orifice diameter) to the approach pipe diameter. For an approach plenum, the beta ratio is essentially zero, for which the head loss is 1.0 times the velocity head through the nozzle. However, whereas an orifice has a coefficient of discharge of about 0.6, a nozzle has a coefficient of discharge of 0.9 to 0.99, meaning that for a given flow rate and diameter, a nozzle will have lower exit velocity and resulting head loss than will an orifice.

Based on some trial calculations, the combination of nozzle sizes for the desired range of flow rates was determined to be 1/4", 1/2", 1", and 1.6". Table 2 shows various parameters for these nozzles at several selected flow rates.

The table shows the total range of flow rates can be covered using one nozzle at a time without having the nozzle velocity go below 649 ft./min. or above 3183 ft./min. By using the nozzles in combination, the upper limit on velocity is reduced to less than 2000 ft./min.

The plenum dimensions are derived from the ASME Code requirements previously mentioned. The ideal cross-sectional shape of the plenum would be circular, but the easiest shape to construct would be square. As a compromise, the plenum is designed to be octagonal. Based on the largest nozzle (1.6"), the octagon should measure 8" from center to edge, but when the other nozzles are added, this dimension becomes 10". The length of the inlet plenum is 16" (ten nozzle diameters), and the length of the outlet plenum is 19" (ten nozzle diameters, measured from the nozzle outlet). Figure 4 shows the layout of the nozzle partition, with the dotted circles indicating ten diameters from the center of each nozzle. It can be seen that no matter which nozzle is used, the nearest wall perpendicular to flow meets ASME requirements.

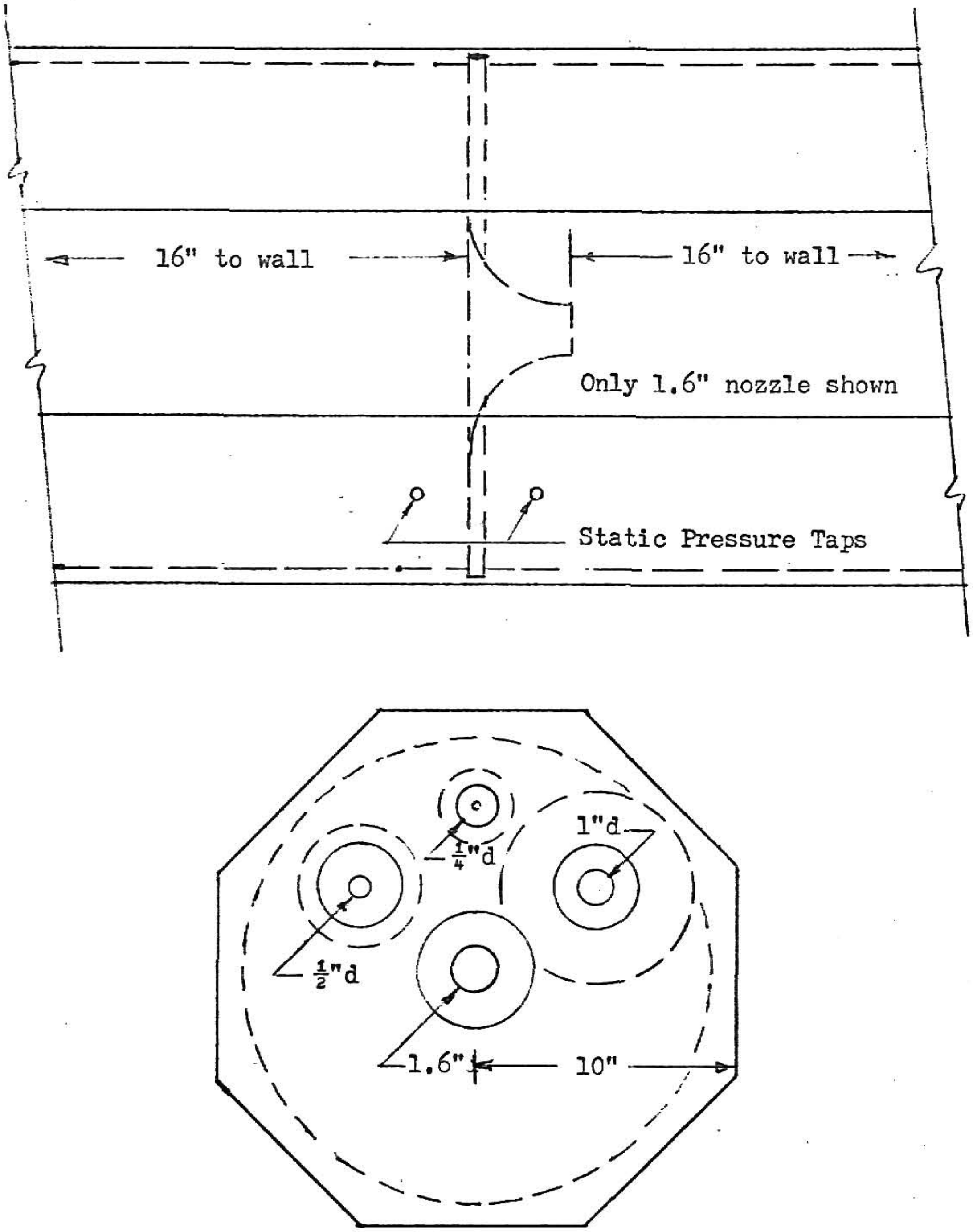
A static pressure tap is installed in each plenum as shown in Figure 4.

TABLE 2

FLOW CHARACTERISTICS OF FOUR ASME NOZZLES

Nozzle	Flow Rate (ft^3/min)	Velocity (ft/min)	Vel. Pressure ("H^2O)	Reynolds Number x 10^{-3}
	.22	649	.022	2.00
	.4	1180	.087	3.64
1/4"	.6	1770	.195	5.46
	.8	2360	.347	7.28
	1.0	2950	.543	9.10
	1.2	3540	.781	10.92
	1.4	4130	1.06	12.80
	1.0	735	.034	4.54
	1.5	1102	.076	6.80
	2.0	1471	.135	9.08
1/2"	2.5	1838	.211	11.34
	3.0	2206	.303	13.62
	3.5	2574	.413	15.89
	4.0	2941	.539	18.15
	4.0	735	.034	9.08
	6.0	1102	.076	13.62
	8.0	1471	.135	18.15
1"	10.0	1838	.211	22.69
	12.0	2206	.303	27.23
	14.0	2574	.413	31.76
	16.0	2941	.539	36.30
	15.0	1074	.072	21.22
	20.0	1433	.128	28.32
	25.0	1791	.200	35.39
1.6"	30.0	2149	.288	42.46
	35.0	2507	.392	49.54
	40.0	2865	.512	56.61
	43.0	3183	.646	63.35

Note: Velocities, velocity pressures and Reynolds numbers calculated; flow rates selected arbitrarily.



LAYOUT OF NOZZLE PARTITION

Figure 4

<u>Outlet Section</u>

The outlet section consists of a nominal 4" diameter duct, straightening vanes, and a 2" long-radius ASME nozzle. Although Figure 1 schematically shows the outlet duct connected to the downstream end of the outlet plenum, the actual configuration positions the duct parallel to and alongside the plenum, to save space. This places the 2" outlet nozzle close to the fan section and requires a connection between the side of the outlet plenum and the side of the duct. Figure 5 shows the relative position of the plenum and duct.

Sizing the connection from plenum to duct is somewhat arbitrary, except construction simplicity suggests a rectangular shape and one dimension is made equal to the inside diameter of the outlet duct (about 3.8"). Using equation [1], page 8, and assuming the other dimension at 10" gives:

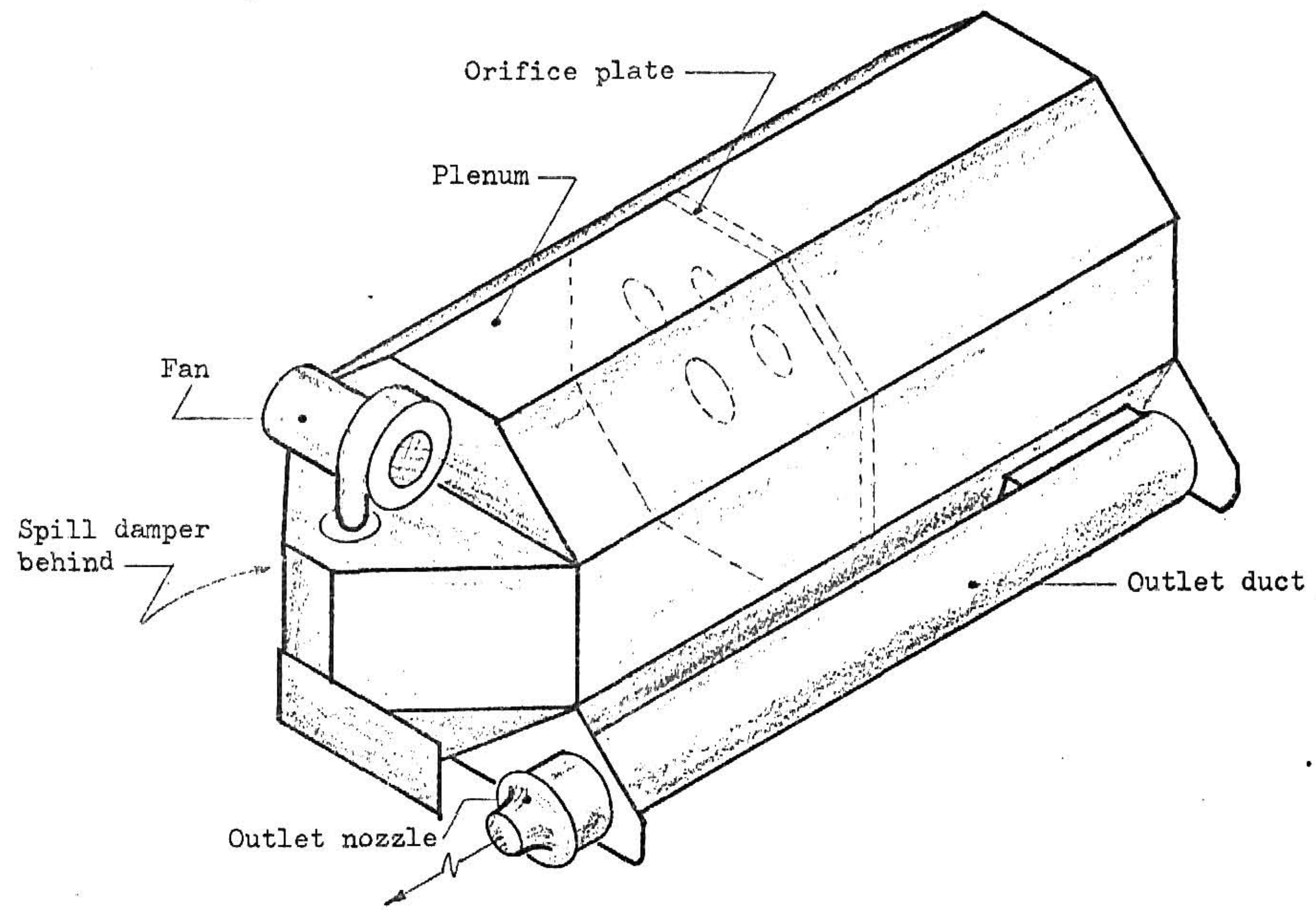
$$43 = \frac{(10 \times 3.8)}{144} \times V$$

$$V = \frac{43 \times 144}{38} = 163 \text{ ft./min.}$$

This average velocity has a velocity pressure so low (0.0016") that the dynamic loss through this connection is acceptable.

Ideally, the air approaching the outlet nozzle would have a flat velocity profile with no twisting component. This condition can be achieved with a very long duct preceded by straightening vanes, but it can be approximated by having straightening vanes followed by a duct 10 diameters long [11].

There are several recommended designs for straightening vanes [3, 11, 14].



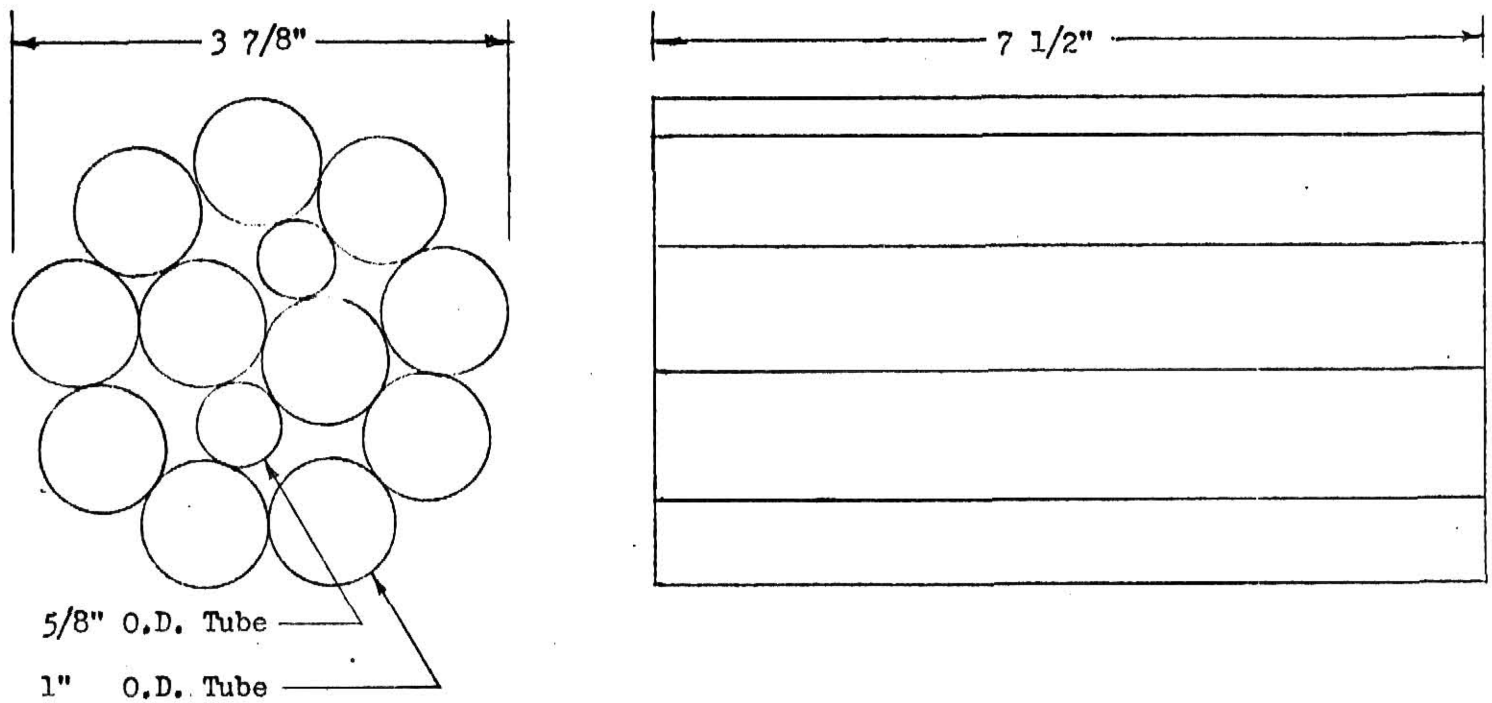
ISOMETRIC VIEW OF CALIBRATOR

Figure 5

This calibrator includes vanes made approximately to ASME specifications. The vanes are 7 1/2" long and consist of eleven 1" outside diameter tubes and two 5/8" outside diameter tubes, arranged as shown in Figure 6. The vanes are positioned such that there are 4.9 diameters of duct downstream of the leaving plane of the vanes.

At the maximum flow rate, the average velocity in the outlet duct is about 500 ft./min., and at the minimum, 2 1/2 ft./min. Thus the velocity pressures and friction losses in the discharge duct are not appreciable compared to the loss through the calibration nozzles.

The holding bracket for the probe of a meter being calibrated by this device is located near the outlet of the discharge nozzle, close enough to the plane of the nozzle outlet that the free stream velocity around the probe is equal to the velocity leaving the nozzle. This is discussed more thoroughly in Chapter IV, CALIBRATION PROCEDURES.



STRAIGHTENING VANES

Figure 6

CHAPTER III

CONSTRUCTION METHODS AND MATERIALS

This calibrator is constructed mostly of plywood, except for the discharge duct, which is commercial polyvinylchloride schedule 40 pipe, and the purchased items, such as fan, nozzles, wiring, and miscellaneous hardware.

Wood was selected for the fan and plenum sections for ease in fabrication, durability, low thermal conductivity, light weight, noise attenuation, and availability.

The nozzle plate is recessed to receive the nozzles so that the inlet face of each nozzle is flush with the wood to minimize turbulence. Each nozzle is set into its recess in silicone rubber.

The plenum is glued together and all interior joints are caulked with silicone rubber to avoid leakage. The interior is painted white to aid visibility and to seal the wood against moisture. The exterior of the plenum is sealed with a clear primer, to which another finish (paint, stain or varnish) may be added if desired.

The calibrator can be operated in either horizontal or vertical (discharge up) position. There are extensions at each end of the plenum to prevent rolling in the horizontal position, and two of these extensions also hold the discharge duct in place. When the plenum is vertical, two of the extensions and a lifting handle serve as feet.

The fan section is made of plywood, and includes the fan mounting bracket, an access door, a removable evase, the discharge nozzle and a holding bracket for the probe of a meter being calibrated. The fan section houses the wiring to the motor, an on-off switch, a fuse, and a terminal strip. The interior is caulked, except at the access door, and painted white. The exterior is sealed and painted.

The fan section is mounted on the plenum section by six wing nuts which are reached through the access door. The fan is removable without disconnecting any wiring by means of a Jones plug mounted in the fan platform.

The fan assembly is so arranged that the motor shaft is horizontal in either operational position of the plenum, and the oil cups for the motor are within 45° of the top in either operational position.

A future heating or cooling section may be inserted between the fan section and the plenum section by bolting the new section to the plenum with wing nuts and bolting the fan section to the new section. There is room on the terminal strip to accommodate additional wiring if the heating section is electric.

The access door serves the dual function of spilling excess air for low flow through the nozzles, as previously discussed, and making inspection and plugging of the nozzles possible. (At low flow, the larger nozzles are plugged.)

The discharge duct is a 40 1/2" length of nominal 4" PVC white schedule 40 pipe. The inside diameter is 3 7/8"; the outside diameter is 4 1/4". This material was selected because the interior is smooth, the material is light and easy to work, and the thermal conductivity is less than sheet steel or aluminum.

One end of the duct is plugged with 1/2" clear plastic, which affords a view of the straightening vanes and the outlet side of the nozzles. The 10" x 3 7/8" outlet opening in the side of the plenum is joined to the 10" x 3 7/8" hole in the duct by plywood which is glued and caulked to prevent leaks.

The straightening vanes are made of aluminum thin wall electrical conduit. The tubes are held together by epoxy glue and held in place in the duct by a force fit.

All of the nozzles are polished spun aluminum, made to ASME specifications. The discharge nozzle has been trimmed to meet space limitations, but not enough to affect its performance.

The static pressure taps in the plenum are made of 1/8" copper tubing, and are mounted flush to the inside of the plenum.

The entire assembly weighs 42 pounds and measures 45 1/2" long, 23" wide, and 22" high.

CHAPTER IV

CALIBRATION PROCEDURES

Collection of Data

The calibration of this device required two separate steps. The first step was to establish the relationship between flow rate through the nozzles and pressure drop across them. As mentioned earlier (p. 4), there are published data about this relationship for nozzles and orifices under a variety of conditions, taking into account velocity of approach, compressibility, and so forth, so orifices and nozzles usually are used without further calibration. However, the low flow rates in this work and the shape of the plenum suggested calibration would be desirable. Further, using two or more nozzles in parallel seemed to justify measuring their performance.

The second step involved determining the velocity profile at the outlet nozzle so that centerline velocity (or some other arbitrary reference point) could be calculated. This velocity is the useful parameter for using the calibrator.

Two techniques were used for establishing the flow versus pressure drop relationship. At flow rates above 12 ft^3/min, pitot tube traverses of the output nozzle were used to measure velocity head, from which flow rate was calculated. At the same time, static pressure drop across the nozzles was measured, from which the flow vs. pressure drop correlation was established. This procedure is discussed in more detail later.

At low flow rates (12 ft^3/min or less), a calibrated positive displacement bell prover generated the airflow through the nozzles and the pressure drop across the nozzles was measured with a hook gage, which reads to the nearest 0.001" of water.

A bell prover is used commonly by natural gas utility companies to calibrate gas flow meters. It consists of a cylinder of accurately known volume inverted and suspended in a tank of oil, which functions as a liquid seal. The cylinder is raised to a calibrated height such that the open end of the cylinder stays submerged in the oil. The cylinder contains air at slightly more than atmospheric pressure (measured by a U-tube manometer mounted on the frame of the prover). A system of weights, cable, pulley, and guide rods allows the cylinder to settle, expelling the known volume of air in the cylinder through an output pipe during a timed interval, from which flow rate is computed. The expelled air is fed into the inlet plenum of the calibrator through a blank-off plate mounted in place of the fan section (location C, Fig. 1) and sealed off to prevent leaks. Thus a known quantity of air flows through the nozzles in a measured time.

The rate at which the cylinder settles is controlled by the amount of counterweight used.

The prover that was used has a total cylinder volume of five cubic feet, but only the middle four (from 1/2 to 4 1/2) was used for calibration. A typical run was conducted as follows:

1. Hook gage leveled and zeroed. Tubing from gage to plenum purged of water and connected.

2. Prover connected by 1" diameter tubing to plenum through blank-off plate.

3. Cylinder raised to top position; cylinder pressure (in inches of water), barometric pressure, and ambient temperature recorded.

4. Prover discharge valve opened, stop watch started at 1/2 ft^3 mark, hook gage adjusted to differential water height, discharge air temperature read, stop watch stopped at 4 1/2 ft^3 mark.

   a. In general, if the total settling time for the cylinder was less that 60 seconds, one time interval was recorded for the entire 1/2 - 4 1/2 ft^3 stroke.

   b. For slower settling rates, two stop watches were used, and times recorded for incremental volumes during the stroke, such as 1/2 - 2 1/2 (two ft^3), 2 1/2 - 4 1/2, or 1/2 - 1 1/2, 1 1/2 - 2 1/2, etc. The smallest timed increment was 1/2 ft^3. This procedure was used to detect non-uniform rates of settling of the cylinder.

Repeated runs demonstrated that the settling rate was uniform if the total elapsed time was less than about three minutes. As the elapsed time increased, smaller increments were timed, and non-uniform settling was detected.

The slowest settling rate measured was 564.0 seconds (9 minutes 24 seconds) to sweep 3 ft^3 through the 1/4" nozzle. Each 1/2 ft^3 increment was timed; the first 1/2 ft^3 measured 97.6 seconds, and the last measured 70.8 seconds.

Within any increment, the hook gage was adjusted during the middle of the time, and this value was assumed to be the average for the increment.*

Subsequent justification for this procedure appeared when the data were treated statistically to establish a correlation. This is discussed in more detail later. Further, at low flow rates (long settling times) the differential head across the nozzle is very low (approximately .050"), and it takes time to get an accurate adjustment of the hook gage.

The range of flow rates and nozzle combinations which were tested by this procedure is as follows:

1/4" nozzle - - - - - - - - - - - - - - 0.3 to 2.2 ft^3/min
1/4" + 1/2" nozzles - - - - - - - - - - 1.62 to 6.82 ft^3/min
1/4" + 1/2" + 1" nozzles - - - - - - 5.43 to 11.90 ft^3/min

The upper limit on the range of flow rates using the bell prover is influenced by two factors:

1. The settling time must be long enough to allow the hook gage to reach equilibrium.
2. The pressure in the cylinder must not get high enough to cause air to bubble out through the oil rather than through the discharge pipe.

The lower limit is reached when the addition of more counterweight plus the friction in the pulleys and guide rods prevents the cylinder from settling.

*This procedure is discussed in Reference 5.

For flow rates greater than 12 ft^3/min, the fan section of the calibrator served as the air source and velocity pressures at the discharge nozzle were measured with a pitot tube. This procedure is well documented /‾3, 11, 14_7 and will not be discussed here.

Treatment of Data

One necessary criterion in dealing with fluid flow data is to establish "standard" conditions. For air, various standard conditions have been defined, such as "dry air at 70°F, 29.92" Hg" or "air with a density of 0.075 lb/ft^3 and absolute viscosity of 1.225×10^{-5} lbm/ft sec", or "dry air at 59°F at sea level". For this work, standard air is considered to have a density of 0.075 lb/ft^3, which is essentially dry air at 70°F, 29.92" Hg. By comparison, saturated air at the same conditions has a density of 0.07425, so the effect of moisture content is, at most, less than 1% at this temperature and pressure.

It is shown in equation /‾19_7, page 63, that the relationship between velocity and velocity pressure depends on the density of the air. The calibration tests on the nozzles in this apparatus were performed over several days, during which the density of the air was not constant and not necessarily 0.075 lb/ft^3. Therefore, all of the data have been normalized by calculating what the flow rate would be for standard air at the measured pressure drops across the nozzle(s).

The measured flow rate was normalized as follows from equation /18 /, p. 63 where $q = 1096.7\ KA\ (h/d)^{1/2}$. It can be shown that, for a given measured velocity head,

$$q_{norm} = q_{actual} (d_{actual} /.075)^{1/2} \qquad [2]$$

where q_{norm} is normalized flow rate

q_{actual} is measured flow rate

d_{actual} is density of air of measured flow rate conditions

.075 is density of standard air

Fan Engineering [11] gives an equation (p. 11) for calculating the density of an air-vapor mixture as follows:

$$d = \frac{(b - e_w h)\ SG + e_w hs}{.7538\ (t + 459.7)} \qquad [3]$$

where d is density in lb/ft^3

b is barometric pressure in "Hg

e_w is vapor pressure of water at temperature t in "Hg

h is relative humidity (no units)

SG is specific gravity (1.0 for air)

s is relative density of water vapor (no units)

t is dry bulb temperature in ^oF

This equation is of the form $\frac{m}{V} = \frac{P}{RT}$ (the ideal gas equation). The numerator of equation [3] is the sum of the partial pressures of the dry air and the water vapor, and the denominator converts ^oF to ^oR and includes the value of R in the appropriate units. The specific gravity term (SG) appears to accomodate gases other than dry air, but in this case (for air) it has the value 1.0.

The relative density (s) is the ratio of the density of water vapor to dry air, and is used because water vapor does not behave exactly as an ideal gas. The value for s can be computed from the following equation (Fan Engineering, p. 7):

$$s = 0.6214 + \frac{h(e_w)^{1/1.42}}{1130} \qquad [4]$$

where the symbols are the same as in equation [3]. If the vapor behaved as an ideal gas, s would be 0.622 (the ratio of the molecular weights of water and air). The correction is very slight for moderate temperatures and relative humidities. The following example will show this.

Measured: $t = 66^{\circ}F$; $b = 29.30"$ Hg; $h = 50\%$ (assumed)

$$s = .6214 + \frac{(.5)(.6442)^{1/1.42}}{1130} = .6214 + \frac{(.5)(.737)}{1130}$$

$$= .6214 + .000326 = .621726$$

Because the pressure of the air in the bell prover is slightly above barometric pressure, the bell pressure must be added to b to get the absolute pressure in the bell. In this case, bell pressure was measured as 0.65" water, or .0475" Hg. Thus

$$d = \frac{(29.30 + .047) - (.644)(.5)(1.0) + (.6442)(.5)(.621726)}{(.7538)(66 + 459.7)}$$

$$= \frac{(29.30 + .047 - .322) + (.201)}{396.2} = .0737 \text{ lb/ft}^3$$

Several conclusions may be drawn from these results:

1. Assuming a value for h (relative humidity), especially in the range of 40% to 60% - typical of most indoor environments - introduces very slight inaccuracy. The second and third groups in the numerator

reduce to: $-e_w h + e_w h(.622) = -.378\ e_w h$. This entire term represents about 1% of the numerator, so missing relative humidity by, say, 10% introduces an error of 0.1%.

2. The influence of the bell pressure is similarly small. Even at the highest bell pressure measured during calibration runs, ignoring this term introduces an error of 0.4%.

3. Calculating s from equation [4] is unnecessarily accurate at normal temperatures and relative humidity.

4. The major influences on density at normal temperatures are dry bulb temperature and ~~barometric~~ pressure.

In computing the air density during the calibration runs, the following equation was used, based on the above conclusions:

$$d = \frac{b - 0.189e_w}{(.754)(t + 460)} \qquad [5]$$

Table 3 shows the results of these calculations of air density for all of the calibration runs. Runs 1-4 were made with bell prover. Runs 5-8 involved velocity pressure traverses where airflow was provided by the fan section.

The saturated vapor pressure, in inches Hg, is the e_w term in equation [5]. It is determined from steam tables for saturated water at dry bulb temperature t [8].

TABLE 3

CALCULATION OF AIR DENSITY

Run number	1	2	3	4	5	6	7	8
Nozzles	1/4	1/4	1/4 + 1/2	1/4 + 1/2 + 1			All	All
Barometric pressure, "Hg	29.30	29.27	29.21	28.89	28.96	28.96	28.96	28.96
Temperature, ^{o}F	66.2	69.0	69.0	72.8	77.0	78.8	81	82
Assumed relative humidity (%)	50	50	50	50	50	50	50	50
Saturated vapor pressure, "Hg	.648	.714	.714	.807	.935	.967	1.066	1.102
Calculated density, lb/ft^3	.0735	.0730	.0728	.0715	.0716	.0715	.0705	.0704

Note: Runs 1-4 were made with the bell prover. Runs 5-8 had airflow provided by the fan section.

The next step was to establish a correlation between normalized flow rate through the nozzles (which is proportional to the mean nozzle velocity) and measured pressure drop. One form which the correlation might take is suggested by the kinetic energy term of the steady state incompressible flow general energy equation (Appendix D, p. 61). $V^2/2g = h$. The ASME Code [3] gives the head loss through a nozzle as a percentage of the velocity head at nozzle outlet where the percentage varies with the ratio of nozzle diameter to approach diameter (beta ratio). Thus the form of the correlation might be:

$$y = ax^b \qquad [6]$$

where y is pressure drop, a and b are constants, and x is flow rate.

According to the general energy equation, the value for b should be 2.0 and the value for a depends on nozzle area, system geometry, Reynolds number, and the units of the various terms.

A computer program was used to calculate a and b and correlation coefficient, r, for equation [6]. Table 4 shows the results of the computer calculations for values of a, b, and r. The equation for r can be found in most books on probability and statistics.

TABLE 4

FITTING DATA TO A POWER EQUATION

	Nozzle Size	a	b	r	Number of Runs
(1)*	1/4"	.4069	1.779	.998	42
(1)	1/4" + 1/2"	.0250	1.901	.999	14
(1)	1/4" + 1/2" + 1	.001556	1.982	.999	3
(2)	1/4" + 1/2" + 1				4
(2)	1/4" + 1/2" + 1" + 1.6"	.0001638	2.0175	.999	4

* (1) Bell prover data (2) Pitot traverse data

Although fewer runs (at different flow rates) were made using the pitot traverse method, 24 readings of velocity pressure were taken for each run (12 each of two diameters), so the amount of raw data was adequate to minimize the effect of random measuring errors [10].

The equations expressing the flow rate vs. head loss for the nozzle combinations thus becomes:

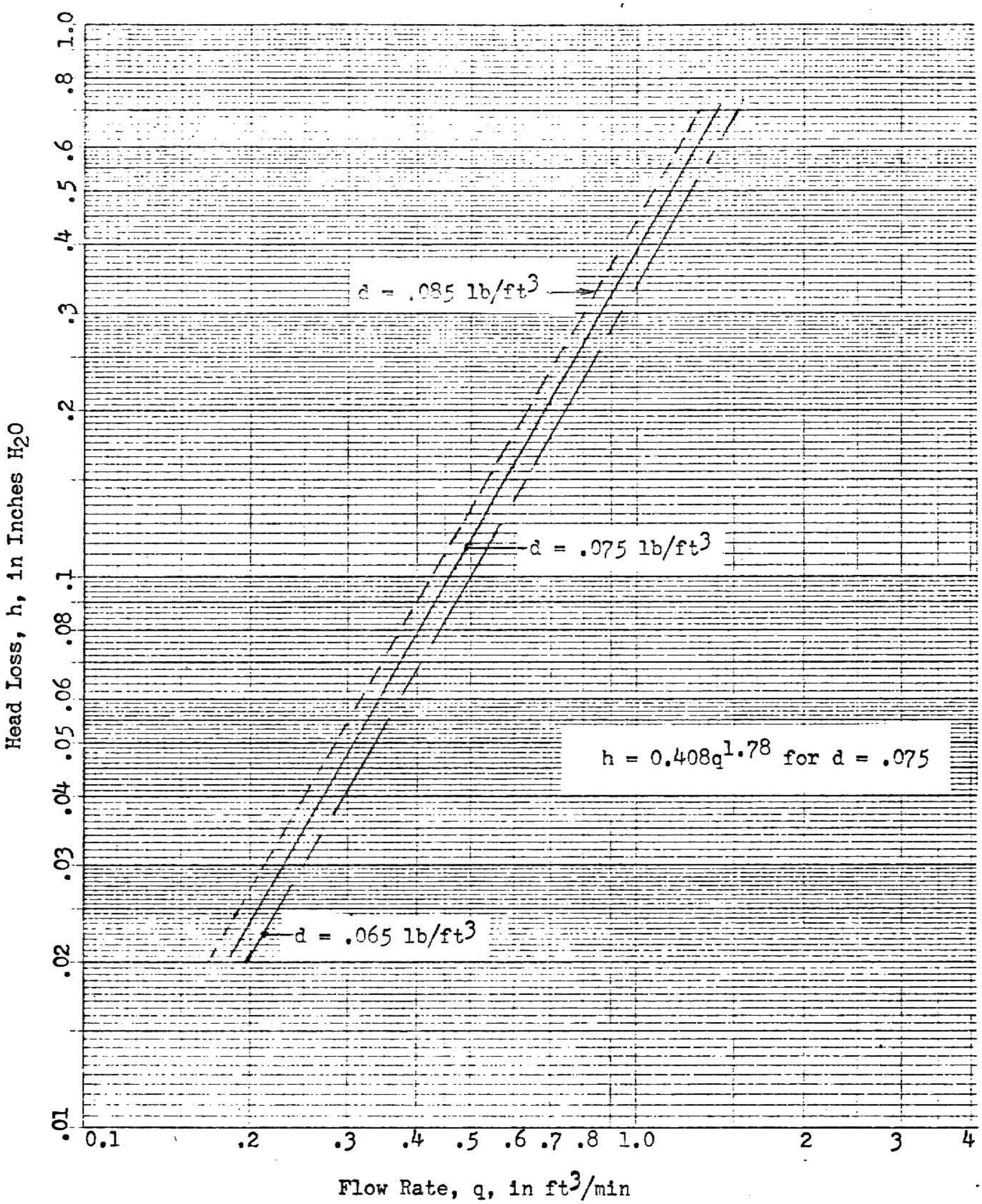
Run	Nozzles	Equation
1,2	1/4"	$h = .407\ q^{1.78}$
3	1/4" + 1/2"	$h = .025\ q^{1.90}$
4,5,6	1/4" + 1/2" + 1"	$h = .00156\ q^{1.98}$
7,8	1/4" + 1/2" + 1" + 1.6"	$h = .000164\ q^{2.02}$

where h is the loss in head and q is the flow rate. Although the normal operating procedure will be to measure h to determine q (and then outlet velocity), the data are presented here as h being a function of q because the calibration procedure was based on that premise.

Figures 7 and 8 show the graphical representation of the above equations, plotted on log-log paper with q being considered the independent variable. The solid line in Figure 7 is the equation at standard (.075) density, and the dotted lines show two other densities. Determination of other densities is made using equation [4].

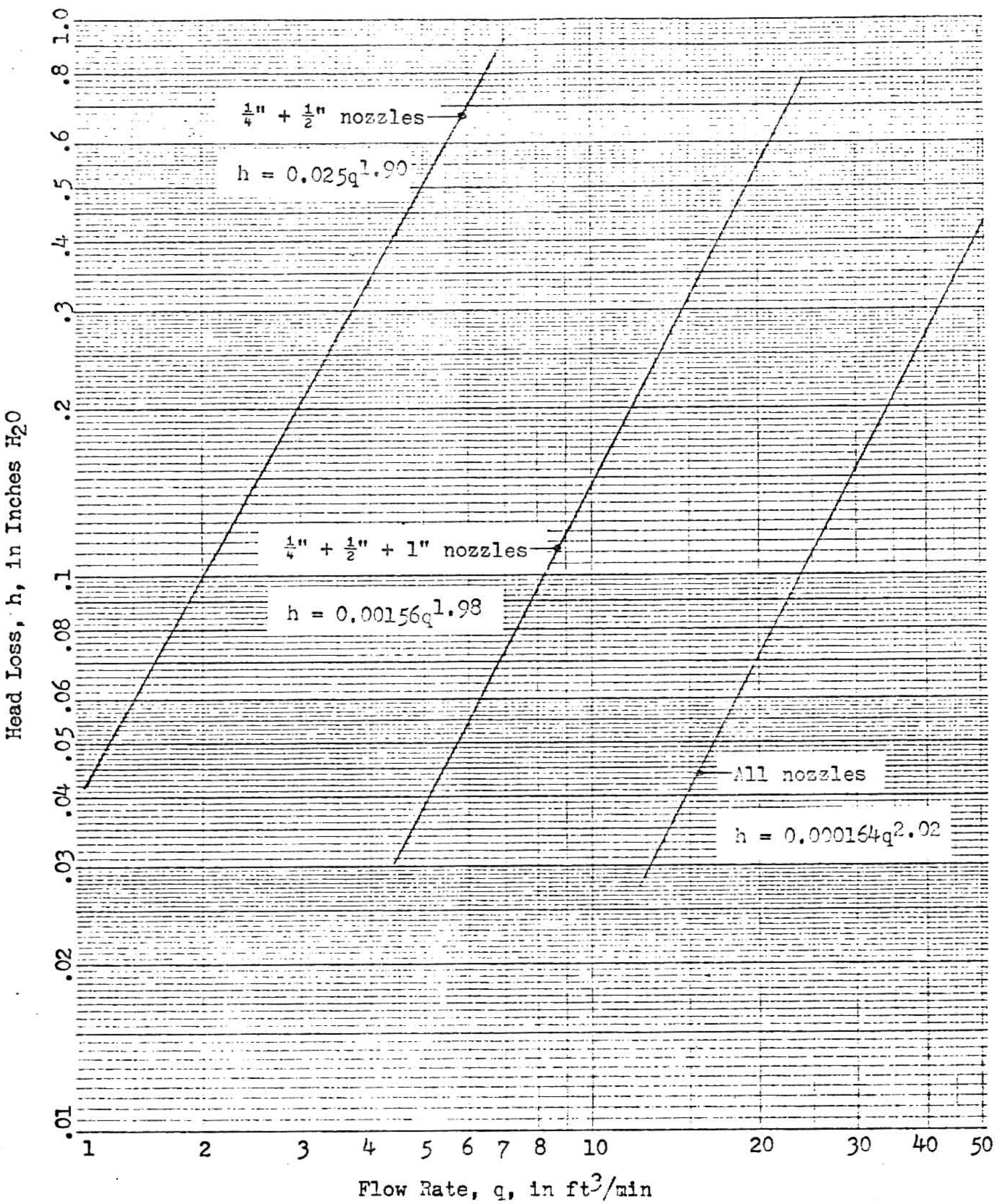
Once the flow rate vs. head loss relationship was established, the second step referred to on page 26 was to determine the velocity profile at the outlet nozzle for various flow rates. If the outlet velocity were constant across the nozzle outlet, then $v = q/A$.

However, the velocity is not constant across any outlet because of the velocity gradient at the boundaries and because the velocity profile approaching the outlet may be influenced by upstream disturbances. In this case, having only



HEAD LOSS VS. FLOW RATE, 1/4" NOZZLE

Figure 7



HEAD LOSS VS. FLOW RATE
FOR THREE NOZZLE COMBINATIONS

Figure 8

five duct diameters downstream of the straightening vanes would make the outlet profile suspect.

The outlet velocity conditions were determined two different, but related, ways. Where the calibration procedure made use of pitot traverses to determine flow rate, velocity pressure was measured at 12 points on each of two diameters.

It is not necessary to calculate the velocity at each pitot position in order to determine flow rate, but the mean velocity, $\bar{V}$, must be determined, which is then used in the equation $q = \bar{V} \times A$, to get flow rate. Mean velocity is calculated as follows:

$$\bar{V} = 1096.7 \times \frac{1}{n} \quad (h_1/d)^{1/2} + \dots (h_n/d)^{1/2} \qquad [7]$$

$$= \frac{1096.7}{nd^{1/2}} (h_1{}^{1/2} + h_2{}^{1/2} \dots h_n{}^{1/2})$$

This procedure used twelve readings on each diameter, so $n = 24$. The positions of the pitot tube relative to the centerline (where radius = 0") are shown below.

Position Number	Distance from Center	Actual Position
(centerline)	0"	——
1	.288"	.29"
2	.500"	.50"
3	.646"	.645"
4	.764"	.765"
5	.866"	.865"
6	.958"	.960"
(Nozzle Wall)	1.0"	

The actual positions used reflect the maximum precision possible on the probe-positioning device. Reference will be made later about the number 6 position being within 4% of the nozzle wall.

Velocity pressure readings were taken at these points on two diameters with a small pitot tube and a micromanometer. The probe of the pitot tube was positioned approximately 1/4" from the plane of the nozzle outlet. The ASHRAE Handbook of Fundamentals [9] gives four diameters (4D), as that distance from the outlet plane within which the maximum airstream velocity is unchanged. In this case, the pitot tube was at 1/8D, and the probe of a commercial meter being calibrated will be at about 3/8D, both well within the constant velocity zone.

The velocity pressure was also taken at the centerline, although this value was not used in equation [7]. However, the data permit comparison of velocity values at various positions across a diameter. Table 5 shows comparisons of mean velocity, $\bar{V}$, and centerline velocity, V_{cl}, for all of the pitot tube runs.

The ratio, $\bar{V}/V_{cl}$ is used to determine the actual velocity across the probe of a meter being calibrated (V_{cl}) from the mean velocity $\bar{V}$ which is calculated from equation 1. This procedure is described in more detail in Chapter V, RECOMMENDATIONS.

TABLE 5

RELATION OF CENTERLINE AND MEAN VELOCITIES
(Pitot Tube Method)

Run	Nozzles	h_{cl}*	d	V_{cl}*	$\bar{V}$*	$\bar{V}/V_{cl}$
5	1/4,1/2,1	.055"	.0716	917	904	.986
		.016	"	495	471	.952
		.013	"	446	411	.922
6	1/4,1/2,1	.023	.0715	593	566	.954
7	All	.229	.0705	1858	1774	.955
		.226	"	1870	1825	.976
8	All	.135	.0704	1426	1394	.978
		.071	"	1033	1011	.979

* h_{cl} is head at the centerline

V_{cl} is velocity at the centerline

$\bar{V}$ is mean velocity (computed)

At the lower flow rates, where the bell prover was the calibrated air source, a slightly different procedure was used. The DISA low velocity anemometer was used to determine the velocity profile from which the relation of centerline velocity, V_{cl}, to mean velocity, $\bar{V}$, was determined graphically. Meter readings (not necessarily ft/min) were taken along a diameter inward every .02" from the outlet nozzle wall until the readings were substantially constant. The ratio V/V_{cl} was plotted versus $(r/R)^2$, where r is the position at which V was read ($r = 0$ at $V = V_{cl}$), and R is 1.0". By graphic integration, a constant is determined which relates $\bar{V}$ to V_{cl} [8].

The results of $\bar{V}/V_{cl}$ vs. pressure drop, h_L, are plotted for all nozzle combinations in Figure 9.

Interpretation of Data

The logical and inevitable question to be asked about any data is whether or not they are accurate. Accuracy implies the degree to which the model (numbers, curves, equations) predicts and conforms to the behavior of the device or phenomenon it attempts to model. The amount of doubt raised about this conformity and predictive power is called uncertainty. The uncertainty is the combined effect of random errors in the apparatus and the random errors of data-gathering.

Uncertainty here can arise in any one of three phenomena - the flow rate, the measured pressure drop across the nozzles, and the outlet centerline velocity. Each of these was analyzed with the intent of quantifying the probable uncertainty in the values of each.

Reference 15 gives an equation for estimating the uncertainty, E_R, of the result, R, of some function of independent variables X and Y. That is, $R = f(X,Y)$.

$$(E_R)^2 = (S_x E_x)^2 + (S_y E_y)^2 \qquad [8]$$

where E_R is the uncertainty of the result

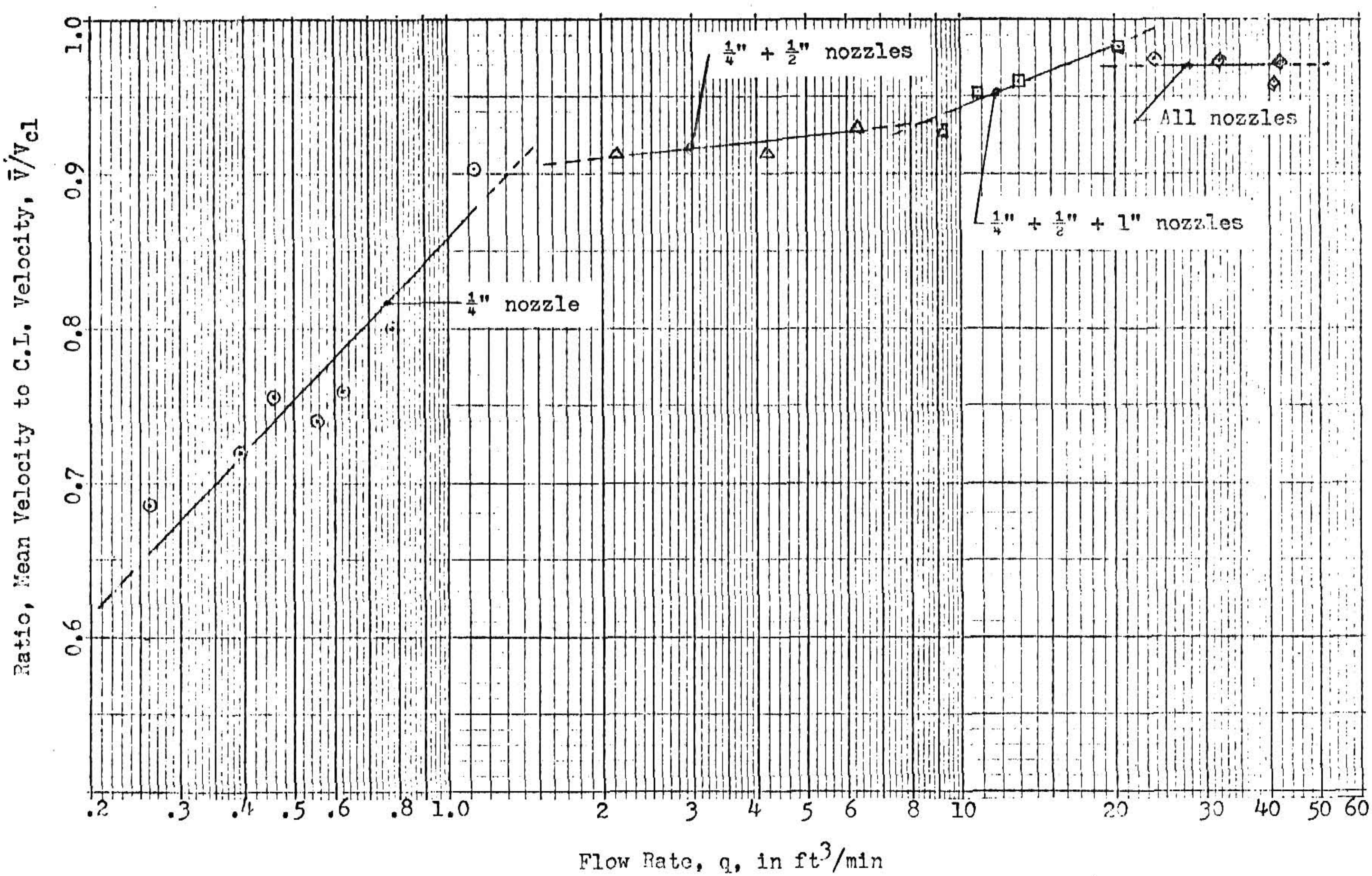
E_x is the uncertainty of X

E_y is the uncertainty of Y

S_x is the sensitivity factor of X

S_y is the sensitivity factor of Y

See reference 15 for the derivation of [8] and the definition of S.



CENTERLINE VELOCITY CURVE

Figure 9

The sensitivity factor of a variable depends upon its functional relationship to the result; that is, the equation by which the result can be determined from known values of the variables.

This procedure can be illustrated in the determination of flow rate by the bell prover. Two instruments (prover and stopwatch) and two readings (reading of prover and stopwatch) are involved, and the functional relationship is the standard rate equation, flow rate = volume/time. Prover volume has 0.3% uncertainty and reading the prover pointer has 0.2% uncertainty. The sensitivity of volume on flow rate is 1.0. From equation [8], the uncertainty of determining volume is 0.36%. The stopwatch uncertainty is 0.03%, but the uncertainty of reading the stopwatch depends on the measured time interval. For the shortest interval (17 seconds), the uncertainty is 1.1%, but for the timed interval at the lowest flow rate (97 seconds), the uncertainty is 0.2%. The sensitivity factor of time on flow rate is -1.0. Combining all of the above in equation [8] yields an uncertainty of flow rate of 0.41% at the lowest flow rate (about 0.3 ft^3/min) and 1.2% at the shortest measured time (6.8 ft^3/min).

Similar analyses were made on air density, nozzle pressure drop, and centerline velocity (DISA and pitot traverse). Table 6 shows the results of these calculations. Finally, the uncertainty of centerline velocity V_{cl}, versus pressure drop across the nozzles, h_L, is plotted in Figure 10.

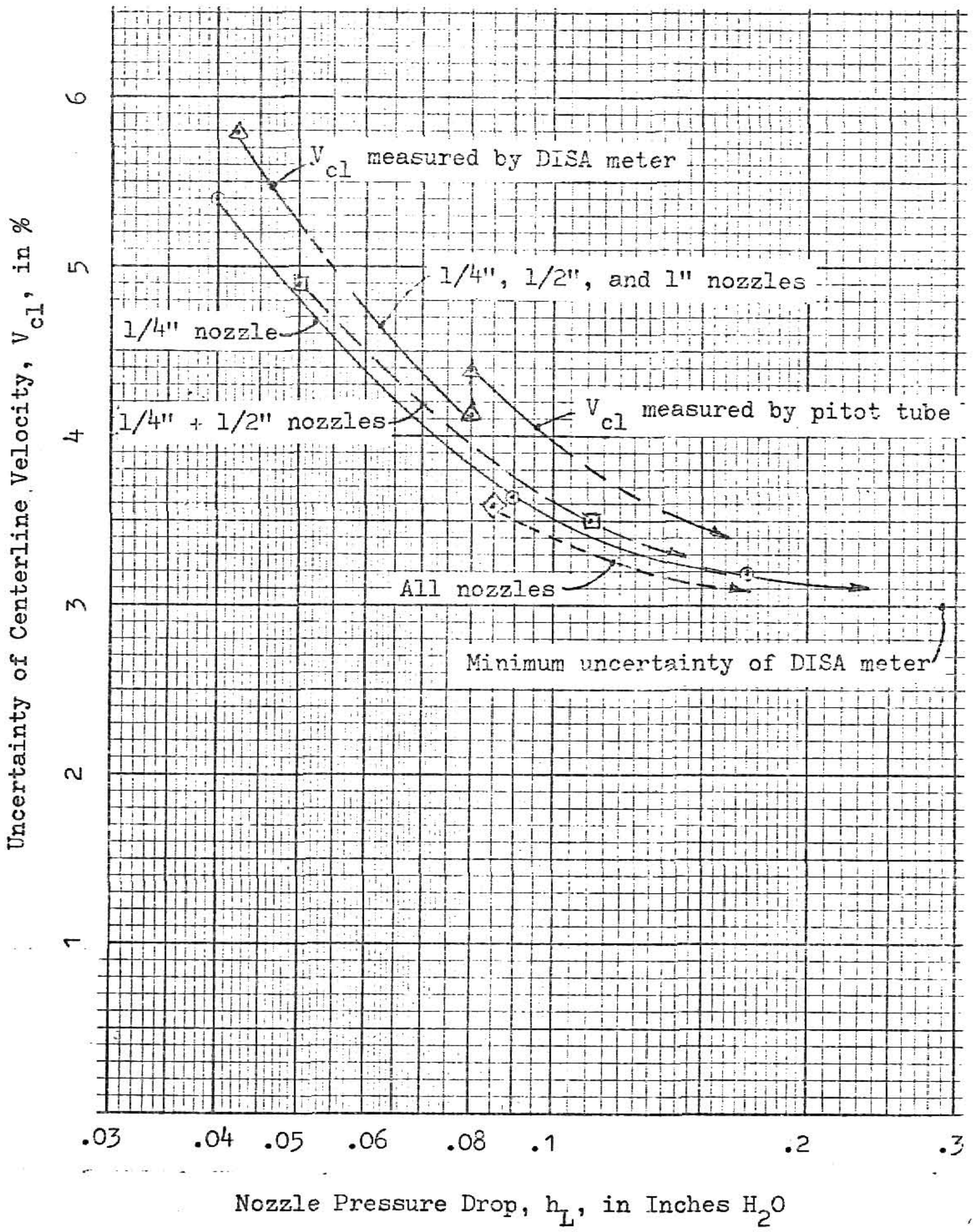
TABLE 6

UNCERTAINTIES AND SENSITIVITIES

Parameter	Instrument and Reading Uncertainty	Sensitivity Factor	Uncertainty of Result
A. Air Density			
1. Barometric pressure			
a. Barometer	0.24%		
b. Equation		1.005	
2. Relative humidity			
a. Estimated value	10.0%		
b. Equation		-0.005	
3. Temperature			
a. Thermometer	1.0%		
b. Equation		-0.132	
4. Result			0.3%
B. Flow Rate			
1. Volume			
a. Bell Prover	0.36%		
b. Equation		1.0	
2. Time			
a. Stopwatch at lowest flow rate	0.20%		
b. Stopwatch at shortest time	1.10%		
c. Equation		-1.0	
3. Result (lowest flow rate)			0.41%
(shortest time)			1.20%
C. Flow Rate vs. Nozzle Pressure Drop			
1. Hook gage			
(lowest flow)	2.50%		
(shortest time)	0.10%		

TABLE 6 (Cont'd)

2. Flow rate (from B-3)	0.41% 1.20%		
3. Regression equation			
(lowest flow)		1.78	
(shortest time)		1.90	
4. Result (lowest flow)			4.5%
(shortest time)			1.2%
D. Velocity Profile			
1. Pitot Tube (10 to 43 ft^3/min)			
a. Micromanometer	6.25%		
b. Positioning pitot tube	0.50%		
c. Velocity head equation		0.5	
d. Result			3.2%
2. DISA meter (.3 to 7.0 ft^3/min)			
a. Meter	1.40%		
b. Positioning probe	0.50%		
c. Equation [Ref. 8]		2.0	
d. Result			3.0%
E. Determining Centerline Velocity from Flow Rate (lowest flow)			
1. Flow Rate	4.5%		
2. Density correction	0.3%		
3. Velocity profile	3.0%		
4. Result			5.4%



CENTERLINE VELOCITY UNCERTAINTY CURVE

Figure 10

CHAPTER V

RECOMMENDATIONS

The user of this calibrator must be aware of its characteristics and limitations. Some suggestions for its use are listed below.

The typical user will measure head loss and want to know flow rate. Therefore, the least squares computer program was used to fit a curve to the data in this fashion:

$$q = C\ h^D \qquad [9]$$

where q is flow rate in ft^3/min

h is head loss in $"H_2O$

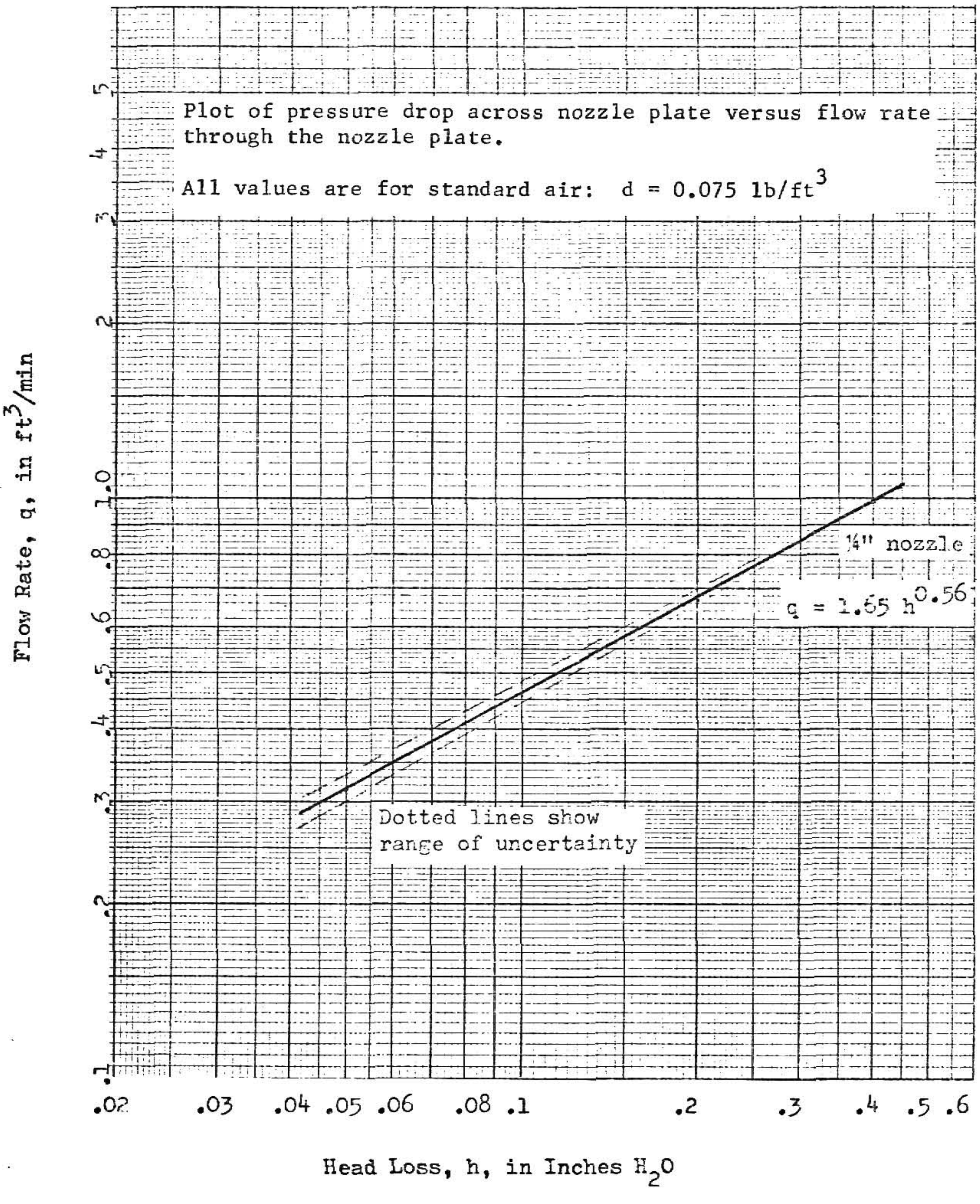
C,D are constants

These equations and their plotted curves appear in Figures 11 and 12. It is important to note that flow rate is in "standard" ft^3/min $(d = 0.075)$.

Mean velocity, $\bar{V}$, at standard conditions is then calculated from $\bar{V} = 45.81\ q$ $(1/A = 45.81$ for the output nozzle, including a slight, but measurable, out-of-roundness). Mean velocity at test conditions must be determined by making a correction for air density at test conditions.

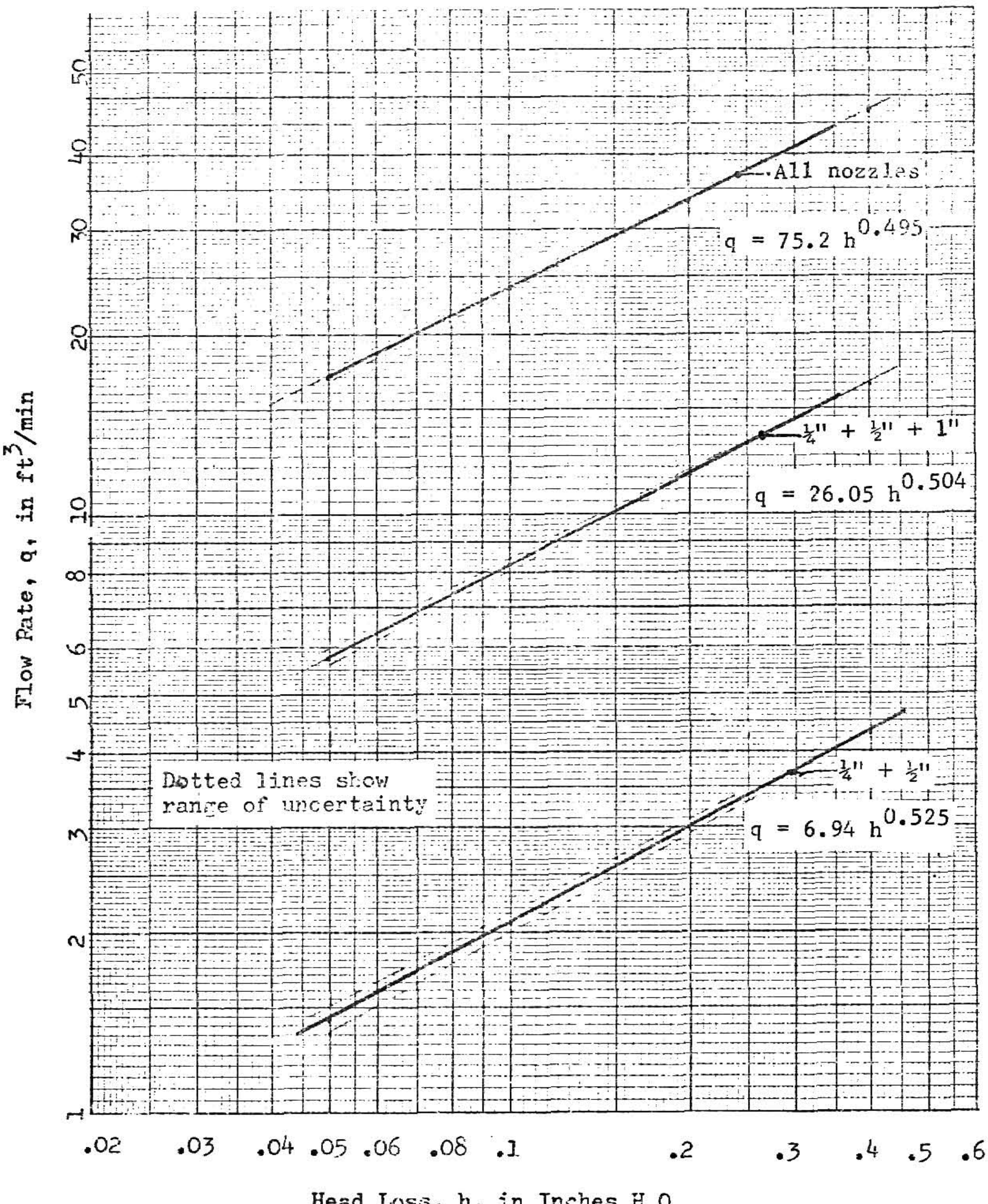
$$\bar{V}_{test} = \bar{V}_{std}\ \left(\frac{.075}{d_{test}}\right) \qquad [10]$$

V_{cl} at test conditions then may be read from Figure 9.



CALIBRATION CURVE, 1/4" NOZZLE

Figure 11



CALIBRATION CURVE, 3 NOZZLE COMBINATIONS

Figure 12

The portion of the nozzle output over which $V = V_{cl}$ varies with flow rate. That is, at the lowest flow rate, the "centerline tube" has a diameter of 0.8". At maximum flow, it is essentially 2.0". Therefore, when a meter is calibrated, the probe must be positioned with care, and the sensing element should be no more than, say, 3/4" long.

At outlet velocities of less than 100 ft/min, the device should be operated in the vertical position so that ambient convection currents do not distort the outlet profile. In any case, calibration should be conducted in a still room where ambient air motion is low.

The next step in the development of the device should be to design and construct a heating section. Velocity profiles should be run at the outlet of this section with and without the evasé in place.

It is recommended that the evasé be used at all flow conditions for the present configuration.

Several tests were made of relative velocities in the plane where the fan section joins the plenum (location C, Figure 1) with and without the evasé in place. As might be expected, the velocities were more uniform and the average velocity was higher with the evasé in place.

CHAPTER VI

CONCLUSIONS

In recognition of the need for calibrating commercial air velocity meters, a device has been designed in compliance with established theory and test codes, constructed by conventional techniques with readily available materials and hardware, and calibrated with standard precision instruments by accepted procedures.

This calibrator will produce known centerline velocities ranging from 15 ft/min to 1950 ft/min. The minimum radius of the "flat" portion of the velocity profile (where $V = V_{cl}$) is 0.4", which occurs at the lowest flow rate. At this flow rate, the estimated uncertainty is 5.4%. As the flow rate increases, the estimated uncertainty decreases to 3%.

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APPENDIX A

MATERIALS AND INSTRUMENTS

List of Materials

1.	Supply fan	Model 2C-781 Dayton centrifugal fan, single inlet, single width, top horizontal discharge, counterclockwise rotation, direct drive 1/45 hp, 120/60/1.
2.	Calibration nozzles	long-radius ASME standard nozzles, aluminum, sizes 1/4", 1/2", 1", 1.6".
3.	Discharge nozzle	long-radius ASME standard nozzle, aluminum, size 2".
4.	Discharge duct	Schedule 40 PVC plastic pipe.
5.	Nozzle plate	1/2" AB grade interior plywood.
6.	Plenum ends and feet	1/2" AB grade interior plywood.
7.	Plenum sides, fan section, evasé	1/4" AB grade interior plywood.
8.	Straightening vanes	1" and 5/8" thinwall aluminum tubing.
9.	Miscellaneous hardware	Jones plug, 12" long piano hinge, #4 nuts and bolts, 1/4" bolts and wing nuts.

Instruments

1.	Barometer	Taylor Instrument Co., temperature-compensated, 29" to 31" Hg scale, least count 0.05" Hg. (No serial number).
2.	Bell prover	Superior Meter Co., with attached thermometer and U-tube manometer. Mechanical Engineering Dept. (KSU) #1366. Bell: 5 ft^3, least count 1/10 ft^3; manometer: 4", least count 1/10".

3. DISA anemometer	DISA, Inc. model 55D80, code 9055F1902, serial 215 with transducer and power supply; scale range 0-100 units, least count 1 unit.
4. Hook gage	F.W. Dwyer model 1420, Mechanical Engineering #4038. Range 0-2", least count 0.001".
5. Microanemometer	Meriam Instrument Co. model A-750, serial 310571, Mechanical Engineering #1342, range 0-20", least count 0.01".
6. Pitot Tube	United Sensor & Control Co., Mechanical Engineering #1717. Least count on positioning bar 0.01".
7. Stop watches	One Racine Select, 0-60 seconds, 0-30 minutes, least count 0.2 seconds. One Meylan Breno, same as above.
8. Thermometer	Fisher Instrument Co. TAG model 581377, range 66-80°F, least count 0.2°F.

APPENDIX B

ABBREVIATIONS AND SYMBOLS

Abbreviations

ASHRAE	American Society of Heating, Refrigerating, and Airconditioning Engineers
ASME	American Society of Mechanical Engineers
OSHA	Occupational Safety and Health Act
PVC	polyvinylchloride

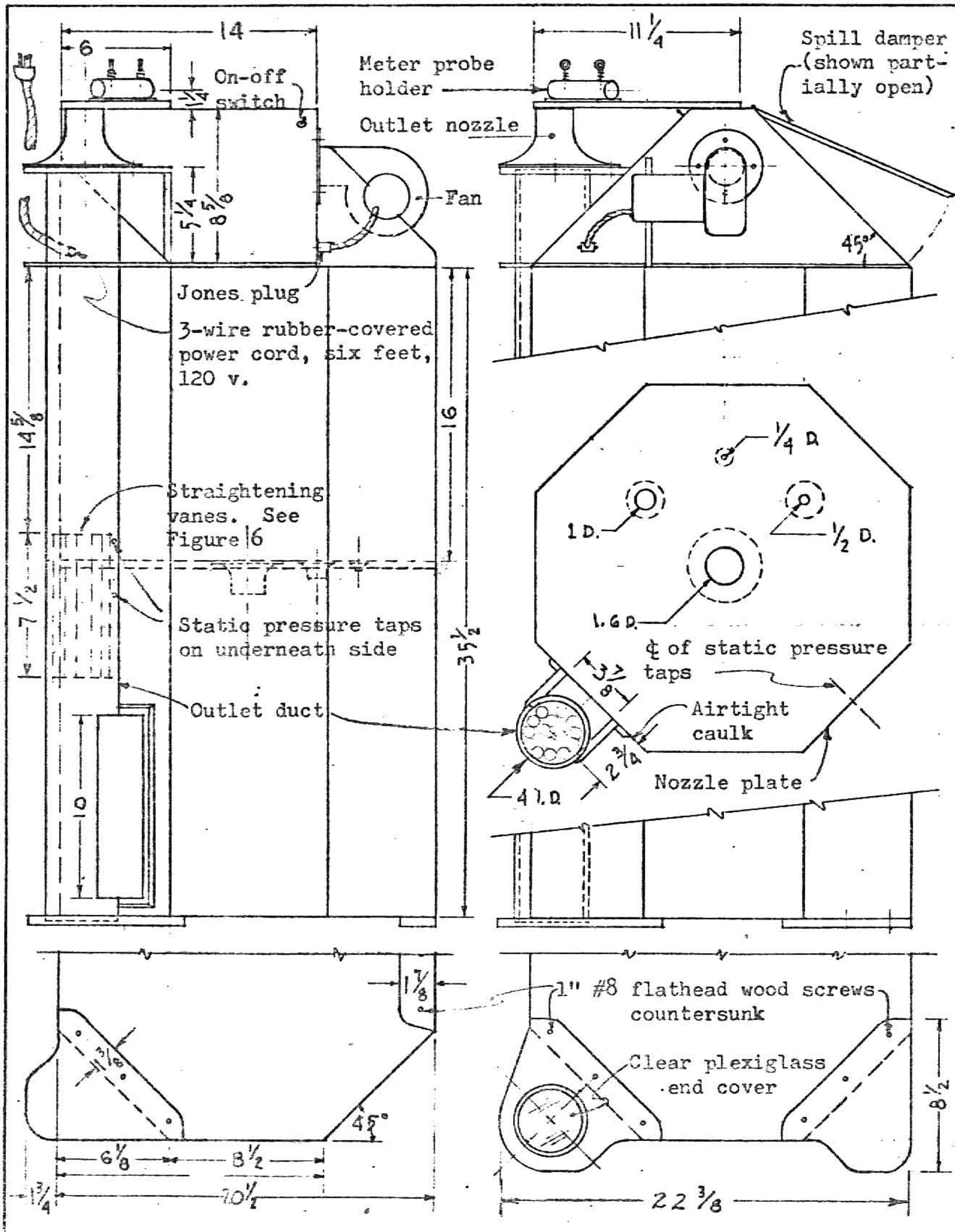
Symbols

a	arbitrary constant	---
A	area	ft^2
b	arbitrary constant	---
b	barometric pressure	"Hg
C	arbitrary constant	---
d	density	lb/ft^3
D	arbitrary constant	---
D	diameter	inches
e_w	vapor pressure of water	"Hg
E_R	uncertainty of a result	%
E_x, E_y	uncertainties of X and Y	%
g	accelaration by gravity	ft/sec^2
G	weight flow rate	lb/min
h	relative humidity	%
h	head	ft-lb/lb
h_L	head loss	ft or in
h_v	velocity head	ft or in
Hg	mercury	---
Hz	hertz	1/sec
K	arbitrary constant	---
lb	force or mass	pound
lbm	mass	pound
m	mass	pound

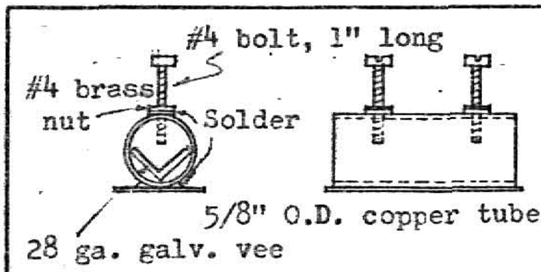
Symbols

n	number (quantity)	---
P	pressure	lb/ft^2
q	volume flow rate	ft^3/min
r	radius	inches
r	correlation coefficient	---
R	gas constant	$lb\text{-}ft/lbm\text{-}^oR$
R	radius	inches
s	relative density	---
SG	specific gravity	---
S_x, S_y	sensitivity factors of X and Y	---
t	temperature	^oF
T	temperature, absolute	^oR
V	volume	ft^3
V	velocity	ft/min
$\underline{V}_{cl}$	centerline velocity	ft/min
$\bar{V}$	mean velocity	ft/min
W	weight	lb
Z	distance above or below datum	ft

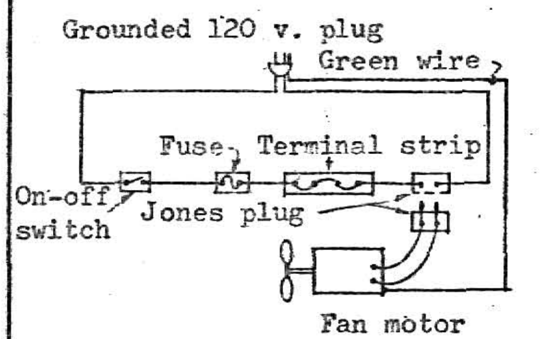
APPENDIX C



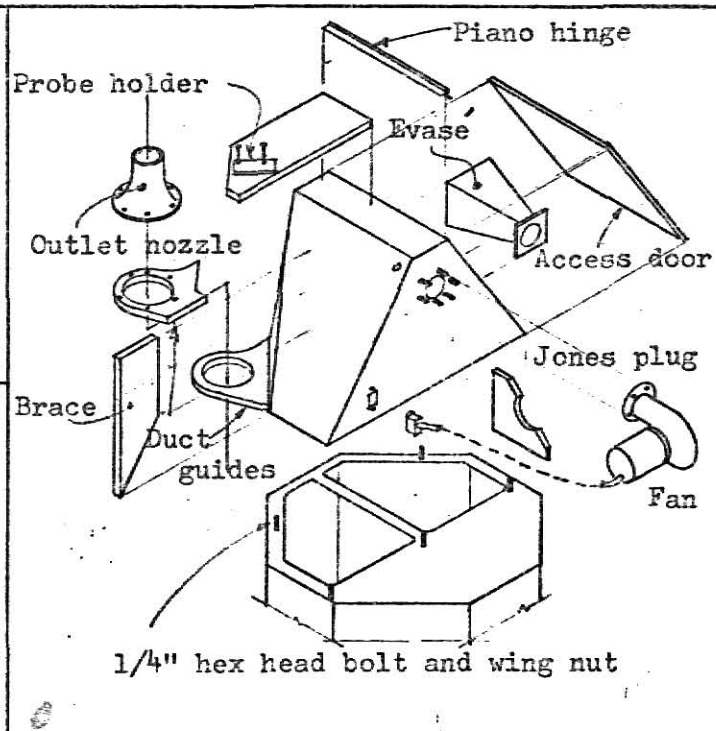
ANEMOMETER CALIBRATOR - PLAN AND SECTION VIEWS	Scale: 1/8" = 1"	R.G.K. 4-27-75	1/2



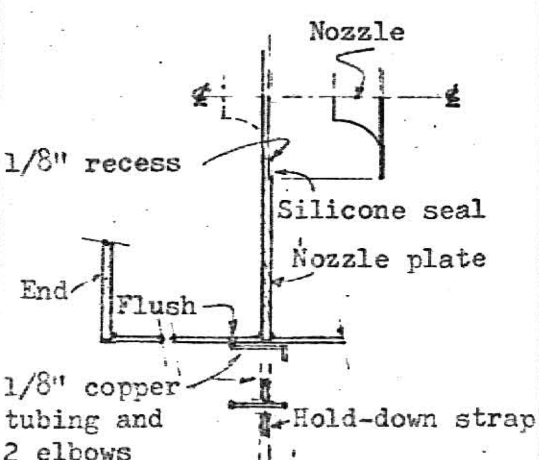
PROBE HOLDER DETAIL



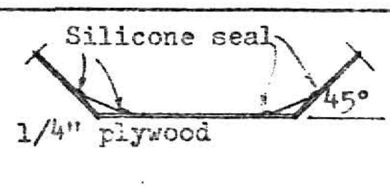
WIRING DIAGRAM



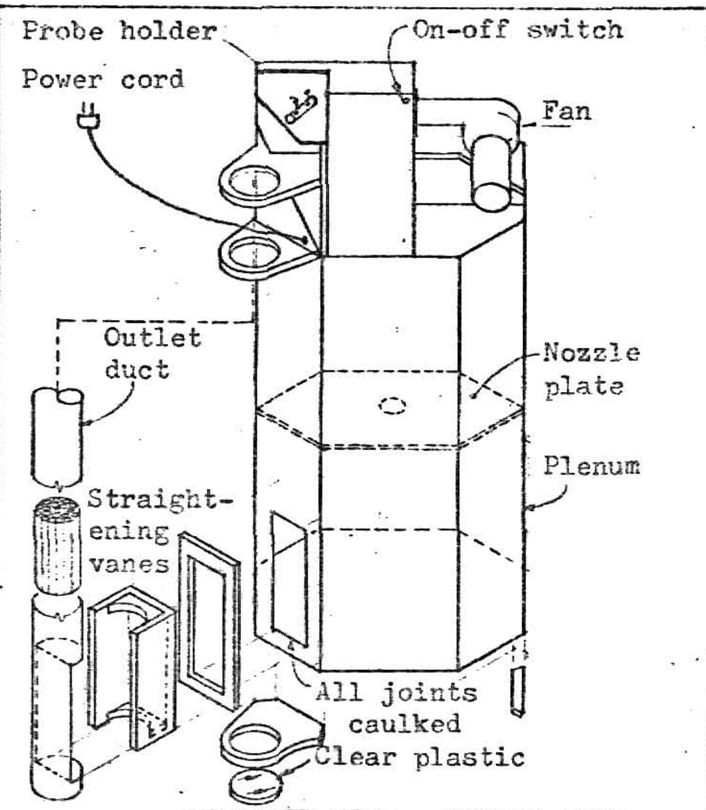
EXPLODED VIEW - FAN SECTION



LONGITUDINAL SECTION THRU PLENUM



TRANSVERSE SECTION THRU PLENUM



EXPLODED VIEW - OUTLET DUCT

ANEMOMETER CALIBRATOR - DETAILS AND SECTION VIEWS	No Scale	7-30-75 EGK	2/2

APPENDIX D

FLUID FLOW AND MEASUREMENT

Introduction

The flow of fluids, either liquids or gases, can be classified several ways. The fluid can be either newtonian, non-newtonian, or ideal (hypothetical case). It may be compressible or incompressible (although all fluids are compressible, liquids are considered incompressible in most practical cases and frequently gases are treated as incompressible). The flow may be steady or unsteady with respect to time; it may be uniform or non-uniform from point to point. It might be rotational or not; supercritical or subcritical. The flow regime may be laminar, turbulent, or transitional [6].

For this work, the fluid (air) is considered newtonian and incompressible, because the changes in density are less than 5%. The flow is subcritical (below sonic velocity), steady at any point with respect to time, but non-uniform from one point to another. Any rotational flow (except in the fan) is unintentional and minor, hence disregarded. The flow regime depends on the Reynolds number, which is a function of the velocity, some characteristic dimension, and the kinematic viscosity.

The Continuity Equation

Steady flow means the velocity profile of the fluid at a given point does not vary with time, although the velocity profile may vary from point to point (non-uniform). The conservation of mass, or continuity, equation at points 1 and 2 in a system may be written as:

$$G = d_1 A_1 V_1 = d_2 A_2 V_2 = \text{constant} \qquad [11]$$

where G is weight flow rate in lb/min

d is fluid density in lb/ft^3

A is area in ft^2

V is mean velocity in ft/min

Since d is constant in incompressible flow, the equation may be written as:

$$\frac{G}{d} = A_1 V_1 = A_2 V_2; \qquad [12]$$

$$q = AV \qquad [13]$$

where q is G/d or volume flow rate in ft^3/min

Energy Balance

The steady state version of the Bernoulli equation is usually written as:

$$Q + Z_1 + \frac{V_1^2}{2g} + \frac{P_1}{d} = Z_2 + \frac{V_2^2}{2g} + \frac{P_2}{d} + W \qquad [14]$$

where Q is heat added to the process

W is work done by the process

Z is distance above an arbitrary datum

Each term has units of foot-pounds per pound, which is usually abbreviated to feet.

In a system where no heat or work is added or removed, and where frictional effects are disregarded, the equation may be written as:

$$(Z_1 - Z_2) + (V_1^2 - V_2^2)/2g = (P_2 - P_1)/d \qquad [15]$$

Usually in gaseous fluid systems, the first term is very small compared to the other two and can be neglected. That is, changes in elevation are usually slight compared to other changes. Equations [14] and [15] also contain another simplification involving the relationship between the $V^2/2g$ term and the actual kinetic energy per pound of fluid. The kinetic energy is greater than $V^2/2g$, but for turbulent flow, the difference is slight.

For steady incompressible flow where friction is present, the equation may be written:

$$\frac{V_1^2}{2g} + \frac{P_1}{d} = \frac{V_2^2}{2g} + \frac{P_2}{d} + h_L \qquad [16]$$

where h_L is loss in head because of friction

In a conduit of constant cross section and no change in air quantity, the velocity terms are equal and can be cancelled out, leaving an expression for the loss in head as:

$$h_L = \frac{P_1}{d} - \frac{P_2}{d} \qquad [17]$$

Thus, at a constant flow rate in a constant area conduit, the effect of friction is to decrease the P/d term. This term is referred to as the static head.

The relationship between velocity, V, and velocity head, h_v, is developed from $h_v = V^2/2g$. Velocity head is determined by measuring total head and static head of a flowing stream with a pitot-static tube connected to the two legs of a U-tube manometer. If the manometer contains a different fluid than the fluid being measured, the manometer reading must be converted to feet of head of the flowing fluid.

Thus, if air is being measured by using a manometer reading inches of water, and the desired units of velocity are feet/minute, the equation becomes:

$$V = (2gh_v)^{1/2} \qquad [18]$$

Inserting the appropriate values yields

$$V = 1096.7\ (h_v/d)^{1/2} \qquad [19]$$

where V = velocity in ft/min

h_v = velocity head equivalent, in inches of water

d = density of air flowing, in lb/ft^3

For standard air (d = 0.075), the equation is:

$$V = 4005\ (h_v)^{1/2} \qquad [20]$$

Measurement Methods

The pitot-static probe method of determining velocity head uses the above relationships to calculate velocity. There are well-established procedures for determining mean velocity, $\bar{V}$, from velocity head readings taken at various points in the stream flow. If the flow area is known, flow rate can be calculated from mean velocity.

If access into the flowing stream is feasible, the pitot-static probe technique is a very satisfactory way to determine flow rate. However, it is not always possible to insert the probe in the stream.

Many other techniques and instruments have been developed to measure flow. Table 7 is a copy of a list of meters which appears in Fluid Meters [3], showing the classes and types of flow meter. One major class of meter uses differential pressure to determine flow, where some functional relationship can be established between static pressure differential and velocity through a known area.

Nozzles are included in the category of differential pressure rate meters, and performance coefficients have been developed for nozzles built to standard specifications. However, recent work has indicated some slight disparities between published coefficients and measured performance [12].

The calibrator described in this thesis uses standard ASME long-radius nozzles, but the standard published flow coefficients were not used. The nozzles were calibrated specifically for the flow rates and configuration used in the calibrator.

TABLE 7

Classification of Fluid Meters and Methods of Fluid Measurement

Division	Class		Type
1. Quantity Meters	Weighing		Weighers
			Tilting Traps
	Volumetric	Liquids	Tank
			Reciprocating Piston
			Rotating Piston or Ring Piston
			Nutating Disk
			Sliding and Rotating Vanes
			Gear and Lobed Impeller (Rotary)
		Gases	Bellows
			Liquid Sealed Drum
2. Rate Meters	Differential Pressure		Venturi
			Flow Nozzle
			Nozzle-Venturi
			Critical Flow Nozzle
			Thick-Plate Rounded-Edge Orifice
			Thin-Plate Square-Edged Orifice
			Concentric
			Eccentric
			Segmental
			Gate or Variable Area
			Centrifugal
			Elbow or Long-Radius Bend
			Turbine Scroll Case
			Guide Vane Speed Ring
			Pitot Tube (Impact Tube)
			Pitot-Static Tube
			Pitot-Venturi
			Linear Resistance or Frictional
			Pipe Section
			Capillary Tube
			Porous Plug
	Area (Geometric)		Gate
			Cone and Float
			Slotted Cylinder and Piston
	Velocity		
	Open or Closed Channel		Cup (Anemometer)
			Propeller
	Closed Channel		Turbine
	Head-Area		Weirs
			Flumes
	Force		
	Open or Closed Channel		Hydrometric Pendulum
			Vane
	Closed Channel		Transverse-Momentum Flow Meters
	Thermal		
	Open or Closed Channel		Hot Wire
			Total Heating
	Other Meters and Methods		
	Open or Closed Channel		Electromagnetic
			Tracers
			Mixtures
			Floats and Screens
	Closed Channel		Pressure-Time
			Sound Velocity
			Light Velocity

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VITA

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Master of Science

Thesis: THE DESIGN, CONSTRUCTION AND CALIBRATION OF A LOW VELOCITY ANEMOMETER CALIBRATOR

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THE DESIGN, CONSTRUCTION AND CALIBRATION OF A LOW VELOCITY ANEMOMETER CALIBRATOR

by

RODNEY GENE KEIF

B. S., Kansas State University, 1949

AN ABSTRACT OF A MASTER'S THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1975

ABSTRACT

The need to measure airflow accurately in occupied spaces and closed conduits has increased rapidly in recent years. Stringent environment control, from laminar flow clean rooms to high velocity industrial exhaust systems to blast tunnels for quick freezing, has been augmented by recent federal legislation which requires that air volumes and velocities meet specific criteria.

There has always been a problem of calibrating airflow meters quickly, inexpensively, and accurately, especially at low velocities of, say, less than fifty feet per minute. This thesis describes the design, construction, and calibration of a device which produces known outlet velocities varying from 15 fpm to 1950 fpm with a calculated uncertainty of 4.3% at the minimum flow rate. The device can be used as a calibrator for commercial air velocity meters.

It consists of an array of four ASME long-radius nozzles mounted in a plenum, a supply fan, and an outlet duct which terminates in a 2" ASME long-radius nozzle to provide as flat a discharge velocity profile as possible. Static pressure drop across the nozzles in the plenum is measured to the nearest 0.001" H_2O with a hook gage, and flow rate and outlet velocity may be determined by consulting the calibration curves provided.

The calibrator is portable and may be used in the horizontal or vertical position.