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Laboratory Methods of Testing Fans for Aerodynamic Performance Rating

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An American National Standard

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AIR MOVEMENT AND CONTROL ASSOCIATION INTERNATIONAL, INC.

The International Authority on Air System Components

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LABORATORY METHOD OF TESTING FANS FOR AERODYNAMIC PERFORMANCE RATING

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JOINT AMCA 210/ASHRAE 51 PROJECT COMMITTEE

Robert Jorgensen, Chairman	AMCA - ASHRAE
Steve Adamski	AMCA - ASHRAE
Hoy R. Bohanon	AMCA - ASHRAE
John Cermak	AMCA - ASHRAE
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Paul R. Saxon	AMCA - ASHRAE
James W. Schwier †	AMCA - ASHRAE
Mark Stevens	AMCA
J. Thomas Sobieski	ASHRAE
William B. Swim	ASHRAE

† deceased

SPECIAL NOTE

This National Voluntary Consensus Standard was developed under the joint auspices of the Air Movement and Control Association International, Inc. (AMCA) and the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE). Consensus is defined as "substantial agreement reached by concerned interests according to the judgement of a duly appointed authority, after a concerted attempt at resolving objections. Consensus implies much more than the concept of a simple majority but not necessarily unanimity." This definition is according to the American National Standards Institute (ANSI) of which both AMCA and ASHRAE are members.

This Foreword is not a part of ANSI/AMCA Standard 210 or ANSI/ASHRAE Standard 51 but is included for information purposes only. See also Appendix I for the History and Authority.

FOREWORD

This standard provides rules for testing fans, under laboratory conditions, to provide rating information. It was prepared by a joint committee consisting of the Air Movement and Control Association International, Inc. (AMCA) 210 Review Committee and the American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE) Standard 51-85R Committee.

The joint committee debated whether the International Standard for laboratory testing of industrial fans, ISO 5801 *Industrial fans: Performance testing using standardised airways*, should be adopted in lieu of preparing a new edition of this standard. The decision to proceed with a ninth edition was based on the conclusion that ISO 5801 allowed the use of measurements that did not meet the uncertainties requirements of this standard. However, certain features of ISO 5801 have been included, most of which were anticipated in the 1985 edition.

The principal changes compared to ANSI/AMCA 210-85//ANSI/ASHRAE 51-85 *Laboratory Methods of Testing Fans for Rating* are:

1) Incorporation of SI units in the text. SI units are primary, I-P units are secondary.
2) Addition of SI equations.
3) Numbering of equations for easier reference.
4) Deletion of tabular and graphical data as unnecessary, since equations are definitive and universal use of computers is anticipated.
5) Addition of Appendix F, giving an example of the iterative solution of Re and C.
6) Addition of Appendix I, giving the history of fan test codes in North America.

Suggestions for improvement of this standard will be welcome. They should be sent to either the Air Movement and Control Association International, Inc., 30 West University Drive, Arlington Heights, Illinois 60004-1893 U.S.A. or the American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329 U.S.A.

DISCLAIMER

CONTENTS

Laboratory Methods of Testing Fans For Aerodynamic Performance Rating

1. Purpose

This standard establishes uniform methods for laboratory testing of fans and other air moving devices to determine aerodynamic performance for rating or guarantee purposes in terms of airflow rate, pressure, power, air density, speed of rotation, and efficiency.

It is not the purpose of this standard to specify the testing procedures to be used for design, production, or field testing.

2. Scope

2.1 This standard may be used as the basis for testing fans, blowers, exhausters, compressors, or other air moving devices when air is used as the test gas.

2.2 The scope of this standard does not cover:
(a) circulating fans such as ceiling fans, desk fans and jet fans.
(b) compressors with interstage cooling.
(c) positive displacement machines.
(d) testing procedures to be used for design, production, or field testing.

2.3 The parties to a test for guarantee purposes may agree on exceptions to this standard in writing prior to the test. However, only tests which do not violate any mandatory requirements of this standard shall be designated as tests conducted in accordance with this standard.

3. Definitions

3.1 Fans

3.1.1 Fan: A device for moving air which utilizes a power driven rotating impeller. A fan shall have at least one inlet opening and at least one outlet opening. The openings may or may not have elements for connection to ductwork.

3.1.2 Boundaries.

3.1.2.1 Fan Inlet and Outlet Boundaries. Fan inlet and outlet boundaries are defined as the interfaces between the fan and the remainder of the system, and are at a plane perpendicular to the air stream where it enters or leaves the fan. Various appurtenances, such as inlet boxes, inlet vanes, inlet cones, silencers, screens, rain hoods, dampers, discharge cones, evasé, etc., may be included as a part of the fan between the inlet and outlet boundaries.

3.1.2.2 Fan Input Power Boundary. The interface between the fan and its driver. Drive or coupling losses may be included as a part of the input power.

3.1.3 Fan Outlet Area. Fan outlet area is the gross inside area measured in the plane(s) of the outlet opening(s). For roof ventilators and unhoused fans, the area shall be considered the gross impeller outlet area for centrifugal types or the gross casing area at the impeller for axial types.

3.1.4 Fan Inlet Area. Fan inlet area is the gross inside area measured in the plane(s) of the inlet connection(s). For converging inlets without connection elements, the inlet area shall be considered to be that where a plane, perpendicular to the airstream, first meets the bell mouth or cone.

3.2 Psychrometrics

3.2.1 Dry-Bulb Temperature. Dry-bulb temperature is the air temperature measured by a dry temperature sensor.

3.2.2 Wet-Bulb Temperature. Wet-bulb temperature is the temperature measured by a temperature sensor covered by a water-moistened wick and exposed to air in motion. When properly measured, it is a close approximation of the temperature of adiabatic saturation.

3.2.3 Wet-Bulb Depression. Wet-bulb depression is the difference between the dry-bulb and wet-bulb temperatures at the same location.

3.2.4 Stagnation (Total) Temperature. Stagnation (total) temperature is the temperature which exists by virtue of the internal and kinetic energy of the air. If the air is at rest, the stagnation (total) temperature will equal the static temperature.

3.2.5 Static Temperature. Static temperature is the temperature which exists by virtue of the internal energy of the air only. If a portion of the internal energy is converted into kinetic energy, the static temperature will be decreased accordingly.

3.2.6 Air Density. Air density is the mass per unit volume of the air.

Note: References which are enclosed in { } are normative for this standard, while those enclosed in [ ] are to be considered informative.

3.2.7 Standard Air. Standard air is air with a density of 1.2 kg/m^3 (0.075 lbm/ft^3)).

3.2.8 Standard Air Properties. Standard air has a ratio of specific heats of 1.4 and a viscosity of 1.8185E-03 $Pa{\cdot}s$ (1.222E-05 $lbm/ft{\cdot}s$). Air at 20 °C (68 °F) temperature, 50% relative humidity, and 101.325 kPa (14.696 psi, 29.92 $in.\ Hg$) barometric pressure has these properties, approximately.

3.3 Pressure

3.3.1 Pressure. Pressure is force per unit area. This corresponds to energy per unit volume of fluid.

3.3.2 Absolute Pressure. Absolute pressure is the value of a pressure when the datum pressure is absolute zero. It is always positive.

3.3.3 Barometric Pressure. Barometric pressure is the absolute pressure exerted by the atmosphere.

3.3.4 Gauge Pressure. Gauge pressure is the value of a pressure when the datum pressure is the barometric pressure at the point of measurement. It may be negative or positive.

3.3.5 Velocity Pressure. Velocity pressure is that portion of the air pressure which exists by virtue of the rate of motion only. It is always positive.

3.3.6 Static Pressure. Static pressure is that portion of the air pressure which exists by virtue of the degree of compression only. If expressed as gauge pressure, it may be negative or positive.

3.3.7 Total Pressure. Total pressure is the air pressure which exists by virtue of the degree of compression and the rate of motion. It is the algebraic sum of the velocity pressure and the static pressure at a point. Thus, if the air is at rest, the total pressure will equal the static pressure.

3.3.8 Pressure Loss. Pressure loss is the decrease in total pressure due to friction and turbulence.

3.4 Fan Performance Variables

3.4.1 Fan Air Density. Fan air density is the density of the air corresponding to the total pressure and the stagnation temperature of the air at the fan inlet [1].

3.4.2 Fan Airflow Rate. Fan airflow rate is the volumetric airflow rate at fan air density.

3.4.3 Fan Total Pressure. Fan total pressure is the difference between the total pressure at the fan outlet and the total pressure at the fan inlet.

3.4.4 Fan Velocity Pressure. Fan velocity pressure is the pressure corresponding to the average air velocity at the fan outlet.

3.4.5 Fan Static Pressure. Fan static pressure is the difference between the fan total pressure and the fan velocity pressure. Therefore, fan static pressure is the difference between the static pressure at the fan outlet and the total pressure at the fan inlet.

3.4.6 Fan Speed. Fan speed is the rotational speed of the impeller. If a fan has more than one impeller, fan speeds are the rotational speeds of each impeller.

3.4.7 Compressibility Coefficient. Compressibility coefficient is a thermodynamic factor which must be applied to determine fan total efficiency from fan airflow rate, fan total pressure, and fan power input. This coefficient is derived in Appendix C. It may be considered to be the ratio of the mean airflow rate through the fan to the airflow rate at fan air density. It is also the ratio of the fan total pressure that would be developed with an incompressible fluid to the fan total pressure that is developed with a compressible fluid.

3.4.8 Fan Power Output. Fan power output is the useful power delivered to the air. This is proportional to the product of fan airflow rate and fan total pressure and compressibility coefficient.

3.4.9 Fan Power Input. Fan power input is the power required to drive the fan and any elements in the drive train which are considered a part of the fan.

3.4.10 Fan Total Efficiency. Fan total efficiency is the ratio of the fan power output to the fan power input.

3.4.11 Fan Static Efficiency. Fan static efficiency is the fan total efficiency multiplied by the ratio of fan static pressure to fan total pressure.

3.5 Miscellaneous

3.5.1 Point of Operation. Point of operation is the relative position on the fan characteristic curve corresponding to a particular airflow rate. It is controlled during a test by adjusting the position of the throttling device, by changing flow nozzles or auxiliary fan characteristics, or by any combination of these.

3.5.2 Free Delivery. Free delivery is the point of operation where the fan static pressure is zero.

3.5.3 Shall and Should. The word "shall" is to be understood as mandatory, the word "should" as advisory.

3.5.4 Shut Off. Shut off is the point of operation where the fan airflow rate is zero.

3.5.5 Determination. A determination is a complete set of measurements for a particular point of operation of a fan. The measurements must be sufficient to determine all fan performance variables defined in 3.4 .

3.5.6 Test. A test is a series of determinations for various points of operation of a fan.

3.5.7 Energy Factor. Energy factor is the ratio of the total kinetic energy of the flow to the kinetic energy corresponding to the average velocity.

3.5.8 Demonstrated Accuracy. Demonstrated accuracy is defined for the purpose of this standard as the accuracy of an instrument or the method established by testing of the instrument or the method against a primary or calibrated instrument or method in accordance with the requirements stated in this standard. (See 5.4.1.1, 5.4.2.1, 5.4.3.1, 5.5.3, 5.6.1.1, and 5.6.2.1)

4. Symbols and Subscripts

4.1 Symbols and Subscripted Symbols

SYMBOL	DESCRIPTION	SI UNIT		I-P UNIT
A	Area of Cross-Section	m^2		ft^2
C	Nozzle Discharge Coefficient		dimensionless	
D	Diameter and Equivalent Diameter	m		ft
D_h	Hydraulic Diameter	m		ft
e	Base of Natural Logarithm (2.718...)		dimensionless	
E	Energy Factor		dimensionless	
F	Beam Load	N		lbf
f	Coefficient of Friction		dimensionless	
H	Fan Power Input	W		hp
H_o	Fan Power Output	W		hp
K_p	Compressibility Coefficient		dimensionless	
L	Nozzle Throat Dimension	m		ft
L_e	Equivalent Length of Straightener	m		ft
$L_{x,x}$	Length of Duct Between Planes x and x′	m		ft
l	Length of Moment Arm	m		$in.$
ln	Natural Logarithm	---		---
M	Chamber Dimension	m		ft
N	Speed of Rotation	rpm		rpm
n	Number of Readings		dimensionless	
P_s	Fan Static Pressure	Pa		$in.\ wg$
P_{sx}	Static Pressure at Plane x	Pa		$in.\ wg$
P_t	Fan Total Pressure	Pa		$in.\ wg$
P_{tx}	Total Pressure at Plane x	Pa		$in.\ wg$
P_v	Fan Velocity Pressure	Pa		$in.\ wg$
P_{vx}	Velocity Pressure at Plane x	Pa		$in.\ wg$
p_b	Corrected Barometric Pressure	Pa		$in.\ Hg$
p_e	Saturated Vapor Pressure at t_w	Pa		$in.\ Hg$
p_p	Partial Vapor Pressure	Pa		$in.\ Hg$
Q	Fan Airflow Rate	m^3/s		cfm
Q_x	Airflow Rate at Plane x	m^3/s		cfm
R	Gas Constant	$J/kg{\cdot}K$		$ft{\cdot}lbf/lbm{\cdot}{}^\circ R$
Re	Reynolds Number		dimensionless	
T	Torque	N•m		lbf•in.
t_d	Dry-Bulb Temperature	$^\circ C$		$^\circ F$
t_s	Stagnation (total) Temperature	$^\circ C$		$^\circ F$
t_w	Wet-Bulb Temperature	$^\circ C$		$^\circ F$
V	Velocity	m/s		fpm
W	Power Input to Motor	W		W
x	Function Used to Determine K_p		dimensionless	
Y	Nozzle Expansion Factor		dimensionless	
y	Thickness of Straightener Element	mm		$in.$
z	Function Used to Determine K_p		dimensionless	
α	Static Pressure Ratio for Nozzles		dimensionless	
β	Diameter Ratio for Nozzles		dimensionless	
γ	Ratio of Specific Heats		dimensionless	
ΔP	Pressure Differential	Pa		$in.\ wg$
η	Motor Efficiency		per unit	
η_s	Fan Static Efficiency		per unit	
η_t	Fan Total Efficiency		per unit	
μ	Dynamic Air Viscosity	$Pa{\cdot}s$		$lbm/ft{\cdot}s$
ρ	Fan Air Density	kg/m^3		lbm/ft^3
ρ_x	Air Density at Plane x	kg/m^3		lbm/ft^3
Σ	Summation Sign	---		---

4.2 Additional Subscripts

SUBSCRIPT	DESCRIPTION
c	Converted value
r	Reading
x	Plane 0, 1, 2....as appropriate
0	Plane 0 (general test area)
1	Plane 1 (fan inlet)
2	Plane 2 (fan outlet)
3	Plane 3 (Pitot traverse station)
4	Plane 4 (duct piezometer station)
5	Plane 5 (nozzle inlet station in chamber)
6	Plane 6 (nozzle discharge station)
7	Plane 7 (outlet chamber measurement station)
8	Plane 8 (inlet chamber measurement station)

5. Instruments and Methods of Measurement

5.1 Accuracy [2] The specifications for instruments and methods of measurement which follow include both accuracy requirements and specific examples of equipment that are capable of meeting those requirements. Equipment other than the examples cited may be used provided the accuracy requirements are met or exceeded. As noted in Appendix E, the use of the same instruments over the entire range of fan performance at constant speed will result in fairly large relative uncertainties near shutoff and near free delivery. This is generally acceptable because fans are not normally rated near these points. However, if this is not acceptable, different instruments should be selected for different points of operation as appropriate. See example in 6.3.4. Laboratory setups may be designed to facilitate such choices easily.

5.1.1 Instrument Accuracy. The specifications regarding accuracy correspond to two standard deviations based on an assumed normal distribution. This is frequently how instrument suppliers identify accuracy, but that should be verified. The calibration procedures, which are specified in this standard , shall be employed to minimize errors. In any calibration process, the large systematic error of the instrument is exchanged for the smaller combination of the systematic error of the standard instrument and the random error of the comparison. Instruments shall be set up, calibrated, and read by qualified personnel trained to minimize errors.

5.1.2 Measurement Uncertainty. It is axiomatic that every test measurement contains some error and that the true value cannot be known because the magnitude of the error cannot be determined exactly. However, it is possible to perform an uncertainties analysis to identify a range of values within which the true value probably lies. A probability of 95% has been chosen as acceptable for this standard.

The standard deviation of random errors can be determined by statistical analysis of repeated measurements. No statistical means are available to evaluate systematic errors, so these must be estimated. The estimated upper limit of a systematic error is called the systematic uncertainty and, if properly estimated, it will contain the true value 99% of the time. The two standard deviation limit of a random error has been selected as the random uncertainty. Two standard deviations yield 95% probability for random errors.

5.1.3 Uncertainty of a Result. The results of a fan test are the various fan performance variables listed in Section 3.4. Each result is based on one or more measurements. The uncertainty in any result can be determined from the uncertainties in the measurement. It is best to determine the systematic uncertainty of the result and then the random uncertainty of the result before combining them into the total uncertainty of the result. This may provide clues on how to reduce the total uncertainty. When the systematic uncertainty is combined in quadrature with the random uncertainty, the total uncertainty will give 95% coverage. In most test situations, it is wise to perform a pre-test uncertainties analysis to identify potential problems. A pre-test uncertainties analysis is not required for each test covered by this standard because it is recognized that most laboratory tests for rating are conducted in facilities where similar tests are repeatedly run. Nevertheless, a pre-test analysis is recommended as is a post-test analysis. The simplest form of analysis is a verification that all accuracy and calibration specifications have been met. The most elaborate analysis would consider all the elemental sources of error including those due to calibration, data acquisition, data reduction, calculation assumptions, environmental effects, and operational steadiness.

The sample analysis given in Appendix E calculates the uncertainty in each of the fan performance variables and, in addition, combines certain ones into a characteristic uncertainty and certain others into an efficiency uncertainty.

5.2 Pressure. The total pressure at a point shall be measured on an indicator, such as a manometer, with one leg open to atmosphere and the other leg connected to a total pressure sensor, such as a total pressure tube or the impact tap of a Pitot-static tube. The static pressure at a point shall be measured on an indicator, such as a manometer, with one leg open to the atmosphere and the other leg connected to a static pressure sensor, such as a static pressure tap or the static tap of a Pitot-static tube. The velocity pressure at a point shall be measured on an indicator, such as a manometer, with one leg connected to a total pressure sensor, such as the impact tap of a

Pitot-static tube, and the other leg connected to a static pressure sensor, such as the static tap of the same Pitot-static tube. The differential pressure between two points shall be measured on an indicator, such as a manometer, with one leg connected to the upstream sensor, such as a static pressure tap, and the other leg connected to the downstream sensor, such as a static pressure tap.

5.2.1 Manometers and Other Pressure Indicating Instruments. Pressure shall be measured on manometers of the liquid column type using inclined or vertical legs or other instruments which provide a maximum uncertainty of 1% of the maximum observed test reading during the test or 1 *Pa* (0.005 *in. wg*) whichever is larger. See Note 1.

5.2.1.1 Calibration. Each pressure indicating instrument shall be calibrated at both ends of the scale and at least nine equally spaced intermediate points in accordance with the following:

(1) When the pressure to be indicated falls in the range of 0 to 2.5 *kPa* (0 to 10 *in. wg*), calibration shall be against a water-filled hook gauge of the micrometer type or a precision micromanometer.

(2) When the pressure to be indicated is above 2.5 *kPa* (10 *in. wg*), calibration shall be against a water-filled hook gauge of the micrometer type, a precision micromanometer, or a water-filled U-tube.

5.2.1.2 Averaging. Since the airflow and the pressures produced by a fan are never strictly steady, the pressure indicated on any instrument will fluctuate with time. In order to obtain a representative reading, either the instrument must be damped or the readings must be averaged in a suitable manner. Averaging can sometimes be accomplished mentally, particularly if the fluctuations are small and regular. Multi-point or continuous record averaging can be accomplished with instruments and analyzers designed for this purpose.

5.2.1.3 Corrections. Manometer readings should be corrected for any difference in specific weight of gauge fluid from standard, any difference in gas column balancing effect from standard, or any change in length of the graduated scale due to temperature. However, corrections may be omitted for temperatures between 14°*C* and 26°*C* (58°*F* and 78°*F*), latitudes between 30° and 60°, and elevations up to 1500 *m* (5000 *ft*).

5.2.2 Pitot-Static Tubes. [3] [4]. The total pressure or the static pressure at a point may be sensed with a Pitot-static tube of the proportions shown in Figure 1.Either or both of these pressure signals can then be transmitted to a manometer or other indicator. If both pressure signals are transmitted to the same indicator, the differential is considered velocity pressure at the point of the impact opening.

5.2.2.1 Calibration. Pitot-static tubes having the proportions shown in Figure 1 are considered primary instruments and need not be calibrated provided they are maintained in the specified condition.

5.2.2.2 Size. The Pitot-static tube shall be of sufficient size and strength to withstand the pressure forces exerted upon it. The outside diameter of the tube shall not exceed 1/30 of the test duct diameter except that when the length of the supporting stem exceeds 24 tube diameters, the stem may be progressively increased beyond this distance. The minimum practical tube diameter is 2.5 *mm* (0.10 *in.*).

5.2.2.3 Support. Rigid support shall be provided to hold the Pitot-static tube axis parallel to the axis of the duct within 1 degree and at the head locations specified in Figure 3 within 1 *mm* (0.05 *in.*) or 25% of the duct diameter, whichever is larger.

5.2.3 Static Pressure Taps. The static pressure at a point may be sensed with a pressure tap of the proportions shown in Figure 2A. The pressure signal can then be transmitted to an indicator.

5.2.3.1 Calibration. Pressure taps having the proportions shown in Figure 2A are considered primary instruments and need not be calibrated provided they are maintained in the specified condition. Every precaution should be taken to ensure that the air velocity does not influence the pressure measurement.

5.2.3.2 Averaging. An individual pressure tap is sensitive only to the pressure in the immediate vicinity of the hole. In order to obtain an average, at least four taps in accordance with Figure 2A shall be manifolded into a piezometer ring. The manifold shall have an inside area at least four times that of each tap. An example in shown in Appendix G.

5.2.3.3 Piezometer Rings. Piezometer rings are specified for upstream and downstream nozzle taps and for outlet duct or chamber measurements unless a Pitot traverse is specified. Measuring planes shall be located as shown in the figure for the appropriate setup.

5.2.4 Total Pressure Tubes. The total pressure in an inlet chamber may be sensed with a stationary tube of the proportions shown in Figure 2B. The pressure signal can then be transmitted to an indicator. The tube shall face directly into the air flow and the open end shall be smooth and free from burrs.

5.2.4.1 Calibration. Total pressure tubes are considered primary instruments and need not be calibrated if they are maintained in a condition conforming to this standard.

5.2.4.2 Averaging. The total pressure tube is sensitive only to the pressure in the immediate vicinity of the open end. However, since the velocity in an inlet chamber can be considered uniform due to the settling means which are employed, a single measurement is representative of the average chamber pressure.

5.2.4.3 Location. Total pressure tubes are specified for inlet chambers. Location shall be as shown in the figure for the appropriate setup.

5.2.5 Other Pressure Measuring Systems. Pressure measuring systems consisting of indicators and sensors other than manometers and Pitot-static tubes, static pressure taps, or total pressure tubes may be used if the combined uncertainty of the system including any transducers does not exceed the combined uncertainty for an appropriate combination of manometers and Pitot-static tubes, static pressure taps, or total pressure tubes. For systems used to determine fan pressure the contribution to combined uncertainty in the pressure measurement shall not exceed that corresponding to 1% of the maximum observed static or total pressure reading during a test (indicator accuracy), plus 1% of the actual reading (averaging accuracy). For systems used to determine fan airflow rate, the combined uncertainty shall not exceed that corresponding to 1% of the maximum observed velocity pressure or pressure differential reading during a test (indicator accuracy) plus 1% of the actual reading (averaging accuracy). See Note 1.

5.3 Airflow Rate. Airflow rate shall be calculated either from measurements of velocity pressure obtained by Pitot traverse or from measurements of pressure differential across a flow nozzle.

5.3.1 Pitot Traverse. Airflow rate may be calculated from the velocity pressures obtained by traverses of a duct with a Pitot-static tube for any point of operation from free delivery to shut off provided the average velocity corresponding to the airflow rate at free delivery at the test speed is at least 12 *m/s* (2400 *fpm*) [5]. See Note 1.

5.3.1.1 Stations. The number and locations of the measuring stations on each diameter and the number of diameters shall be as specified in Figure 3.

5.3.1.2 Averaging. The stations shown in Figure 3 are located on each diameter according to the log-linear rule [6]. The arithmetic mean of the individual velocity measurements made at these stations will be the mean velocity through the measuring section for a wide variety of profiles [7].

5.3.2 Nozzles. Airflow rate may be calculated from the pressure differential measured across a flow nozzle or bank of nozzles for any point of operation from free delivery to shut off provided the average velocity at the nozzle discharge corresponding to the airflow rate at free delivery at the test speed is at least 14 *m/s* (2800 *fpm*) [5].

5.3.2.1 Size. The nozzle or nozzles shall conform to Figure 4. Nozzles may be of any convenient size. However, when a duct is connected to the inlet of the nozzle, the ratio of nozzle throat diameter to the diameter of the inlet duct shall not exceed 0.5.

5.3.2.2 Calibration. The standard nozzle is considered a primary instrument and need not be calibrated if maintained in the specified condition. Coefficients have been established for throat dimensions $L = 0.5\ D$ and $L = 0.6\ D$, shown in Figure 4 [8]. Throat dimension $L = 0.6\ D$ is recommended for new construction.

5.3.2.3 Chamber Nozzles. Nozzles without integral throat taps may be used for multiple nozzle chambers in which case upstream and downstream pressure taps shall be located as shown in the figure for the appropriate setup. Alternatively, nozzles with throat taps may be used in which case the throat taps located as shown in Figure 4 shall be used in place of the downstream pressure taps shown in the figure for the setup and the piezometer for each nozzle shall be connected to its own indicator.

5.3.2.4 Ducted Nozzles. Nozzles with integral throat taps shall be used for ducted nozzle setups. Upstream pressure taps shall be located as shown in the figure for the appropriate setup. Downstream taps are the integral throat taps and shall be located as shown in Figure 4.

5.3.2.5 Taps. All pressure taps shall conform to the specification in 5.2.3 regarding geometry, number, and manifolding into piezometer rings.

5.3.3 Other Airflow Measuring Methods. Airflow measuring methods which utilize meters or traverses other than flow nozzles or Pitot traverses may be used if the uncertainty introduced by the method does not exceed that introduced by an appropriate flow nozzle or Pitot traverse method. The contribution to the combined uncertainty in the airflow rate measurement shall not exceed that corresponding to 1.2% of the discharge coefficient for a flow nozzle [9].

Note 1: The specification permitting an indicator uncertainty based on the maximum observed reading during the test leads to combined relative uncertainties in both fan pressure and fan airflow rate that are higher at low values of the fan pressure or fan airflow rate than at high values of those test results. This is generally acceptable because fans are not usually rated at the low pressure or low flow portions of their characteristic curves. If there is a need to reduce the uncertainty at either low flow or low pressure, then the instruments chosen to measure the corresponding quantity must be selected with suitable accuracy (lower uncertainties) for those conditions.

5.4 Power. Power shall be determined from the rpm and beam load measured on a reaction dynamometer, the rpm and torque measured on a torsion element, or the electrical input measured on a calibrated motor.

5.4.1 Reaction Dynamometers. A cradle or torque table type reaction dynamometer having a demonstrated accuracy of ± 2% of observed reading may be used to measure power.

5.4.1.1 Calibration. A reaction dynamometer shall be calibrated through its range of usage by suspending weights from a torque arm. The weights shall have certified accuracies of ± 0.2%. The length of the torque arm shall be determined to an accuracy of ± 0.2%.

5.4.1.2 Tare. The zero torque equilibrium (tare) shall be checked before and after each test. The difference shall be within 0.5% of the maximum value measured during the test.

5.4.2 Torque. A torque meter having a demonstrated accuracy of ± 2% of observed reading may be used to determine power.

5.4.2.1 Calibration. A torque device shall have a static calibration and may have a running calibration through its range of usage. The static calibration shall be made by suspending weights from a torque arm. The weights shall have certified accuracies of ± 0.2%. The length of the torque arm shall be determined to an accuracy of ± 0.2%.

5.4.2.2 Tare. The zero torque equilibrium (tare) and the span of the readout system shall be checked before and after each test. In each case, the difference shall be within 0.5% of the maximum value measured during the test.

5.4.3 Calibrated Motors. A calibrated electric motor may be used with suitable electrical meters to measure power. It shall have a demonstrated accuracy of ±2%.

5.4.3.1 Calibration. The motor shall be calibrated through its range of usage against an absorption dynamometer except as provided in 5.4.3.4. The absorption dynamometer shall be calibrated by suspending weights from a torque arm. The weights shall have certified accuracies of ± 0.2%. The length of the torque arm shall be determined to an accuracy of ± 0.2%.

5.4.3.2 Meters. Electrical meters shall have certified accuracies of ±1.0% of observed reading. It is preferable that the same meters be used for the test as for the calibration.

5.4.3.3 Voltage. The motor input voltage during the test shall be within 1% of the voltage observed during calibration. If air flows over the motor from the fan under test, similar airflow shall be provided during calibration.

5.4.3.4 IEEE. Polyphase induction motors may be calibrated using the IEEE Segregated Loss Method {1}.

5.4.4 Averaging. Since the power required by a fan is never strictly steady, the torque measured on any instrument will fluctuate with time. In order to obtain a true reading, either the instrument must be damped or the readings must be averaged in a suitable manner. Averaging can sometimes be accomplished mentally, particularly if the fluctuations are small and regular. Multi-point or continuous record averaging can be accomplished with instruments and analyzers designed for this purpose.

5.5 Speed. Speed shall be measured with a revolution counter and chronometer, a stroboscope and chronometer, a precision instantaneous tachometer, or an electronic counter-timer.

5.5.1 Strobes. A stroboscopic device triggered by the line frequency of a public utility is considered a primary instrument and need not be calibrated if it is maintained in good condition.

5.5.2 Chronometers. A quality watch, with a sweep second hand or a digital display of seconds, that keeps time within two minutes per day is considered a primary instrument.

5.5.3 Other Devices. Any other device which has a demonstrated accuracy of ±0.5% of the value being measured may be used. Friction driven counters shall not be used when they can influence the speed due to drag.

5.6 Air Density. Air density shall be determined from measurements of wet-bulb temperature, dry-bulb temperature, and barometric pressure. Other parameters may be measured and used if the maximum error in the calculated density does not exceed 0.5%.

5.6.1 Thermometers. Wet-bulb and dry-bulb temperatures shall be measured with thermometers or other instruments with demonstrated accuracies of ±1°*C* (± 2°*F*) and readabilities of 0.5°*C* (1°*F*) or finer.

5.6.1.1 Calibration. Thermometers shall be calibrated over the range of temperatures to be encountered during test against a thermometer with a calibration that is traceable to the National Institute of Standards and Technology (NIST) or other national physical measures recognized as equivalent by NIST.

5.6.1.2 Wet-Bulb. The wet-bulb thermometer shall have an air velocity over the water-moistened wick-covered bulb of 3.5 to 10 *m/s* (700 to 2000 *fpm*) [10]. The dry-bulb thermometer shall be mounted upstream of the wet-bulb thermometer. Wet-bulb and dry-bulb thermometers should be matched.

5.6.2 Barometers. Barometric pressure shall be measured with a mercury column barometer or other instrument with a demonstrated accuracy of ±170 *Pa* (± 0.05 *in. Hg*) and readable to 34 *Pa* (0.01 *in. Hg*) or finer.

5.6.2.1 Calibration. Barometers shall be calibrated against a mercury column barometer with a calibration that is traceable to the National Institute of Standards and Technology or other national physical measures recognized as equivalent by NIST. A convenient method of doing this is to use an aneroid barometer as a transfer instrument and carry it back and forth to the Weather Bureau Station for comparison [11]. A permanently mounted mercury column barometer should hold its calibration well enough so that comparisons every three months should be sufficient. Transducer type barometers shall be calibrated for each test. Barometers shall be maintained in good condition.

5.6.2.2 Corrections. Barometric readings shall be corrected for any difference in mercury density from standard or any change in length of the graduated scale due to temperature. Refer to manufacturer's instructions and ASHRAE 41.3, Appendix B [12].

6. Equipment and Setups

6.1 Setups. Ten setups are diagramed in Figures 7 through 16.

6.1.1 Installation Types. There are four categories of installation types which are used with fans. They are [13]:

A: free inlet, free outlet
B: free inlet, ducted outlet
C: ducted inlet, free outlet
D: ducted inlet, ducted outlet.

6.1.2 Selection Guide. The following may be used as a guide to the selection of a proper setup.

(1) Figures 7 through 10 may be used for tests of installation types B or D.

In order to qualify for installation type D an inlet duct simulation shall be used.

(2) Figures 11 through 15 may be used for tests of installation types A, B, C, or D.

In order to qualify for installation type A the fan must be used without any auxiliary inlet bell or outlet duct.

In order to quality for installation type B an outlet duct shall be used and this may be of the short duct variety.

In order to qualify as installation type C an inlet duct simulation shall be used and no outlet duct shall be used.

In order to qualify as installation type D an inlet duct simulation shall be used and an outlet duct shall be used. The outlet duct may be of the short duct variety.

(3) Figure 16 may be used for tests of installation types C or D.

In order to qualify for installation type D an outlet duct shall be used and this may be of the short duct variety.

6.1.3 Leakage. The ducts, chambers, and other equipment utilized should be designed to withstand the pressure and other forces to be encountered. All joints between the fan and the measuring plane and across the nozzle wall should be designed for minimum leakage, because no leakage correction is permitted.

6.2 Ducts. A duct may be incorporated in a laboratory setup to provide a measuring plane or to simulate the conditions the fan is expected to encounter in service or both. The dimension D in the test setup figures is the inside diameter of a circular cross-section duct or equivalent diameter of a rectangular cross-section duct with inside transverse dimensions a and b where

$$D = \sqrt{4ab/\pi} \qquad \text{Eq. 6.1}$$

6.2.1 Flow Measuring Ducts. Ducts with measuring planes for airflow determination shall be straight and have uniform circular cross-sections. Pitot traverse ducts shall be at least 10 diameters long with the traverse plane located between 8.5 and 8.75 diameters from the upstream end. Such ducts may serve as an inlet or an outlet duct as well as to provide a measuring plane. Ducts connected to the upstream side of a flow nozzle shall be between 6.5 and 6.75 diameters long when used only to provide a measuring plane or between 9.5 and 9.75 diameters long when used as an outlet duct as well.

6.2.2 Pressure Measuring Ducts. Ducts with planes for pressure measurements shall be straight and may have either uniform circular or rectangular cross-sections. Outlet ducts with piezometer rings shall be at least 10 diameters long with the piezometer plane located between 8.5 and 8.75 diameters from the upstream end.

6.2.3 Short Ducts. Short outlet ducts which are used to simulate installation types B and D, but in which no measurements are taken shall be between 2 and 3 equivalent diameters long and an area within 1% of the fan outlet area and a uniform shape to fit the fan outlet [14].

6.2.4 Inlet Duct Simulation. Inlet bells or inlet bells and one equivalent duct diameter of inlet duct may be mounted on the fan inlet to simulate an inlet duct. The bell and duct shall be of the same size and shape as the fan inlet boundary connection.

6.2.5 Transformation Pieces. Transformation pieces shall be used when a duct with a measuring plane is to be connected to the fan and it is of a size or shape that differs from the fan connection. Such pieces shall not contain any converging element that makes an angle with the duct axis of greater than 7.5° or a diverging element that makes an angle with the duct axis of greater than 3.5°. The axes of the fan opening and duct shall coincide. See Figure 5. Connecting ducts and elbows of any size and shape may be used between a duct which provides a measuring plane and a chamber.

6.2.6 Duct Area. Outlet ducts used to provide measuring stations shall be not more than 5.0% larger or smaller than the fan outlet area. Inlet ducts used to provide measuring stations shall be not more than 12.5% larger nor 7.5% smaller than the fan inlet area. [15]

6.2.7 Roundness. The portion of a Pitot traverse duct within one-half duct diameter of either side of the plane of measurement shall be round within 0.5% of the duct diameter. The remainder of the duct shall be round within 1% of the duct diameter. The area of the plane of measurement shall be determined from the average of four diameters measured at 45° increments. The diameter measurements shall be accurate to 0.2%.

6.2.8 Straighteners. Straighteners are specified so that flow lines will be approximately parallel to the duct axis. Straighteners shall be used in all ducts which provide measuring planes. The downstream plane of the straightener shall be located between 5 and 5.25 duct diameters upstream of the plane of the Pitot traverse or piezometer station. The form of the straightener shall be as specified in Figure 6. To avoid excessive pressure drop through the flow straightener, careful attention to construction tolerances and details is important. [16]

6.3 Chambers. A chamber may be incorporated in a laboratory setup to provide a measuring station or to simulate the conditions the fan is expected to encounter in service or both. A chamber may have a circular or rectangular cross-sectional shape. The dimension M in the test setup diagram is the inside diameter of a circular chamber or the equivalent diameter of dimensions a and b where

$$M = \sqrt{4ab/\pi} \qquad \text{Eq. 6.2}$$

6.3.1 Outlet Chambers. An outlet chamber (Figure 11 or 12) shall have a cross-sectional area at least nine times the area of the fan outlet or outlet duct for fans with axis of rotation perpendicular to the discharge flow and a cross-sectional area at least sixteen times the area of the fan outlet or outlet duct for fans with axis of rotation parallel to the discharge flow. [17]

6.3.2 Inlet Chambers. Inlet chambers (Figures 13, 14, or 15) shall have a cross-sectional area at least five times the fan inlet area .

6.3.3 Flow Settling Means. Flow settling means shall be installed in chambers where indicated on the test setup figures to provide proper airflow patterns.

Where a measuring plane is located downstream of the settling means, the settling means is provided to ensure a substantially uniform airflow ahead of the measuring plane. In this case, the maximum local velocity at a distance 0.1 M downstream of the screen shall not exceed the average velocity by more than 25% unless the maximum local velocity is less than 2 *m/s* (400 *fpm*).

Where a measuring plane is located upstream of the settling means, the purpose of the settling screen is to absorb the kinetic energy of the upstream jet, and allow its normal expansion as if in an unconfined space. This requires some backflow to supply the air to mix at the jet boundaries, but the maximum reverse velocity shall not exceed 10% of the calculated Plane 2 or Plane 6 mean jet velocity.

Where measuring planes are located on both sides of the settling means within the chamber, the requirements for each side as outlined above shall be met.

Any combinations of screens or perforated plates that will meet these requirements may be used, but in general a reasonable chamber length for the settling means is necessary to meet both requirements. Screens of square mesh round wire with open areas of 50% to 60% are suggested and several will usually be needed to meet the above performance specifications. A performance check will be necessary to verify the flow settling means are providing proper flow patterns.

6.3.4 Multiple Nozzles. Multiple nozzles shall be located as symmetrically as possible. The centerline of each nozzle shall be at least 1.5 nozzle throat diameters from the chamber wall. The minimum distance between centers of any two nozzles in simultaneous use shall be three times the throat diameter of the larger nozzle.

The uncertainty of the airflow rate measurement can be reduced by changing to a smaller nozzle or combination of nozzles for the lower airflow rate range of the fan.

6.4 Variable Supply and Exhaust Systems. A means of varying the point of operation shall be provided in a laboratory setup.

6.4.1 Throttling Devices. Throttling devices may be used to control the point of operation of the fan. Such devices shall be located on the end of the duct or chamber and should be symmetrical about the duct or chamber axis.

6.4.2 Auxiliary Fans. Auxiliary fans may be used to control the point of operation of the test fan. They shall be designed to provide sufficient pressure at the desired flow rate to overcome losses through the test setup. Flow adjustment means, such as dampers, fan blade or fan inlet vane pitch control, or speed control may be required. Auxiliary fans shall not surge or pulsate during tests.

7. Observations and Conduct of Test

7.1 General Test Requirements

7.1.1 Determinations. The number of determinations required to establish the performance of a fan over the range from shut off to free delivery will depend on the shapes of the various characteristic curves. Plans shall be made to vary the opening of the throttling device in such a way that the test points will be well spaced. At least eight determinations shall be made. Additional determinations may be required to define curves for fans which exhibit dips or other discontinuities in one or more of the characteristic curves. When performance at only one point of operation or only over a portion of the performance range is required, the number of determinations shall be sufficient to define the performance range of interest, but at least three points are required to define a short curve for a single point of interest.

7.1.2 Equilibrium. Equilibrium conditions shall be established before each determination. To test for equilibrium, trial observations shall be made until steady readings are obtained. Ranges of air delivery over which equilibrium cannot be established shall be recorded.

7.1.3 Stability. Any bi-stable performance points (airflow rates at which two different pressure values can be measured) shall be reported. When they are a result of hysteresis, the points shall be identified as that for decreasing airflow rate and that for increasing airflow rate.

7.2 Data to be Recorded

7.2.1 Test Unit. The description of the test unit shall be recorded. The nameplate data should be copied. Dimensions should be checked against a drawing and a copy of the drawing attached to the data.

7.2.2 Test Setup. The description of the test setup including specific dimensions shall be recorded. Reference may be made to the figures in this standard. Alternatively, a drawing or annotated photograph of the setup may be attached to the data.

7.2.3 Instruments. The instruments and apparatus used in the test shall be listed. Names, model numbers, serial numbers, scale ranges, and calibration information should be recorded.

7.2.4 Test Data. Test data for each determination shall be recorded. Readings shall be made simultaneously whenever possible.

7.2.4.1 All Tests. For all types of tests, three readings of ambient dry-bulb temperature (t_{do}), ambient wet-bulb temperature (t_{wo}), ambient barometric pressure (p_b), fan outlet dry-bulb temperature (t_{d2}), fan speed (N), and either beam load (F), torque (T), or power input to motor (W) shall be recorded unless the readings are steady in which case only one need be recorded.

7.2.4.2 Pitot Test. For Pitot traverse tests, one reading each of velocity pressure (P_{v3r}) and static pressure (P_{s3r}) shall be recorded for each Pitot station. In addition, three readings of traverse-plane dry-bulb temperature (t_{d3}) shall be recorded unless the readings are steady in which case only one need be recorded.

7.2.4.3 Duct Nozzle Tests. For duct nozzle tests, one reading each of pressure drop (ΔP), approach dry-bulb temperature (t_{d4}), and approach static pressure (P_{s4}) shall be recorded.

7.2.4.4 Chamber Nozzle Tests. For chamber nozzle tests, the nozzle combinations and one reading each of pressure drop (ΔP), approach dry-bulb temperature (t_{d5}), approach static pressure (P_{s5}), shall be recorded.

7.2.4.5 Inlet Chamber Tests. For inlet chamber tests, one reading each of inlet chamber dry-bulb temperature (t_{d8}) and inlet chamber total pressure (P_{t8}) shall be recorded.

7.2.4.6 Outlet Chamber Tests. For outlet chamber tests, one reading each of outlet chamber dry-bulb temperature (t_{d7}) and outlet chamber static pressure (P_{s4}) shall be recorded.

7.2.4.7 Outlet Duct Chamber Tests. For outlet duct chamber tests, one reading each of outlet duct dry-bulb temperature (t_{d4}) and outlet duct static pressure (P_{s4}) shall be recorded.

7.2.4.8 Low Pressure Tests. For tests where P_s is less than 1 *kPa* (4 *in. wg*) the temperatures may be considered uniform throughout the test setup and only t_{do} and t_{wo} need be measured.

7.2.5 **Personnel.** The names of test personnel shall be listed with the data for which they are responsible.

8. Calculations

8.1 Calibration Correction. Calibration correction, when required, shall be applied to individual readings before averaging or other calculations. Calibration correction need not be made if the correction is smaller than one half the maximum allowable error as specified in Section 5.

8.2 Density and Viscosity of Air

8.2.1 Atmospheric Air Density. The density of atmospheric air (ρ_o) shall be determined from measurements, taken in the general test area, of dry-bulb temperature (t_{do}), wet-bulb temperature (t_{wo}), and barometric pressure (p_b) using the following formulae [18].

$$p_e = 3.25 t_{wo}^2 + 18.6 t_{wo} + 692 \text{ Pa} \qquad \text{Eq. 8.1 SI}$$

$$p_e = (2.96\text{E-04}) t_{wo}^2 - (1.59\text{E-02}) t_{wo} + 0.41, \qquad \text{Eq. 8.1 I-P}$$

$$p_p = p_e - p_b \left(\frac{t_{do} - t_{wo}}{1500} \right), \text{ and} \qquad \text{Eq. 8.2 SI}$$

$$p_p = p_e - p_b \left(\frac{t_{do} - t_{wo}}{2700} \right), \text{ and} \qquad \text{Eq. 8.2 I-P}$$

$$\rho_o = \frac{(p_b - 0.378\, p_p)}{R\,(t_{do} + 273.15)} \qquad \text{Eq. 8.3 SI}$$

$$\rho_o = \frac{70.73\,(p_b - 0.378\, p_p)}{R\,(t_{do} + 459.67)} \qquad \text{Eq. 8.3 I-P}$$

The first equation (8.1) is approximately correct for p_e for a range of t_{wo} between 4°*C* and 32°*C* (40°*F* and 90° *F*). More precise values of p_e can be obtained from the ASHRAE Handbook of Fundamentals [19]. The gas constant (R) may be taken as 287.1 *J/kgK* (53.35 *ft•lbf/-lbm•°R*) for air.

8.2.2 Duct or Chamber Air Density. The density of air in a duct or chamber at Plane x (ρ_x) may be calculated by correcting the density of atmospheric air (ρ_o) for the pressure (P_{sx}) and temperature (t_{dx}) at Plane x using

$$\rho_x = \rho_o \left(\frac{t_{do} + 273.15}{t_{dx} + 273.15} \right) \left(\frac{P_{sx} + p_b}{p_b} \right) \qquad \text{Eq. 8.4 SI}$$

$$\rho_x = \rho_o \left(\frac{t_{do} + 459.67}{t_{dx} + 459.67} \right) \left(\frac{P_{sx} + 13.63 p_b}{13.63 p_b} \right) \qquad \text{Eq. 8.4 I-P}$$

8.2.3 Fan Air Density. The fan air density (ρ) shall be calculated from the density of atmospheric air (ρ_o), the total pressure at the fan inlet (P_{t1}), and the total temperature at the fan inlet (t_{s1}) using

$$\rho = \rho_o \left(\frac{P_{s1} + p_b}{p_b} \right) \left(\frac{t_{do} + 273.15}{t_{s1} + 273.15} \right) \qquad \text{Eq. 8.5 SI}$$

$$\rho = \rho_o \left(\frac{P_{s1} + 13.63 p_b}{13.63 p_b} \right) \left(\frac{t_{do} + 459.67}{t_{s1} + 459.67} \right) \qquad \text{Eq. 8.5 I-P}$$

On all outlet duct and outlet chamber setups, P_{t1} is equal to zero and t_{t1} is equal to t_{do}. On all inlet chamber setups, P_{t1} is equal to P_{t8} and t_{s1} is equal to t_{d8}. On the inlet duct setup, t_{s1} is equal to t_{d3} and P_{t1} may be considered equal to P_{t3} for fan air density calculations.

8.2.4 Dynamic Air Viscosity. The viscosity (μ) shall be calculated from

$$\mu = (17.23 + 0.048 t_d)\text{ E-06} \qquad \text{Eq. 8.6 SI}$$

$$\mu = (11.00 + 0.018 t_d)\text{ E-06} \qquad \text{Eq. 8.6 I-P}$$

The value for 20°*C* (68°*F*) air, which is 1.819E-05 *Pa-s* (1.222E-05 *lbm/ft•s*), may be used between 4°*C* (40°*F*) and 40°*C* (100°*F*) [20].

8.3 Fan Airflow Rate at Test Conditions

8.3.1 Velocity Traverse. The fan airflow rate may be calculated from velocity pressure measurements (P_{v3}) taken by Pitot traverse.

8.3.1.1 Velocity Pressure. The velocity pressure (P_{v3}) corresponding to the average velocity shall be obtained by taking the square roots of the individual measurements (P_{v3r}) (see Figure 3), summing the roots, dividing the sum by the number of measurements (n), and squaring the quotient as indicated by

$$P_{v3} = \left(\frac{\Sigma \sqrt{P_{v3r}}}{n} \right)^2 \qquad \text{Eq. 8.7}$$

8.3.1.2 Velocity. The average velocity (V_3) shall be obtained from the density at the plane of traverse (ρ_3) and the corresponding velocity pressure (P_{v3}) using

$$V_3 = \sqrt{\frac{2P_{v3}}{\rho_3}} \qquad \text{Eq. 8.8 SI}$$

$$V_3 = 1097\sqrt{\frac{P_{v3}}{\rho_3}} \qquad \text{Eq. 8.8 I-P}$$

8.3.1.3 Airflow Rate. The airflow rate (Q_3) at the Pitot traverse plane shall be obtained from the velocity (V_3) and the area (A_3) using

$$Q_3 = V_3 A_3 \qquad \text{Eq. 8.9}$$

8.3.1.4 Fan Airflow Rate. The fan airflow rate at test conditions (Q) shall be obtained from the equation of continuity,

$$Q = Q_3\,(\rho_3/\rho) \qquad \text{Eq. 8.10}$$

8.3.2 Nozzle. The fan airflow rate may be calculated from the pressure differential (ΔP) measured across a single nozzle or a bank of multiple nozzles. [21]

8.3.2.1 Alpha Ratio. The ratio of absolute nozzle exit pressure to absolute approach pressure shall be calculated from

$$a = \frac{P_{s6} + p_b}{P_{sx} + p_b} \text{ or} \qquad \text{Eq. 8.11 SI}$$

$$a = \frac{P_{s6} + 13.63p_b}{P_{sx} + 13.63p_b} \text{ or} \qquad \text{Eq. 8.11 I-P}$$

$$a = 1 - \frac{\Delta P}{\rho_x R(t_{dx} + 273.15)} \qquad \text{Eq. 8.12 SI}$$

$$a = 1 - \frac{5.187\,\Delta P}{\rho_x R(t_{dx} + 459.67)} \qquad \text{Eq. 8.12 I-P}$$

The gas constant (R) may be taken as 287.1 *J/kg•K* (53.35 *ft•lbf/lbm•°R*) for air. Plane x is Plane 4 for duct approach or Plane 5 for chamber approach.

8.3.2.2 Beta Ratio. The ratio (β) of nozzle exit diameter (D_6) to approach duct diameter (D_x) shall be calculated from

$$\beta = D_6/D_x \qquad \text{Eq. 8.13}$$

For a duct approach $D_x = D_4$. For a chamber approach, $D_x = D_5$, and β may be taken as zero.

8.3.2.3 Expansion Factor. The expansion factor (Y) may be obtained from

$$Y = \left[\frac{\gamma}{\gamma - 1}\,\alpha^{2/\gamma}\,\frac{1 - \alpha^{(\gamma-1)/\gamma}}{1 - \alpha}\right]^{1/2} \left[\frac{1 - \beta^4}{1 - \beta^4\alpha^{2/\gamma}}\right]^{1/2} \qquad \text{Eq. 8.14}$$

The ratio of specific heats (γ) may be taken as 1.4 for air. Alternatively, the expansion factor for air may be approximated with sufficient accuracy by:

$$Y = 1 - (0.548 + 0.71\,\beta^4)\,(1 - \alpha) \qquad \text{Eq. 8.15}$$

8.3.2.4 Energy Factor. The energy factor (E) may be determined by measuring velocity pressures (P_{vr}) upstream of the nozzle at standard traverse stations and calculating

$$E = \frac{\left(\dfrac{\Sigma\left(P_{vr}^{\ 3/2}\right)}{n}\right)}{\left(\dfrac{\Sigma\left(P_{vr}^{\ 1/2}\right)}{n}\right)^3} \qquad \text{Eq. 8.16}$$

Sufficient accuracy can be obtained for setups qualifying under this standard by setting $E = 1.0$ for chamber approach or $E = 1.043$ for duct approach [8].

8.3.2.5 Reynolds Number. The Reynolds number (Re) based on nozzle exit diameter (D_6) in *m* (*ft*) shall be calculated from

$$Re = \frac{D_6 V_6 \rho_6}{\mu} \qquad \text{Eq. 8.17 SI}$$

$$Re = \frac{D_6 V_6 \rho_6}{60\mu} \qquad \text{Eq. 8.17 I-P}$$

using properties of air as determined in 8.2 and the appropriate velocity (V_6) in *m/s (fpm)*. Since the velocity determination depends on Reynolds number an approximation must be employed. It can be shown that

$$Re = \frac{\sqrt{2}}{\mu} C D_6 Y \sqrt{\frac{\Delta P \rho_x}{1 - E\beta^4}} \qquad \text{Eq. 8.18 SI}$$

$$Re = \frac{1097}{60\,\mu} C D_6 Y \sqrt{\frac{\Delta P \rho_x}{1 - E\beta^4}} \qquad \text{Eq. 8.18 I-P}$$

For duct approach $\rho_x = \rho_4$. For chamber approach $\rho_x = \rho_5$, and β may be taken as zero.

Refer to Appendix F for an example of an iterative process to determine *Re* and *C*.

8.3.2.6 Discharge Coefficient. The nozzle discharge coefficient (*C*) shall be determined from

$$C = 0.9986 - \frac{7.006}{\sqrt{Re}} + \frac{134.6}{Re} \quad \text{for} \quad \frac{L}{D} = 0.6$$

Eq. 8.19

$$C = 0.9986 - \frac{6.688}{\sqrt{Re}} + \frac{131.5}{Re} \quad \text{for} \quad \frac{L}{D} = 0.5$$

Eq. 8.20

for *Re* of 12,000 and above [8].

Refer to Appendix F for an example of an iterative process to determine *Re* and *C*.

8.3.2.7 Airflow Rate for Ducted Nozzle. The volume airflow rate (Q_4) at the entrance to a ducted nozzle shall be calculated from

$$Q_4 = \frac{CA_6\, Y\sqrt{2\Delta P / \rho_4}}{\sqrt{1 - E\mathrm{B}^4}} \qquad \text{Eq. 8.21 SI}$$

$$Q_4 = \frac{1097\, CA_6 Y\sqrt{\Delta P / \rho_4}}{\sqrt{1 - E\beta^4}} \qquad \text{Eq. 8.21 I-P}$$

The area (A_6) is measured at the plane of the throat taps.

8.3.2.8 Airflow Rate for Chamber Nozzles. The volume airflow rate (Q_5) at the entrance to a nozzle or multiple nozzles with chamber approach shall be calculated from

$$Q_5 = Y \sqrt{\frac{2\Delta P}{\rho_5}}\, \Sigma\left(CA_6\right) \qquad \text{Eq. 8.22 SI}$$

$$Q_5 = 1097\ Y \sqrt{\frac{\Delta P}{\rho_5}}\, \Sigma\left(CA_6\right) \qquad \text{Eq. 8.22 I-P}$$

The coefficient (*C*) and area (A_6) must be determined for each nozzle and their products summed as indicated. The area (A_6) is measured at the plane of the throat taps or the nozzle exit for nozzles without throat taps.

8.3.2.9 Fan Airflow Rate. The fan airflow rate (*Q*) at test conditions shall be obtained from the equation of continuity,

$$Q = Q_x\,(\rho_x/\rho) \qquad \text{Eq. 8.23}$$

where Plane x is either Plane 4 or Plane 5 as appropriate.

8.4 Fan Velocity Pressure at Test Conditions

8.4.1 Pitot Traverse. When Pitot traverse measurements are made, the fan velocity pressure (P_v) shall be determined from the velocity pressure (P_{v3}) using

$$P_v = P_{v3}\left(\frac{\rho_3}{\rho_2}\right)\left(\frac{A_3}{A_2}\right)^2 \qquad \text{Eq. 8.24}$$

Whenever P_{s3} and P_{s2} differ by less than 1 *kPa* (4 *in. wg*), ρ_2 may be considered equal to ρ_3.

8.4.2 Nozzle. When airflow rate is determined from nozzle measurements, the fan velocity pressure (P_v) shall be calculated from the velocity (V_2) and density (ρ_2) at the fan outlet using

$$Q_2 = Q\,(\rho/\rho_2) \qquad \text{Eq. 8.25}$$

$$V_2 = Q_2/A_2 \qquad \text{Eq. 8.26}$$

and

$$P_v = \frac{\rho_2 V_2^2}{2} \qquad \text{Eq. 8.27 SI}$$

$$P_v = \rho_2 \left(\frac{V_2}{1097}\right)^2 \qquad \text{Eq. 8.27 I-P}$$

or

$$P_v = \left(\frac{Q\rho}{A_2}\right)^2 \frac{1}{2\rho_2} \qquad \text{Eq. 8.28 SI}$$

$$P_v = \left(\frac{Q\rho}{1097A_2}\right)^2 \frac{1}{\rho_2} \qquad \text{Eq. 8.28 I-P}$$

For outlet duct setups, whenever P_{s4} and P_{s2} differ by less than 1 *kPa* (4 *in. wg*), ρ_2 may be considered equal to ρ_4.

8.5 Fan Total Pressure at Test Conditions. The fan total pressure shall be calculated from measurements of pressures in ducts or chambers corrected for pressure losses in measuring ducts which occur between the fan and the measuring stations.

8.5.1 Averages. Certain averages shall be calculated from measurements as follows:

8.5.1.1 When a Pitot traverse is used for pressure measurement: the average velocity pressure (P_{v3}) shall be as determined in 8.3.1.1. The average velocity (V_3) shall be as determined in 8.3.1.2, and the average static pressure (P_{s3}) shall be calculated from

$$P_{s3} = \frac{\Sigma P_{s3r}}{n} \qquad \text{Eq. 8.29}$$

8.5.1.2 Duct Piezometer. When a duct piezometer is used for pressure measurement the average static pressure (P_{s4}) shall be the measured value (P_{s4r}).The average velocity (V_4) shall be calculated from the airflow rate (Q) as determined in 8.3.2.9, and

$$V_4 = \left(\frac{Q}{A_4}\right)\left(\frac{\rho}{\rho_4}\right) \qquad \text{Eq. 8.30}$$

and the average velocity pressure (P_{v4}) shall be calculated from

$$P_{v4} = \frac{\rho_4 V_4^2}{2} \qquad \text{Eq. 8.31 SI}$$

$$P_{v4} = \rho_4 \left(\frac{V_4}{1097}\right)^2 \qquad \text{Eq. 8.31 I-P}$$

8.5.1.3 Chamber. When a chamber piezometer or total pressure tube is used for pressure measurement, the average static pressure (P_{s7}) shall be the measured value (P_{s7r}) and the average total pressure (P_{t8}) shall be the measured value (P_{t8r}).

8.5.2 Pressure Losses. Pressure losses shall be calculated for measuring ducts and straighteners which are located between the fan and the measuring station.

8.5.2.1 Hydraulic Diameter. The hydraulic diameter for round ducts is the actual diameter (D). The hydraulic diameter for rectangular ducts shall be calculated from the inside traverse dimensions a and b using

$$D_h = 2\,ab/(a + b) \qquad \text{Eq. 8.32}$$

8.5.2.2 Reynolds Number. The Reynolds number (Re) based on the hydraulic diameter (D_h) in *m* (*ft*) shall be calculated from

$$Re = \frac{D_h V \rho}{\mu} \qquad \text{Eq. 8.33 SI}$$

$$Re = \frac{D_h V \rho}{60\mu} \qquad \text{Eq. 8.33 I-P}$$

using properties of air as determined in 8.2 and the appropriate velocity (V) in *m/s* (*fpm*).

8.5.2.3 Coefficient of Friction. The coefficient of friction (f) shall be determined from [22]:

$$f = \frac{0.14}{Re^{0.17}} \qquad \text{Eq. 8.34}$$

8.5.2.4 Straightener Equivalent Length. [22] The ratio of equivalent length of a straightener (L_e) to hydraulic diameter (D_h) shall be determined from the element thickness (y) and equivalent diameter (D) using

$$\frac{L_e}{D_h} = \frac{15.04}{\left[1 - 26.65\left(\frac{y}{D}\right) + 184.6\left(\frac{y}{D}\right)^2\right]^{1.83}} \qquad \text{Eq. 8.35}$$

This expression is exact for round duct straighteners and sufficiently accurate for rectangular duct straighteners.

8.5.3 Inlet Total Pressure. The total pressure at the fan inlet (P_{t1}) shall be calculated as follows:

8.5.3.1 Open Inlet. When the fan draws directly from atmosphere, P_{t1} shall be considered equal to atmospheric pressure, which is zero gauge, so that

$$P_{t1} = 0 \qquad \text{Eq. 8.36}$$

8.5.3.2 Inlet Chamber. When the fan is connected to an inlet chamber, P_{t1} shall be considered equal to the chamber pressure (P_{t8}), so that

$$P_{t1} = P_{t8} \qquad \text{Eq. 8.37}$$

8.5.3.3 Inlet Duct. When the fan is connected to an inlet duct, P_{t1} shall be considered equal to the algebraic sum of the average static pressure (P_{s3}) and the average velocity pressure (P_{v3}) corrected for the friction due to the length of duct ($L_{1,3}$) between the measuring station and the fan, so that

$$P_{t1} = P_{s3} + P_{v3} - f\frac{L_{1,3}}{D_{h3}} P_{v3} \qquad \text{Eq. 8.38}$$

Pressure P_{s3} will be less than atmospheric and its value will be negative.

8.5.4 Outlet Total Pressure. The total pressure at the fan outlet (P_{t2}) shall be calculated as follows:

8.5.4.1 Open Outlet. When the fan discharges directly to atmosphere, the static pressure at the outlet (P_{s2}) shall be considered equal to atmospheric pressure, which is zero gauge, so that

$$P_{t2} = P_{v2} = P_v \qquad \text{Eq. 8.39}$$

The value of P_v shall be as determined in 8.4.

8.5.4.2 Outlet Chamber. When the fan discharges directly into an outlet chamber, the static pressure at the outlet (P_{s2}) shall be considered equal to the average chamber pressure (P_{s7}), so that

$$P_{t2} = P_{s7} + P_{v2} = P_{s7} + P_v \qquad \text{Eq. 8.40}$$

The value of P_v shall be as determined in 8.4.

8.5.4.3 Short Duct. When the fan discharges through an outlet duct without a measuring station either to atmosphere or into an outlet chamber, the pressure loss of the duct shall be considered zero and calculations made according to 8.5.4.1 or 8.5.4.2.

8.5.4.4 Piezometer Outlet Duct. When the fan discharges into a duct with a piezometer ring, P_{t2} shall be considered equal to the sum of the average static pressure (P_{s4}) and the average velocity pressure (P_{v4}) corrected for the friction due to both the equivalent length of the straightener (L_e) and the length of duct ($L_{2,4}$) between the fan and the measuring station, so that

$$P_{t2} = P_{s4} + P_{v4} + f\left(\frac{L_{2,4}}{D_{h4}} + \frac{L_e}{D_{h4}}\right) P_{v4} \qquad \text{Eq. 8.41}$$

8.5.4.5 Pitot Outlet Duct. When the fan discharges into a duct with a Pitot traverse, P_{t2} shall be considered equal to the sum of the average static pressure (P_{s3}) and the average velocity pressure (P_{v3}) corrected for the friction due to both the equivalent length of the straightener (L_e) and the length of duct ($L_{2,3}$) between the fan and the measuring station, so that

$$P_{t2} = P_{s3} + P_{v3} + f\left(\frac{L_{2,3}}{D_{h3}} + \frac{L_e}{D_{h3}}\right) P_{v3} \qquad \text{Eq. 8.42}$$

8.5.5 Fan Total Pressure. The fan total pressure (P_t) shall be calculated from

$$P_t = P_{t2} - P_{t1} \qquad \text{Eq. 8.43}$$

This is an algebraic expression so that if P_{t1} is negative, P_t will be numerically greater than P_{t2}.

8.6 Fan Static Pressure at Test Conditions

The fan static pressure (P_s) shall be calculated from

$$P_s = P_t - P_v \qquad \text{Eq. 8.44}$$

8.7 Fan Power Input at Test Conditions

8.7.1 Reaction Dynamometer. When a reaction dynamometer is used to measure torque, the fan power input (H) shall be calculated from the beam load (F), using the moment arm (l), and the fan speed (N) using

$$H = \frac{2\pi F l N}{60} \qquad \text{Eq. 8.45 SI}$$

$$H = \frac{2\pi F l N}{33000 \times 12} \qquad \text{Eq. 8.45 I-P}$$

8.7.2 Torsion Element. When a torsion element is used to measure torque, the fan power input (H) shall be calculated from the torque (T) and the speed (N) using

$$H = \frac{2\pi T N}{60} \qquad \text{Eq. 8.46 SI}$$

$$H = \frac{2\pi T N}{33000 \times 12} \qquad \text{Eq. 8.46 I-P}$$

8.7.3 Calibrated Motor. When a calibrated electric motor is used to measure input, the fan power input (H) may be calculated from the power input to the motor (W) and the motor efficiency (η) using

$$H = W\eta \qquad \text{Eq. 8.47 SI}$$

$$H = \frac{W\eta}{745.7} \qquad \text{Eq. 8.47 I-P}$$

8.8 Fan Efficiency

8.8.1 Fan Power Output. The fan power output (H_o) would be proportional to the product of fan airflow rate (Q) and fan total pressure (P_t) if air were incompressible. Since air is compressible, thermodynamic effects influence output and a compressibility coefficient (K_p) must be applied making output proportional to $Q\,P_t\,K_p$ [23].

$$H_o = Q P_t K_p \qquad \text{Eq. 8.48 SI}$$

$$H_o = \frac{Q\,P_t\,K_p}{6362} \qquad \text{Eq. 8.48 I-P}$$

8.8.2 Compressibility Factor. The compressibility coefficient (K_p) may be determined from

$$x = \frac{P_t}{P_{t1} + p_b} \qquad \text{Eq. 8.49 SI}$$

$$x = \frac{P_t}{P_{t1} + 13.63 p_b} \text{ and} \qquad \text{Eq. 8.49 I-P}$$

$$z = \left(\frac{\gamma - 1}{\gamma}\right)\left(\frac{H/Q}{P_{t1} + p_b}\right) \qquad \text{Eq. 8.50 SI}$$

$$z = \left(\frac{\gamma - 1}{\gamma}\right)\left(\frac{6362\,H/Q}{P_{t1} + 13.63 p_b}\right) \qquad \text{Eq. 8.50 I-P}$$

using

$$K_p = \left(\frac{\ln(1 + x)}{x}\right)\left(\frac{z}{\ln(1 + z)}\right) \qquad \text{Eq. 8.51}$$

which may be evaluated directly [23]. P_t, P_{t1}, p_b, H, and Q are all test values. The isentropic exponent (γ) may be taken as 1.4 for air.

8.8.3 Fan Total Efficiency. The fan total efficiency (η_t) is the ratio of fan power output to fan power input or

$$\eta_t = \frac{Q P_t K_p}{H} \qquad \text{Eq. 8.52 SI}$$

$$\eta_t = \frac{Q P_t K_p}{6362\ H} \qquad \text{Eq. 8.52 I-P}$$

8.8.4 Fan Static Efficiency. The fan static efficiency (η_s) may be calculated from the fan total efficiency (η_t) and the ratio of fan static pressure to fan total pressure using

$$\eta_s = \eta_t\left(\frac{P_s}{P_t}\right) \qquad \text{Eq. 8.53}$$

8.9 Conversion to Nominal Constant Values of Density and Speed.

During a laboratory test, the air density and speed of rotation may vary slightly from one determination to another. It may be desirable to convert the results calculated for test conditions to those that would prevail at nominal constant density, nominal constant speed, or both. This may be done provided the nominal constant density (ρ_c) is within 10% of the actual density (ρ) and the nominal constant speed (N_c) is within 5% of the actual speed (N).

8.9.1 Compressibility Factor Ratio. In order to make the conversions it is necessary to determine the ratio of the compressibility coefficient for actual conditions to that for nominal conditions (K_p/K_{pc}).

This can be accomplished using previously calculated values of x and z for actual conditions as follows:

$$\frac{z}{z_c} = \left(\frac{P_{t1c} + p_{bc}}{P_{t1} + p_b}\right)\left(\frac{\rho}{\rho_c}\right)\left(\frac{N}{N_c}\right)^2\left(\frac{\gamma_c}{\gamma_c - 1}\right)\left(\frac{\gamma - 1}{\gamma}\right) \qquad \text{Eq. 8.54 SI}$$

$$\frac{z}{z_c} = \left(\frac{P_{t1c} + 13.63 p_{bc}}{P_{t1} + 13.63 p_b}\right)\left(\frac{\rho}{\rho_c}\right)\left(\frac{N}{N_c}\right)^2\left(\frac{\gamma_c}{\gamma_c - 1}\right)\left(\frac{\gamma - 1}{\gamma}\right) \qquad \text{Eq. 8.54 I-P}$$

(Since the ratios of specific heats γ_c and γ are equal for air at laboratory conditions, the last two factors may be omitted in these and the following equations.)

$$z_c = z/(z/z_c) \qquad \text{Eq. 8.55}$$

$$\ln(1 + x_c) = \ln(1 + x)\left(\frac{\ln(1 + z_c)}{\ln(1 + z)}\right)\left(\frac{\gamma - 1}{\gamma}\right)\left(\frac{\gamma_c}{\gamma_c - 1}\right) \qquad \text{Eq. 8.56}$$

$$x_c = e^{\ln(1 + x_c)} - 1, \text{ and} \qquad \text{Eq. 8.57}$$

$$\frac{K_p}{K_{pc}} = \left(\frac{z}{z_c}\right)\left(\frac{x_c}{x}\right)\left(\frac{\gamma}{\gamma - 1}\right)\left(\frac{\gamma_c - 1}{\gamma_c}\right) \qquad \text{Eq. 8.58}$$

8.9.2 Conversion Formulae. Actual test results may be converted to nominal test results using the following [23]:

$$Q_c = Q\left(\frac{N_c}{N}\right)\left(\frac{K_p}{K_{pc}}\right) \qquad \text{Eq. 8.59}$$

$$P_{tc} = P_t\left(\frac{N_c}{N}\right)^2\left(\frac{\rho_c}{\rho}\right)\left(\frac{K_p}{K_{pc}}\right) \qquad \text{Eq. 8.60}$$

$$P_{vc} = P_v\left(\frac{N_c}{N}\right)^2\left(\frac{\rho_c}{\rho}\right) \qquad \text{Eq. 8.61}$$

$$P_{sc} = P_{tc} - P_{vc} \qquad \text{Eq. 8.62}$$

$$H_c = H\left(\frac{N_c}{N}\right)^3\left(\frac{\rho_c}{\rho}\right)\left(\frac{K_p}{K_{pc}}\right) \qquad \text{Eq. 8.63}$$

$$\eta_{tc} = \eta_t, \text{ and} \qquad \text{Eq. 8.64}$$

$$\eta_{sc} = \eta_{tc}\left(\frac{P_{sc}}{P_{tc}}\right) \qquad \text{Eq. 8.65}$$

9. Report and Results of Test

9.1 Report. The report of a laboratory fan test shall include object, results, test data, and descriptions of the test fan including appurtenances, test figure and installation type, test instruments and personnel as outlined in Section 7. The test report shall also state the inlet, outlet, and power boundaries of the fan and what appurtenances were included with them. The laboratory shall be identified by name and location.

9.2 Performance Curves. The results of a fan test shall be presented as performance curves. Typical fan performance curves are shown in Figure 17.

9.2.1 Coordinates and Labeling. Performance curves shall be drawn with fan airflow rate as abscissa. Fan pressure and fan power input shall be plotted as ordinates. Fan total pressure, fan static pressure, or both may be shown. If all results were obtained at the same speed or if results were converted to a nominal speed, such speed shall be listed; otherwise a curve with fan speed as ordinate shall be drawn. If all results were obtained at the same air density or if results were converted to a nominal density, such density shall be listed; otherwise a curve with fan air density as ordinate shall be drawn. Curves with fan total efficiency or fan static efficiency as ordinates may be drawn. Barometric pressure shall be listed when fan pressures exceed 2.5 *kPa* (10 *in. wg*).

9.2.2 Test Points. The results for each determination shall be shown on the performance curve as a series of "circled" points, one for each variable plotted as ordinate.

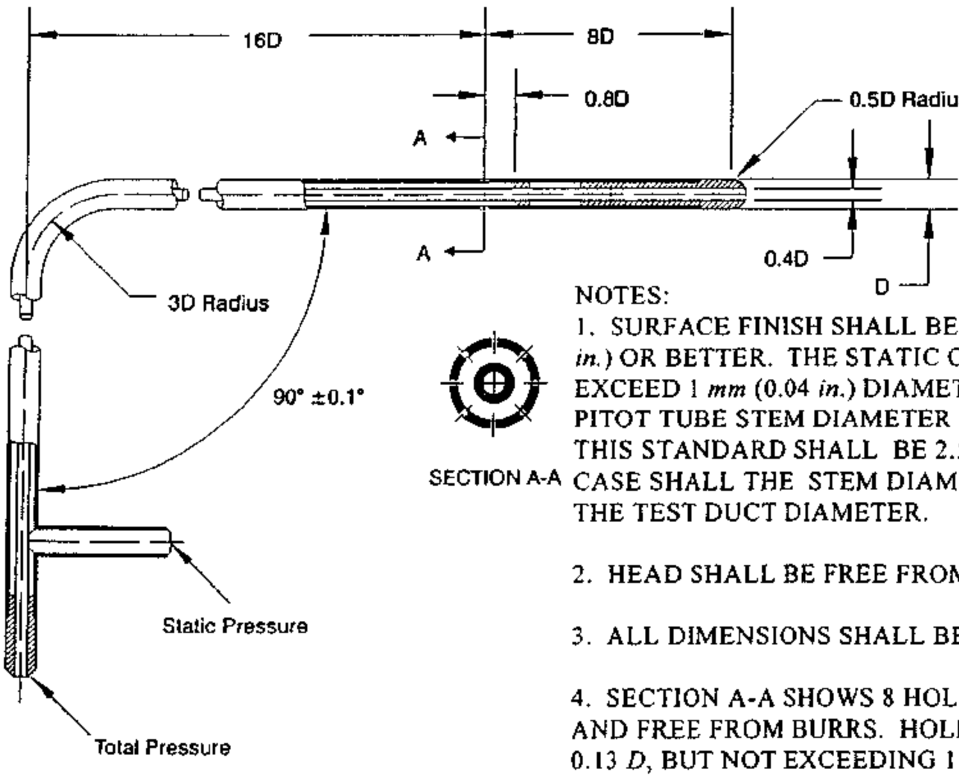
9.2.3 Curve-Fitting. Curves for each variable shall be obtained by drawing a curve or curves using the test points for reference. The curves shall not depart from the test points by more than 0.5% of any test value and the sum of the deviations shall approximate zero.

9.2.4 Discontinuities. When discontinuities exist they shall be identified with a broken line. If equilibrium cannot be established for any determination, the curves joining the points for that determination with adjacent points shall be drawn as broken lines.

9.2.5 Identification. Performance curve sheets shall list the test fan and test setup. Sufficient details shall be listed to identify clearly the fan and setup. Otherwise a report containing such information shall be referenced.

10. Normative References

The following reference is normative: {1} IEEE 112-1984(R1996) *Standard Test Procedure for Polyphase Induction Motors and Generators,* The Institute of Electrical and Electronics Engineers, New York, NY, U.S.A. (AMCA #1149)



NOTES:

1. SURFACE FINISH SHALL BE 0.8 micrometer (32 *micro-in.*) OR BETTER. THE STATIC ORIFICES MAY NOT EXCEED 1 *mm* (0.04 *in.*) DIAMETER. THE MINIMUM PITOT TUBE STEM DIAMETER RECOGNIZED UNDER THIS STANDARD SHALL BE 2.5 *mm* (0.10 *in.*). IN NO CASE SHALL THE STEM DIAMETER EXCEED 1/30 OF THE TEST DUCT DIAMETER.

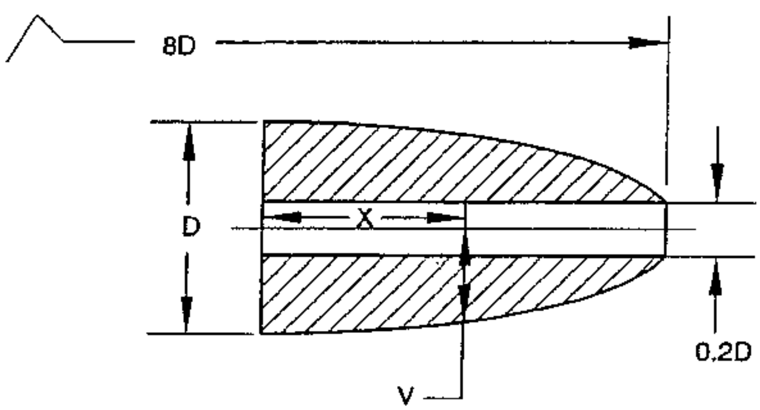
2. HEAD SHALL BE FREE FROM NICKS AND BURRS

3. ALL DIMENSIONS SHALL BE WITHIN ±2%.

4. SECTION A-A SHOWS 8 HOLES EQUALLY SPACED AND FREE FROM BURRS. HOLE DIAMETER SHALL BE 0.13 *D*, BUT NOT EXCEEDING 1 *mm* (0.04 *in.*) HOLE DEPTH SHALL NOT BE LESS THAN THE HOLE DIAMETER

PITOT-STATIC TUBE WITH SPHERICAL HEAD

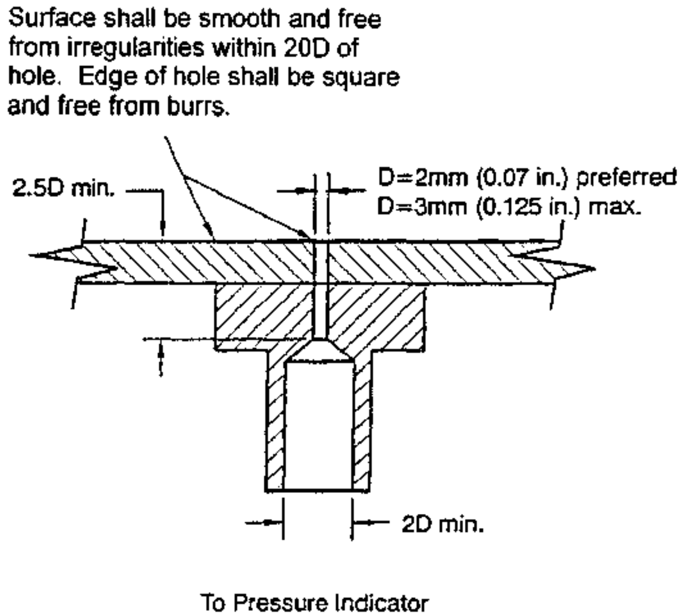
ALL OTHER DIMENSIONS ARE THE SAME AS FOR SPHERICAL HEAD PITOT-STATIC TUBES.



X/D	V/D	X/D	V/D
0.000	0.500	1.602	0.314
0.237	0.496	1.657	0.295
0.336	0.494	1.698	0.279
0.474	0.487	1.730	0.266
0.622	0.477	1.762	0.250
0.741	0.468	1.796	0.231
0.936	0.449	1.830	0.211
1.025	0.436	1.858	0.192
1.134	0.420	1.875	0.176
1.228	0.404	1.888	0.163
1.313	0.388	1.900	0.147
1.390	0.371	1.910	0.131
1.442	0.357	1.918	0.118
1.506	0.343	1.920	0.109
1.538	0.333	1.921	0.100
1.570	0.323		

ALTERNATE PITOT-STATIC TUBE WITH ELLIPSOIDAL HEAD

Figure 1 Pitot-Static Tubes



NOTE: A 2 *mm* (0.07 *in.*) HOLE IS THE MAXIMUM SIZE WHICH WILL ALLOW SPACE FOR A SMOOTH SURFACE 20 *D* FROM THE HOLE WHEN INSTALLED 38 *mm* (1.5 *in.*) FROM A PARTITION, SUCH AS IN FIGURES 9, 10, 11, 12, 15.

Figure 2A Static Pressure Tap

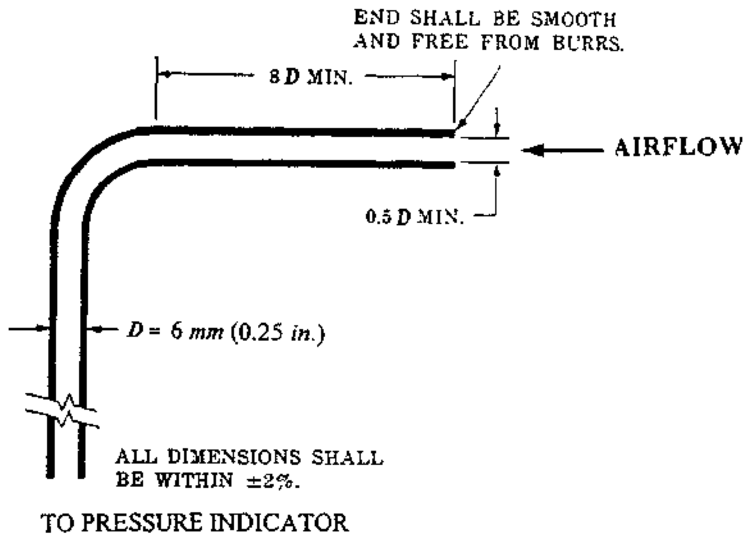
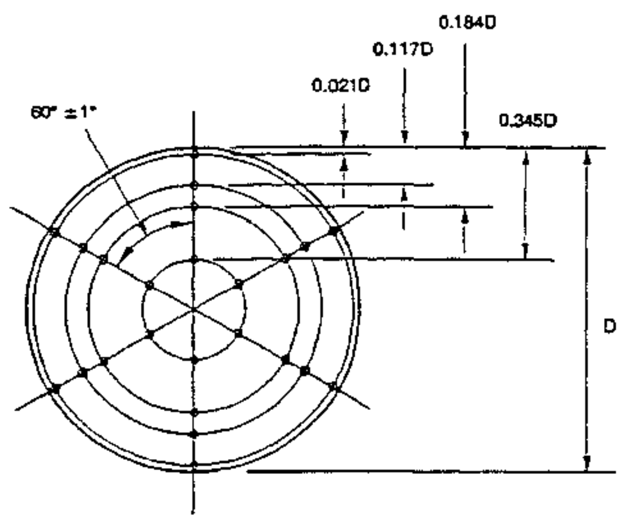
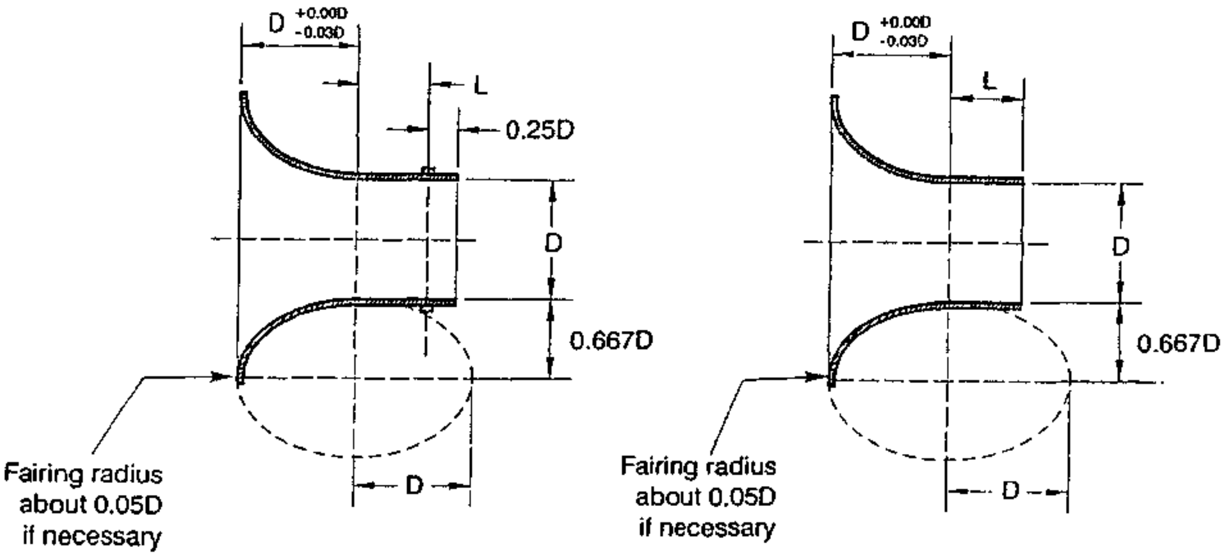


Figure 2B Total Pressure Tube



1. *D* IS THE AVERAGE OF FOUR MEASUREMENTS AT TRAVERSE PLANE AT 45° ANGLES MEASURED TO ACCURACY OF 0.2% *D*.
2. TRAVERSE DUCT SHALL BE ROUND WITHIN 0.5% *D* AT TRAVERSE PLANE AND FOR A DISTANCE OF 0.5 *D* ON EITHER SIDE OF TRAVERSE PLANE.
3. ALL PITOT POSITIONS ±0.0025d RELATIVE TO INSIDE DUCT WALLS.

Figure 3 Traverse Points in a Round Duct



NOZZLE WITH THROAT TAPS NOZZLE WITHOUT THROAT TAPS

NOTES

1. The nozzle shall have a cross-section consisting of elliptical and cylindrical portions, as shown. The cylindrical portion is defined as the nozzle throat.
2. The cross-section of the elliptical portion is one quarter of an ellipse, having the large axis D and the small axis 0.667 D. A three-radii approximation to the elliptical form that does not differ at any point in the normal direction more than 1.5% from the elliptical form shall be used. The adjacent arcs, as well as the last arc, shall smoothly meet and blend with the nozzle throat. The recommended approximation which meets these requirements is shown in Figure 4B by Cermak, J., Memorandum Report to AMCA 210/ASHRAE 51P Committee, June 16, 1992.
3. The nozzle throat dimension L shall be either $0.6D \pm 0.005D$ (recommended), or $0.5D \pm 0.005D$.
4. The nozzle throat dimension D shall be measured in situ to an accuracy of $0.001D$ at the throat entrance (at a distance L from the nozzle exit towards the nozzle inlet, and at the nozzle exit. At each of four locations 45° ± 2° apart, the measured throat diameter shall be up to $0.002D$ greater but not less than the mean diameter at the nozzle exit.
5. The nozzle surface in the direction of flow from the nozzle inlet towards the nozzle exit shall fair smoothly so that a straight-edge may be rocked over the surface without clicking. The macro-pattern of the surface shall not exceed $0.001D$, peak-to-peak. The edge of the nozzle exit shall be square, sharp, and free of burrs, nicks or roundings.
6. In a chamber, the use of either of the nozzle types shown above is permitted. A nozzle with throat taps shall be used when the discharge is direct into a duct, and the nozzle outlet should be flanged.
7. A nozzle with throat taps shall have four such taps conforming to Figure 2A, located 90° ± 2°apart. All four taps shall be connected to a piezometer ring.

Figure 4A Nozzles

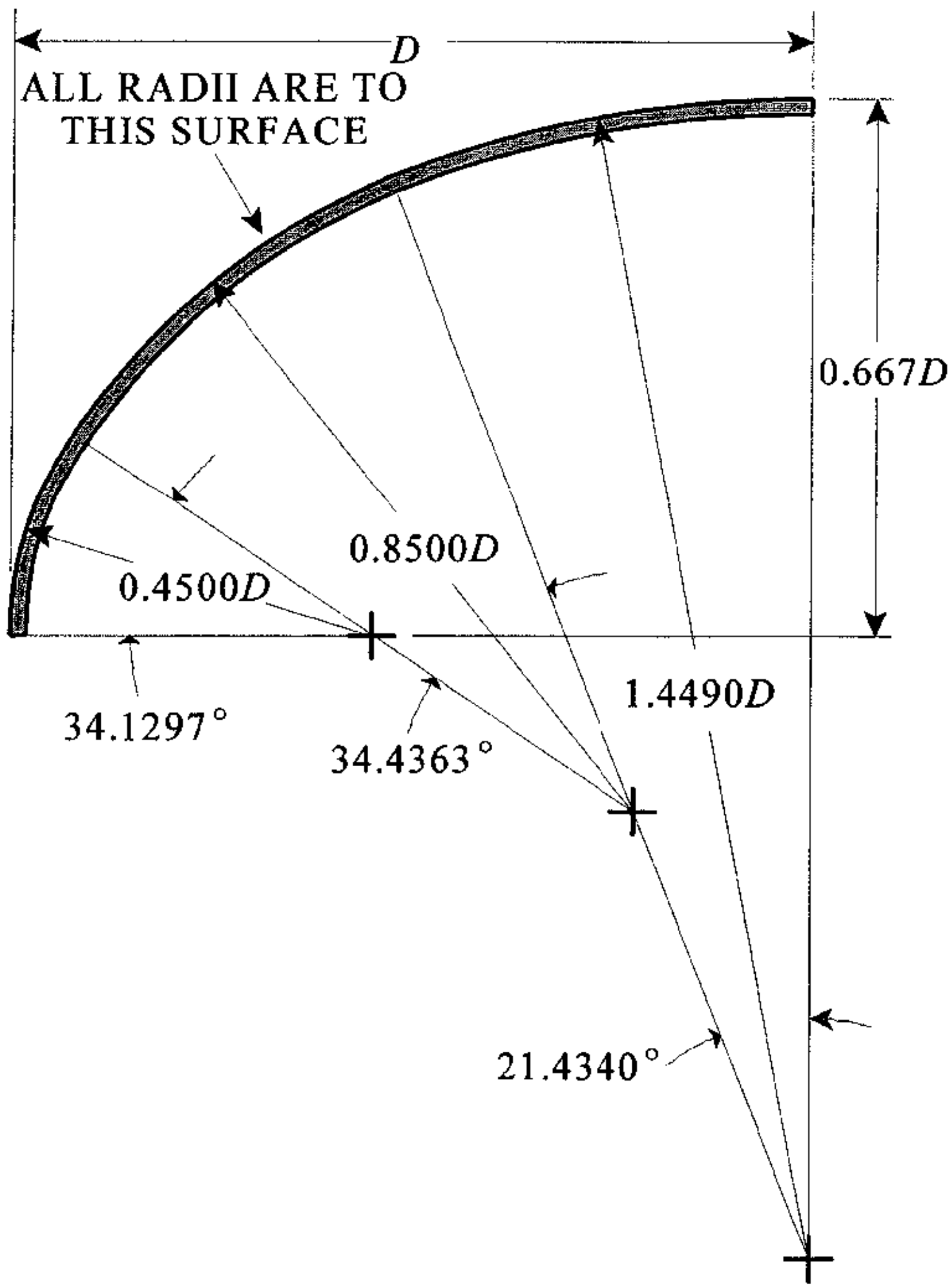


Figure 4B Three Arc Approximation of Elliptical Nozzle

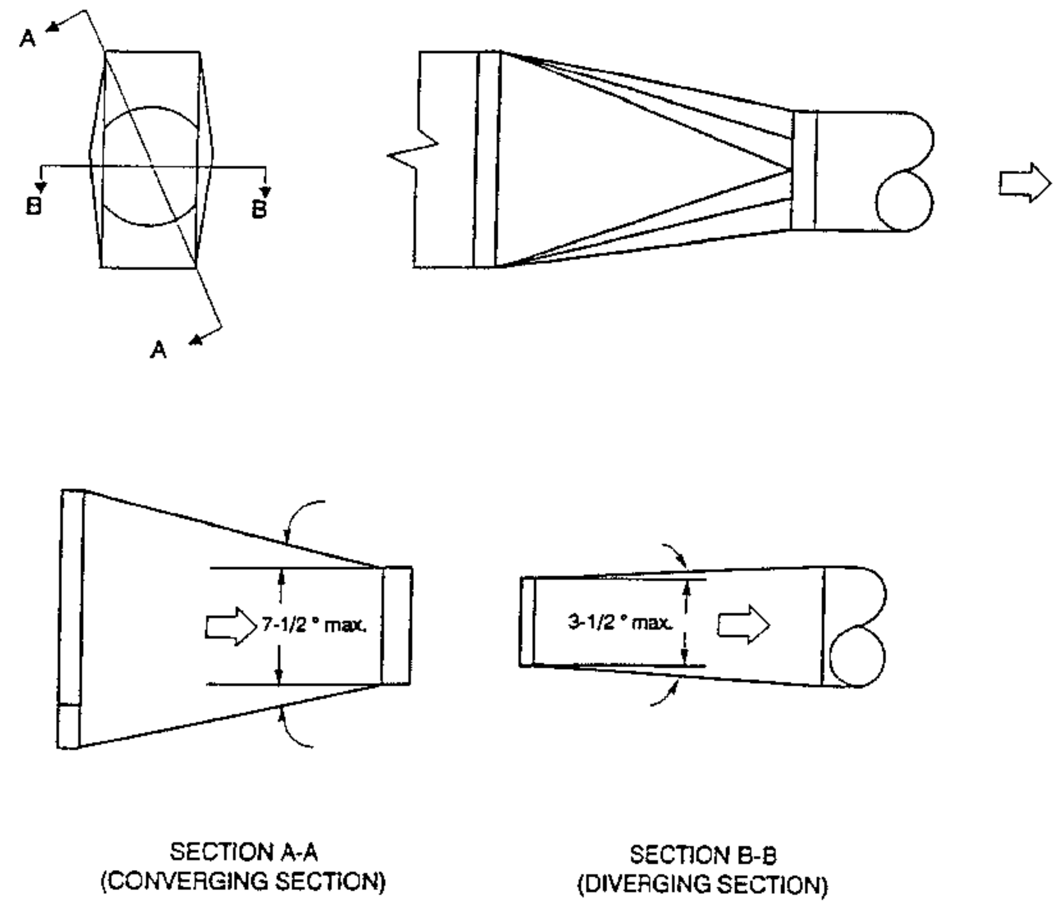
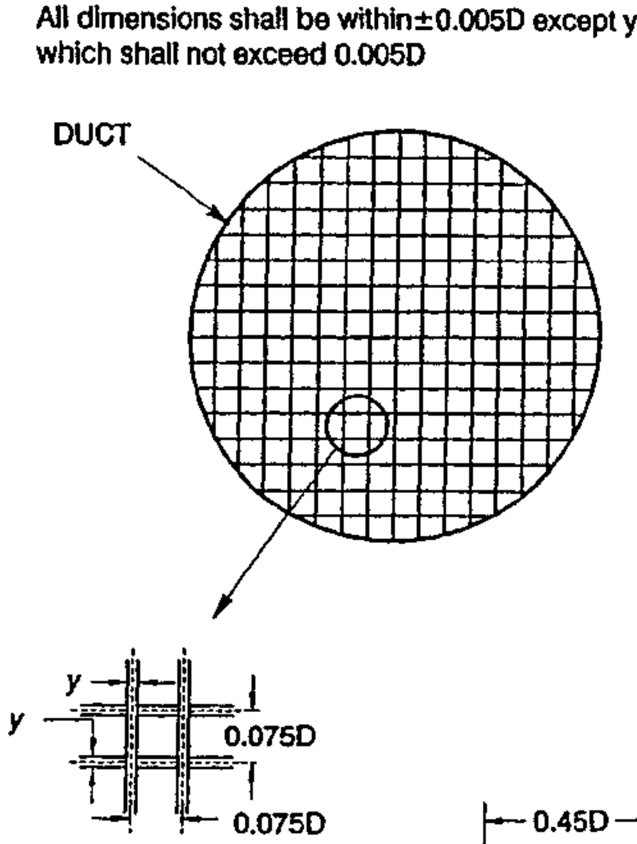


Figure 5 Transformation Piece



NOTE: Cell sides shall be flat and straight. Where y ≥ 3 *mm* (0.125 *in.*), the leading edge of each segment shall have a chamfer of 1.3 *mm* (0.05 *in.*) per side. The method of joining cell segments (such as tack welds) shall be kept to the minimum required for mechanical integrity and shall result in minimum protrusion into the fluid stream.

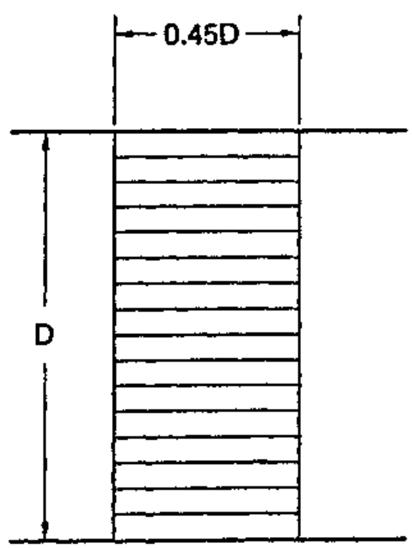
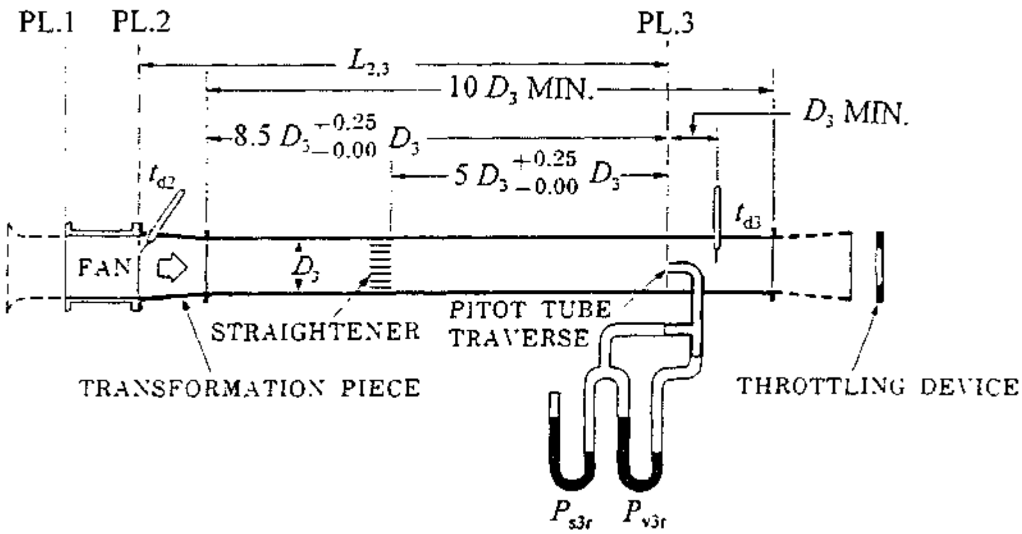


Figure 6 Flow Straightener



NOTES

1. Dotted lines on fan inlet indicate an inlet bell and one equivalent duct diameter which may be used for inlet duct simulation. The duct friction shall not be considered.
2. Dotted lines on the outlet indicate a diffuser cone which may be used to approach more nearly free delivery.

FLOW AND PRESSURE FORMULAE

$$P_{v3} = \left(\frac{\Sigma\sqrt{P_{v3r}}}{n}\right)^2$$

$$*V_3 = \sqrt{2}\sqrt{\frac{P_{v3}}{\rho_3}}$$

$$Q_3 = V_3 A_3$$

$$Q = Q_3\left(\frac{\rho_3}{\rho}\right)$$

$$P_{s3} = \frac{\Sigma P_{s3r}}{n}$$

$$P_v = P_{v3}\left(\frac{A_3}{A_2}\right)^2\left(\frac{\rho_3}{\rho_2}\right)$$

$$P_{t1} = 0$$

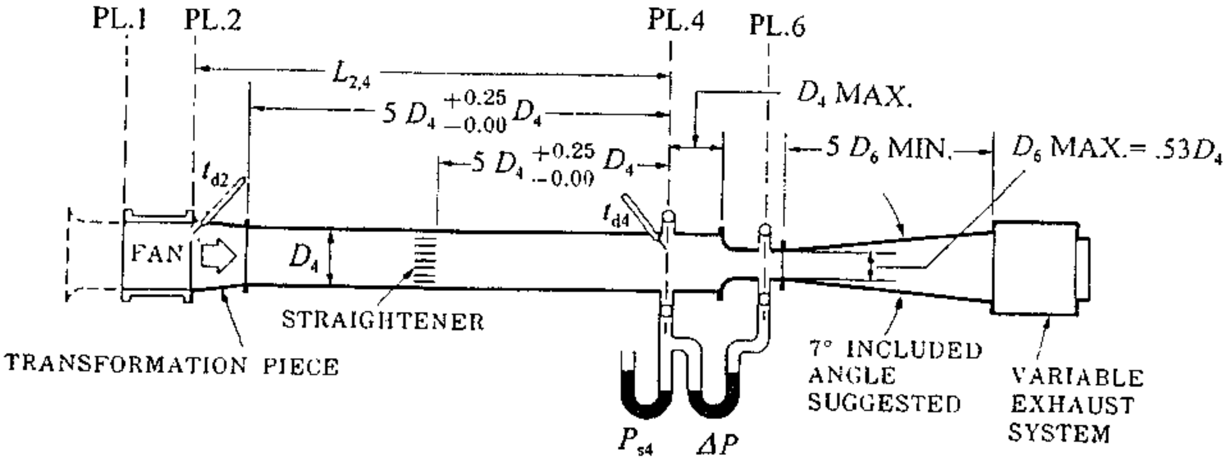
$$P_{t2} = P_{s3} + P_{v3} + f\left(\frac{L_{2,3}}{D_{h3}} + \frac{L_e}{D_{h3}}\right)P_{v3}$$

$$P_t = P_{t2} - P_{t1}$$

$$P_s = P_t - P_v$$

*The formulae given above are the same in both the SI and the I-P systems, except for V_3: In the I-P version, the constant $\sqrt{2}$ is replaced with the value 1097.

Figure 7 Outlet Duct Setup-Pitot Traverse in Outlet Duct



NOTES

1. Dotted lines on fan inlet indicate an inlet bell and one equivalent duct diameter which may be used for inlet duct simulation. The duct friction shall not be considered.
2. This figure may terminate at Plane 6 and interchangeable nozzles may be employed. In this case $\Delta P = P_{s4}$.
3. Variable exhaust system may be an auxiliary fan or a throttling device.

FLOW AND PRESSURE FORMULAE

$$*Q_4 = \frac{\sqrt{2}\ CA_6 Y \sqrt{\Delta P / \rho_4}}{\sqrt{1 - E\beta^4}}$$

$$Q = Q_4 \left(\frac{\rho_4}{\rho} \right)$$

$$V_4 = \frac{Q_4}{A_4}$$

$$*P_{v4} = \left(\frac{V_4}{\sqrt{2}} \right)^2 \rho_4$$

$$P_v = P_{v4} \left(\frac{A_4}{A_2} \right)^2 \left(\frac{\rho_4}{\rho_2} \right)$$

$$P_{t1} = 0$$

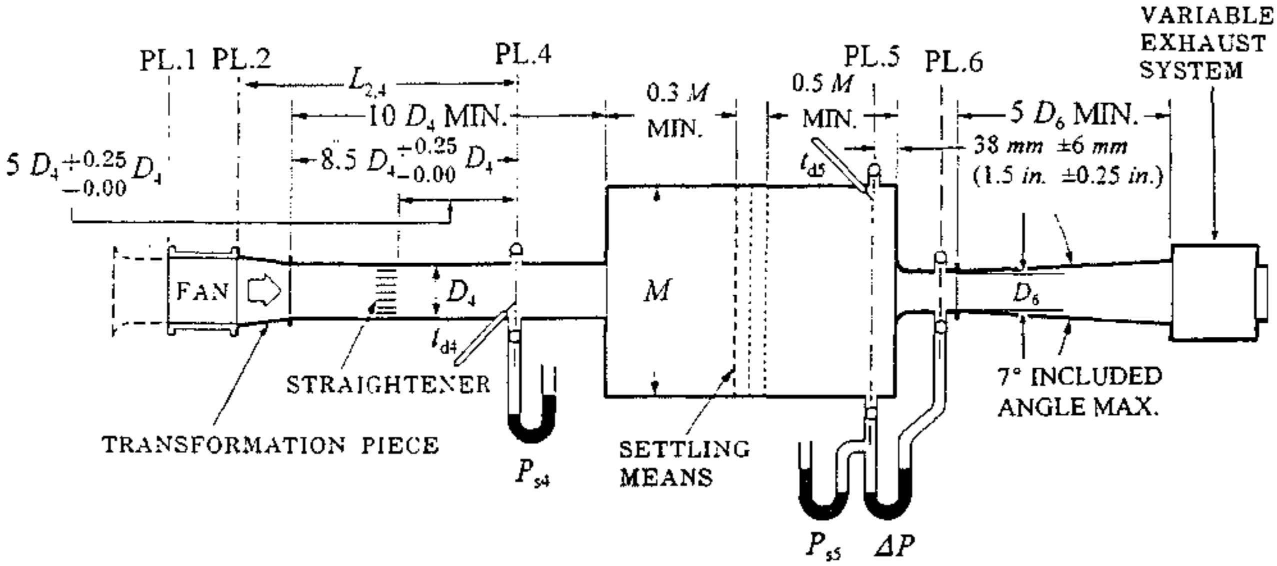
$$P_{t2} = P_{s4} + P_{v4} + f \left(\frac{L_{2,4}}{D_{h4}} + \frac{L_e}{D_{h4}} \right) P_{v4}$$

$$P_t = P_{t2} - P_{t1}$$

$$P_s = P_t - P_v$$

*The formulae given above are the same in both the SI and the I-P systems, except for Q_4 and P_{v4}: In the I-P version, the constant $\sqrt{2}$ is replaced with the value 1097.

Figure 8 Outlet Duct Setup-Nozzle on End of Outlet Duct



NOTES

1. Dotted lines on fan inlet indicate an inlet bell and one equivalent duct diameter which may be used for inlet duct simulation. The duct friction shall not be considered.
2. Additional ductwork of any size including elbows may be used to connect between the chamber and the exit of the 10 D minimum test duct.
3. Variable exhaust system may be an auxiliary fan or a throttling device.
4. Minimum M is determined by the requirements of 6.3.1 for this figure.

FLOW AND PRESSURE FORMULAE

$$*Q_5 = \sqrt{2}\ CA_6 Y \sqrt{\Delta P / \rho_5}$$

$$Q = Q_5 \left(\frac{\rho_5}{\rho} \right)$$

$$V_4 = \left(\frac{Q}{A_4} \right) \left(\frac{\rho}{\rho_4} \right)$$

$$*P_{v4} = \left(\frac{V_4}{\sqrt{2}} \right)^2 \rho_4$$

$$P_v = P_{v4} \left(\frac{A_4}{A_2} \right) \left(\frac{\rho_4}{\rho_2} \right)$$

$$P_{t1} = 0$$

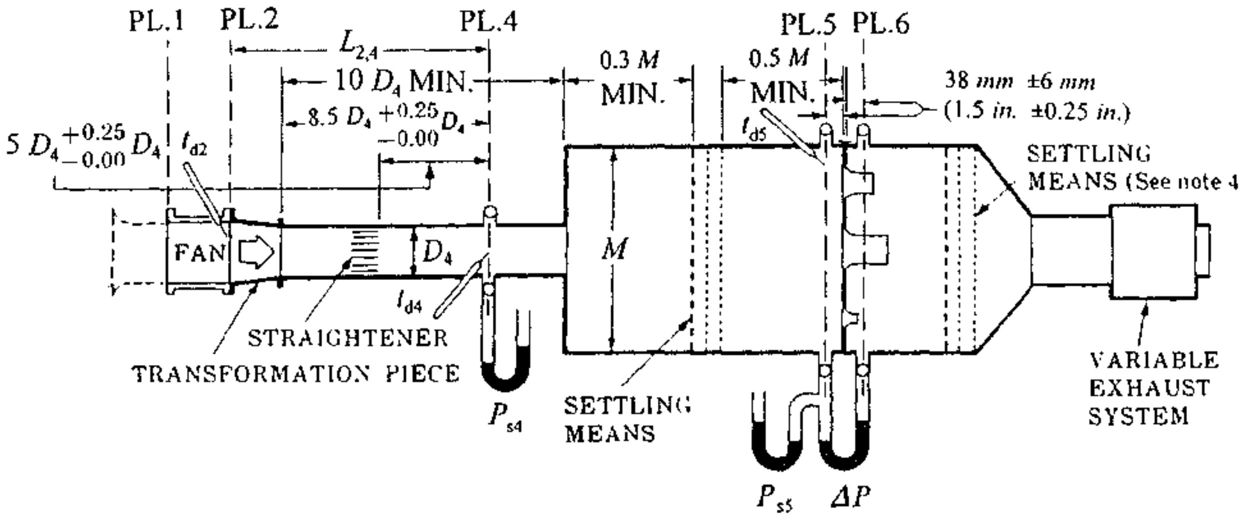
$$P_{t2} = P_{s4} + P_{v4} + f \left(\frac{L_{2,4}}{D_{h4}} + \frac{L_e}{D_{h4}} \right) P_{v4}$$

$$P_t = P_{t2} - P_{t1}$$

$$P_s = P_t - P_v$$

*The formulae given above are the same in both the SI and the I-P systems, except for Q_5 and P_{v4}: In the I-P version, the constant $\sqrt{2}$ is replaced with the value 1097.

Figure 9 Outlet Duct Setup-Nozzle on End of Chamber



NOTES

1. Dotted lines on fan inlet indicate an inlet bell and one equivalent duct diameter which may be used for inlet duct simulation. The duct friction shall not be considered.
2. Additional ductwork of any size, including elbows, may be used to connect between the chamber and the exit of the 10 D minimum test duct.
3. Variable exhaust system may be an auxiliary fan or a throttling device.
4. The distance from the exit face of the largest nozzle to the downstream settling means shall be a minimum of 2.5 throat diameters of the largest nozzle.
5. Minimum M is determined by the requirements of 6.3.1 for this figure.

FLOW AND PRESSURE FORMULAE

$$*Q_5 = \sqrt{2}\, Y\sqrt{\Delta P / \rho_5}\ \Sigma\left(CA_6\right)$$

$$Q = Q_5\left(\frac{\rho_5}{\rho}\right)$$

$$V_4 = \left(\frac{Q}{A_4}\right)\left(\frac{\rho}{\rho_4}\right)$$

$$*P_{v4} = \left(\frac{V_4}{\sqrt{2}}\right)^2 \rho_4$$

$$P_v = P_{v4}\left(\frac{A_4}{A_2}\right)^2\left(\frac{\rho_4}{\rho_2}\right)$$

$$P_{t1} = 0$$

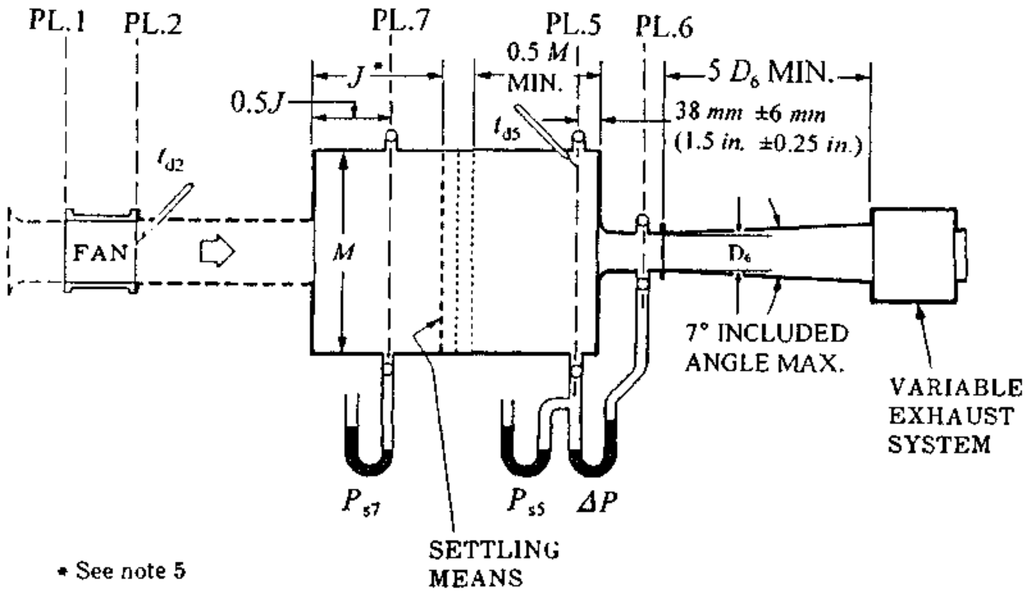
$$P_{t2} = P_{s4} + P_{v4} + f\left(\frac{L_{2,4}}{D_{h4}} + \frac{L_e}{D_{h4}}\right) P_{v4}$$

$$P_t = P_{t2} - P_{t1}$$

$$P_s = P_t - P_v$$

*The formulae given above are the same in both the SI and the I-P systems, except for Q_5 and P_{v4}: In the I-P version, the constant $\sqrt{2}$ is replaced with the value 1097.

Figure 10 Outlet Duct Setup-Multiple Nozzles in Chamber



NOTES

1. Dotted lines on fan inlet indicate an inlet bell and one equivalent duct diameter which may be used for inlet duct simulation. The duct friction shall not be considered.
2. Dotted lines on fan outlet indicate a uniform duct 2 to 3 equivalent diameters long and of an area within ± 1 % of the fan outlet area and a shape to fit the fan outlet. This may be used to simulate an outlet duct. The outlet duct friction shall not be considered.
3. The fan may be tested without outlet duct in which case it shall be mounted on the end of the chamber.
4. Variable exhaust system may be an auxiliary fan or a throttling device.
5. Dimension J shall be at least 1.0 times the fan equivalent discharge diameter for fans with axis of rotation perpendicular to the discharge flow and at least 2.0 times the fan equivalent discharge diameter for fans with axis of rotation parallel to the discharge flow.
6. Temperature t_{d2} may be considered equal to t_{d5}.
7. For the purpose of calculating the density at Plane 5 only, P_{s5} may be considered equal to P_{s7}.

FLOW AND PRESSURE FORMULAE

$$*Q_5 = \sqrt{2}\, C A_6 Y \sqrt{\Delta P / \rho_5}$$

$$Q = Q_5 \left(\frac{\rho_5}{\rho}\right)$$

$$V_2 = \left(\frac{Q}{A_2}\right)\left(\frac{\rho}{\rho_2}\right)$$

$$*P_{v2} = \left(\frac{V_2}{\sqrt{2}}\right)^2 \rho_2$$

$$P_v = P_{v2}$$

$$P_{t1} = 0$$

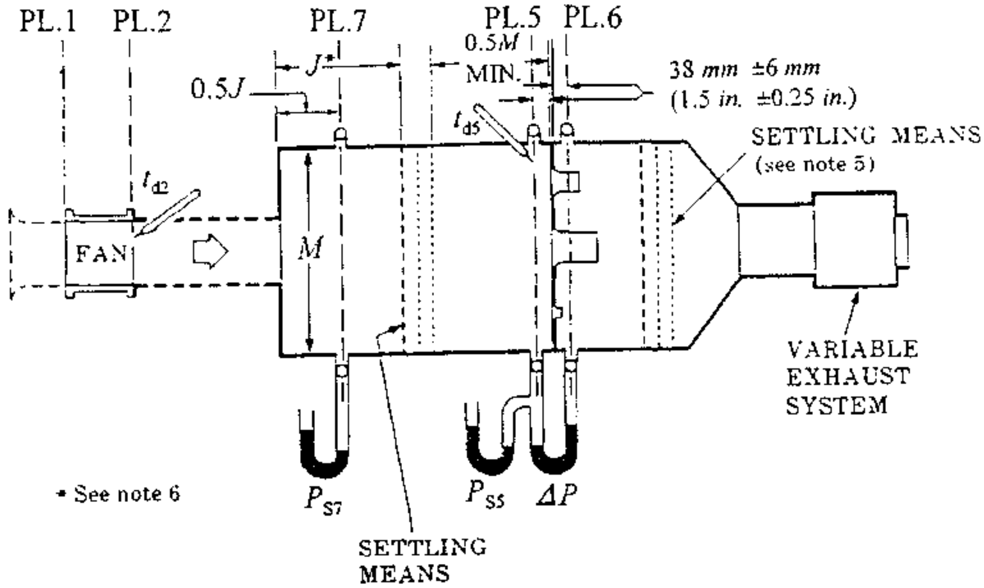
$$P_{t2} = P_{s7} + P_v$$

$$P_t = P_{t2} - P_{t1}$$

$$P_s = P_t - P_v$$

*The formulae given above are the same in both the SI and the I-P systems, except for Q_5 and P_{v2}: In the I-P version, the constant $\sqrt{2}$ is replaced with the value 1097.

Figure 11 Outlet Chamber Setup-Nozzle on End of Chamber



NOTES

1. Dotted lines on fan inlet indicate an inlet bell and one equivalent duct diameter which may be used for inlet duct simulation. The duct friction shall not be considered.
2. Dotted lines on fan outlet indicate a uniform duct 2 to 3 equivalent diameters long and of an area within ± 1 % of the fan outlet area and a shape to fit the fan outlet. This may be used to simulate an outlet duct. The outlet duct friction shall not be considered.
3. The fan may be tested without outlet duct in which case it shall be mounted on the end of the chamber.
4. Variable exhaust system may be an auxiliary fan or a throttling device.
5. The distance from the exit face of the largest nozzle to the downstream settling means shall be a minimum of 2.5 throat diameters of the largest nozzle.
6. Dimension J shall be at least 1.0 times the fan equivalent discharge diameter for fans with axis of rotation perpendicular to the discharge flow and at least 2.0 times the fan equivalent discharge diameter for fans with axis of rotation parallel to the discharge flow.
7. Temperature t_{d2} may be considered equal to t_{d5}.
8. For the purpose of calculating the density at Plane 5 only, P_{s5} may be considered equal to P_{s7}.

FLOW AND PRESSURE FORMULAE

$$*Q_5 = \sqrt{2}\ Y\sqrt{\Delta P / \rho_5}\ \Sigma\left(CA_6\right)$$

$$P_v = P_{v2}$$

$$Q = Q_5\left(\frac{\rho_5}{\rho}\right)$$

$$P_{t1} = 0$$

$$V_2 = \left(\frac{Q}{A_2}\right)\left(\frac{\rho}{\rho_2}\right)$$

$$P_{t2} = P_{s7} + P_v$$

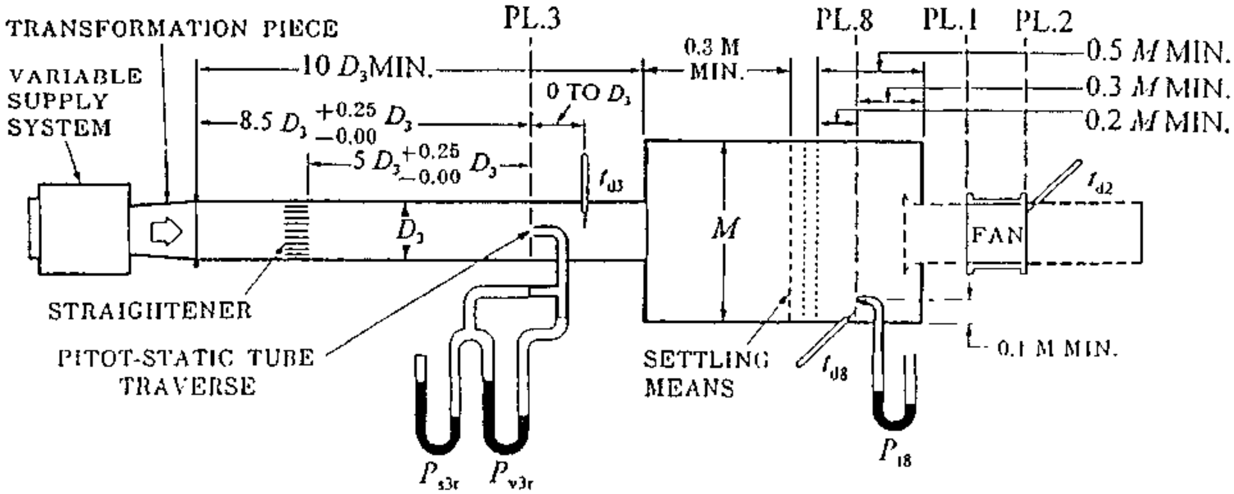
$$P_t = P_{t2} - P_{t1}$$

$$*P_{v2} = \left(\frac{V_2}{\sqrt{2}}\right)^2 \rho_2$$

$$P_s = P_t - P_v$$

*The formulae given above are the same in both the SI and the I-P systems, except for Q_5 and P_{v2}: In the I-P version, the constant $\sqrt{2}$ is replaced with the value 1097.

Figure 12 Outlet Chamber Setup-Multiple Nozzles in Chamber



NOTES

1. Dotted lines on fan inlet indicate an inlet bell and one equivalent duct diameter which may be used for inlet duct simulation. The duct friction shall not be considered.
2. Dotted lines on fan outlet indicate a uniform duct 2 or 3 equivalent diameters long and of an area within ± 1% of the fan outlet area and a shape to fit the fan outlet. This may be used to simulate an outlet duct. The outlet duct friction shall not be considered.
3. Additional ductwork of any size including elbows may be used to connect between the chamber and the exit of the 10 D minimum test duct.
4. Variable supply system may be an auxiliary fan or a throttling device.

FLOW AND PRESSURE FORMULAE

$$P_{v3} = \left(\frac{\Sigma\sqrt{P_{v3r}}}{n}\right)^2$$

$$P_v = P_{v3}\left(\frac{A_3}{A_2}\right)^2\left(\frac{\rho_3}{\rho_2}\right)$$

$$*V_3 = \sqrt{2}\sqrt{\frac{P_{v3}}{\rho_3}}$$

$$P_{t1} = P_{t8}$$

$$Q_3 = V_3 A_3$$

$$P_{t2} = P_v$$

$$Q = Q_3\left(\frac{\rho_3}{\rho}\right)$$

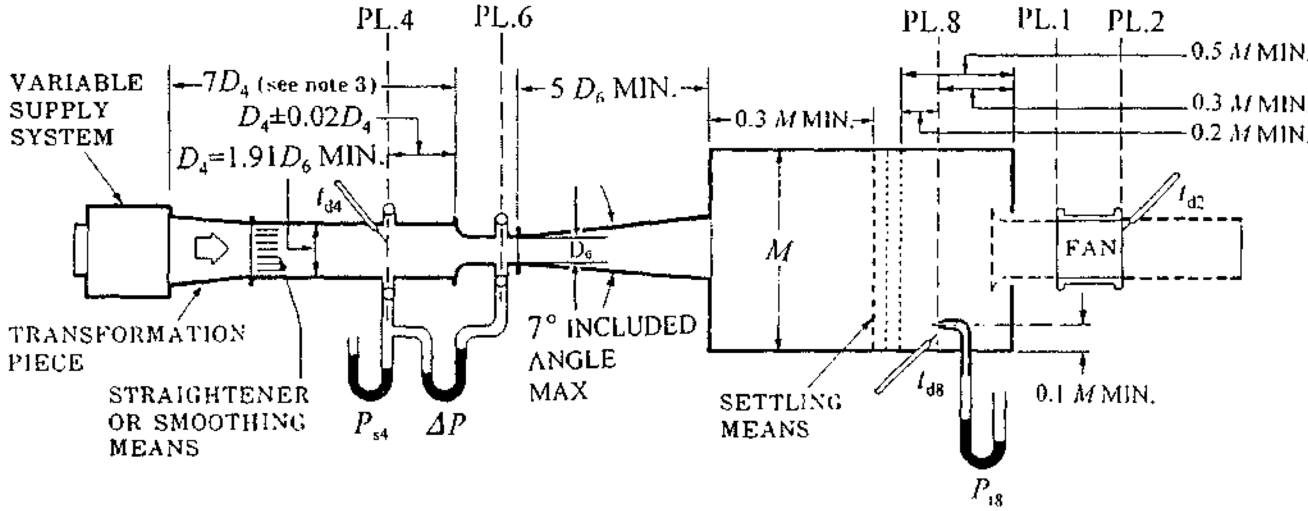
$$P_t = P_{t2} - P_{t1}$$

$$P_{s3} = \frac{\Sigma P_{s3r}}{n}$$

$$P_s = P_t - P_v$$

*The formulae given above are the same in both the SI and the I-P systems, except for V_3: In the I-P version, the constant $\sqrt{2}$ is replaced with the value 1097.

Figure 13 Inlet Chamber Setup-Pitot Traverse in Duct



NOTES

1. Dotted lines on fan inlet indicate an inlet bell and one equivalent duct diameter which may be used for inlet duct simulation. The duct friction shall not be considered.
2. Dotted lines on fan outlet indicate a uniform duct 2 to 3 equivalent diameters long and of an area within ± 1% of the fan outlet area and a shape to fit the fan outlet. This may be used to simulate an outlet duct. The outlet duct friction shall not be considered.
3. Duct length 7 D_4 may be shortened to not less than 2 D_4 when it can be demonstrated, by a traverse of D_4 by Pitot-static tube located a distance D_4 upstream from the nozzle entrance or downstream from the straightener or smoothing means, that the energy ratio E is less than 1.1 when the velocity is greater than (6.1 *m/s*) (1200 *fpm*). Smoothing means such as screens, perforated plates, or other media may be used.
4. Variable supply system may be an auxiliary fan or a throttling device. One or more supply systems, each with its own nozzle, may be used.

FLOW AND PRESSURE FORMULAE

$$*Q_4 = \frac{\sqrt{2}\; C A_6 Y \sqrt{\Delta P / \rho_4}}{\sqrt{1 - E\beta^4}}$$

$$P_v = P_{v2}$$

$$Q = Q_4 \left(\frac{\rho_4}{\rho} \right)$$

$$P_{t1} = P_{t8}$$

$$P_{t2} = P_v$$

$$V_2 = \left(\frac{Q}{A_2} \right) \left(\frac{\rho}{\rho_2} \right)$$

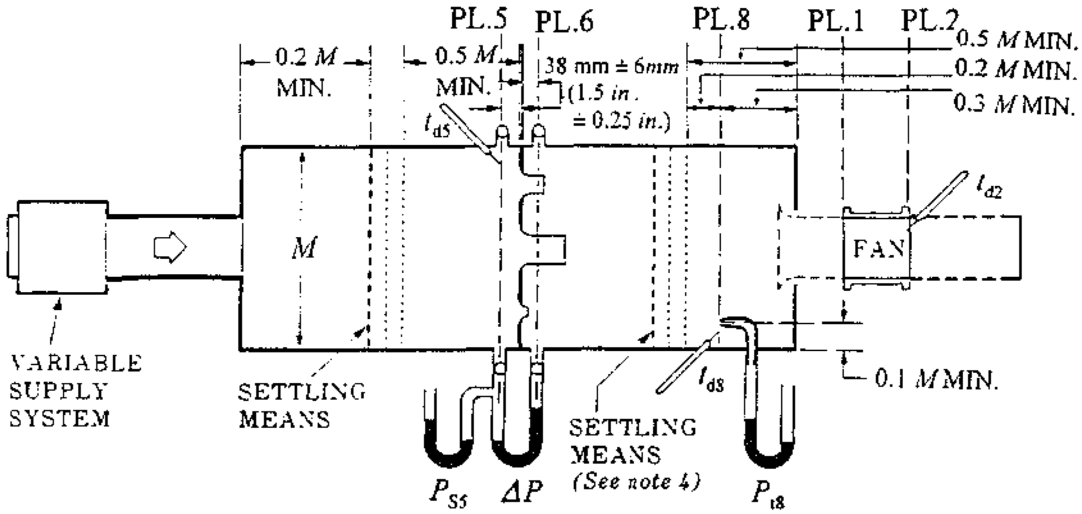
$$P_t = P_{t2} - P_{t1}$$

$$*P_{v2} = \left(\frac{V_2}{\sqrt{2}} \right)^2 \rho_2$$

$$P_s = P_t - P_v$$

*The formulae given above are the same in both the SI and the I-P systems, except for Q_4 and P_{v2}: In the I-P version, the constant $\sqrt{2}$ is replaced with the value 1097.

Figure 14 Inlet Chamber Setup-Ducted Nozzle on Chamber



NOTES

1. Dotted lines on fan inlet indicate an inlet bell and one equivalent duct diameter which may be used for inlet duct simulation. The duct friction shall not be considered.
2. Dotted lines on fan outlet indicate a uniform duct 2 to 3 equivalent diameters long and of an area within ± 1% of the fan outlet area and a shape to fit the fan outlet. This may be used to simulate an outlet duct. The outlet duct friction shall not be considered.
3. Variable supply system may be an auxiliary fan or throttling device.
4. The distance from the exit face of the largest nozzle to the downstream settling means shall be a minimum of 2.5 throat diameters of the largest nozzle.
5. For the purpose of calculating the density at Plane 5 only, P_{s5} may be considered equal to $(P_{t8} + \Delta P)$

FLOW AND PRESSURE FORMULAE

$$*Q_5 = \sqrt{2}\, Y \sqrt{\Delta P / \rho_5}\, \Sigma\left(CA_6\right)$$

$$Q = Q_5 \left(\frac{\rho_5}{\rho}\right)$$

$$V_2 = \left(\frac{Q}{A_2}\right)\left(\frac{\rho}{\rho_2}\right)$$

$$*P_{v2} = \left(\frac{V_2}{\sqrt{2}}\right)^2 \rho_2$$

$$P_v = P_{v2}$$

$$P_{t1} = P_{t8}$$

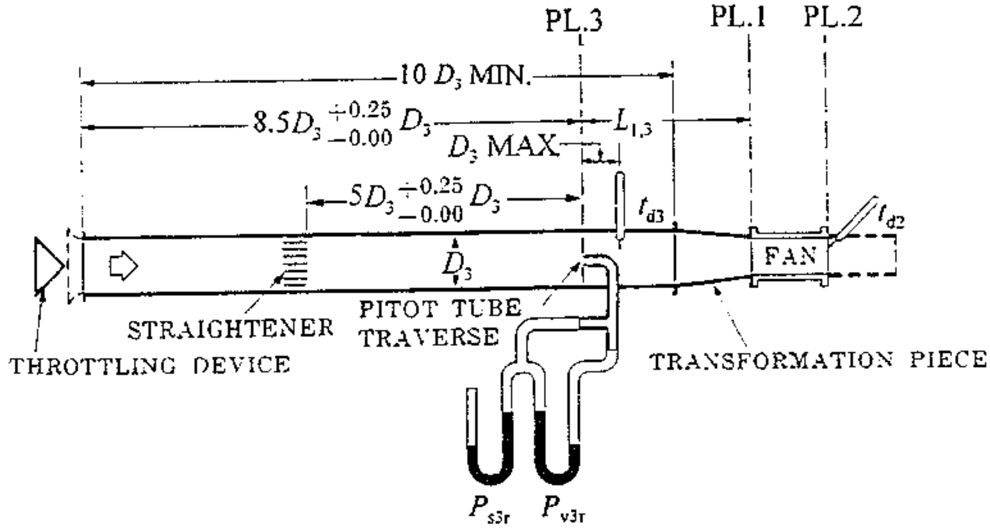
$$P_{t2} = P_v$$

$$P_t = P_{t2} - P_{t1}$$

$$P_s = P_t - P_v$$

*The formulae given above are the same in both the SI and the I-P systems, except for Q_5 and P_{v2}: In the I-P version, the constant $\sqrt{2}$ is replaced with the value 1097.

Figure 15 Inlet Chamber Setup-Multiple Nozzles in Chamber



NOTES

1. Dotted lines on inlet indicate an inlet bell which may be used to approach more nearly free delivery.
2. Dotted lines on fan outlet indicate a uniform duct 2 to 3 equivalent diameters long and of an area within ± 1.0% of the fan outlet area and a shape to fit the fan outlet. This may be used to simulate an outlet duct. The outlet duct friction shall not be considered.

FLOW AND PRESSURE FORMULAE

$$P_{v3} = \left(\frac{\Sigma \sqrt{P_{v3r}}}{n} \right)^2$$

$$*V_3 = \sqrt{2} \sqrt{\frac{P_{v3}}{\rho_3}}$$

$$Q_3 = V_3 A_3$$

$$Q = Q_3 \left(\frac{\rho_3}{\rho} \right)$$

$$P_{s3} = \frac{\Sigma P_{s3r}}{n}$$

$$P_v = P_{v3} \left(\frac{A_3}{A_2} \right)^2 \left(\frac{\rho_2}{\rho_3} \right)$$

$$P_{t1} = P_{s3} + P_{v3} - f \left(\frac{L_{1,3}}{D_{h3}} \right) P_{v3}$$

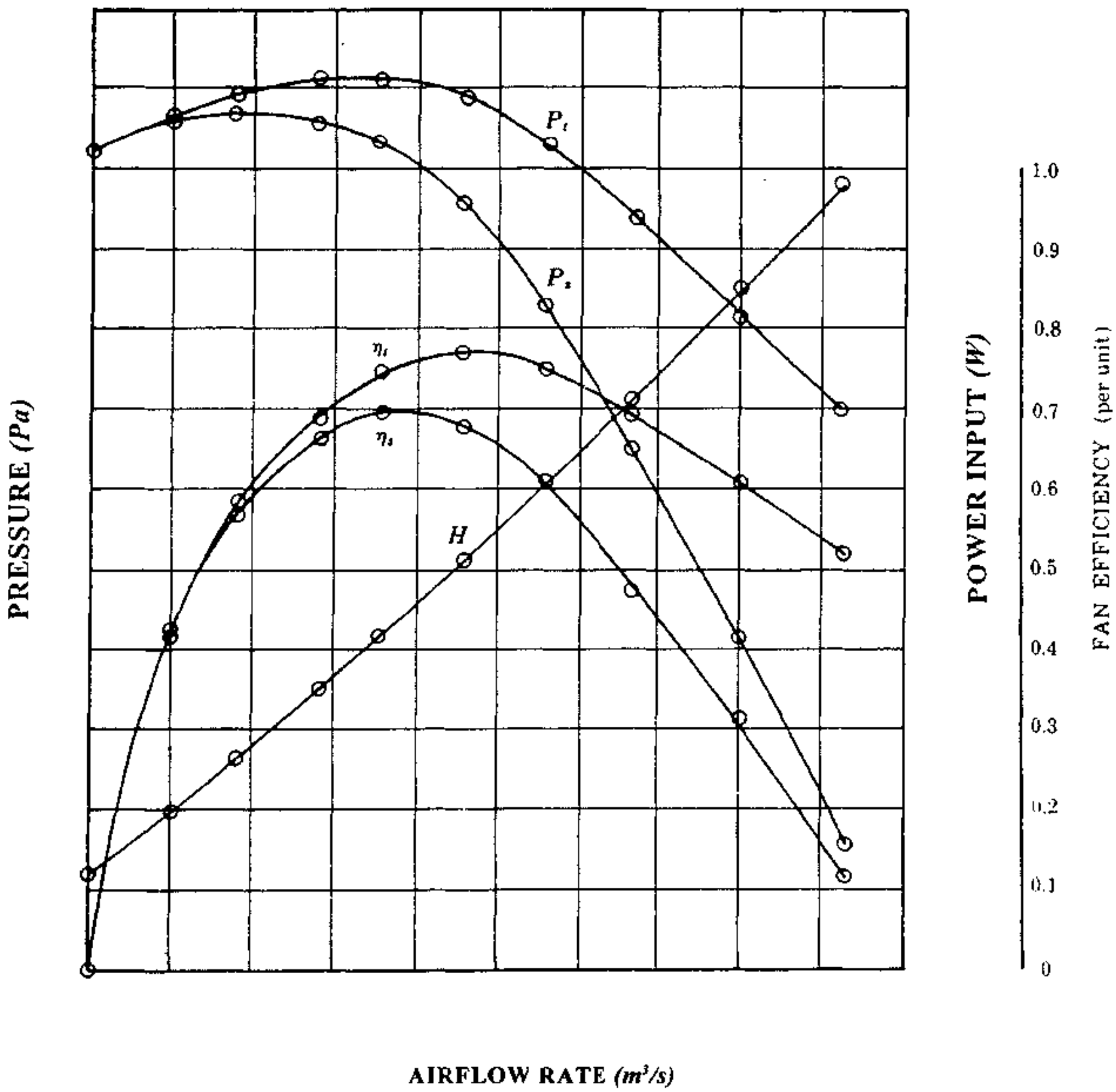
$$P_{t2} = P_v$$

$$P_t = P_{t2} - P_{t1}$$

$$P_s = P_t - P_v$$

*The formulae given above are the same in both the SI and the I-P systems, except for V_3: In the I-P version, the constant $\sqrt{2}$ is replaced with the value 1097.

Figure 16 Inlet Duct Setup-Pitot Traverse in Inlet Duct



FAN__________ INSTALLATION TYPE__________ FAN SPEED__________ RPM

IMPELLER TIP DIAMETER (NOMINAL)__________

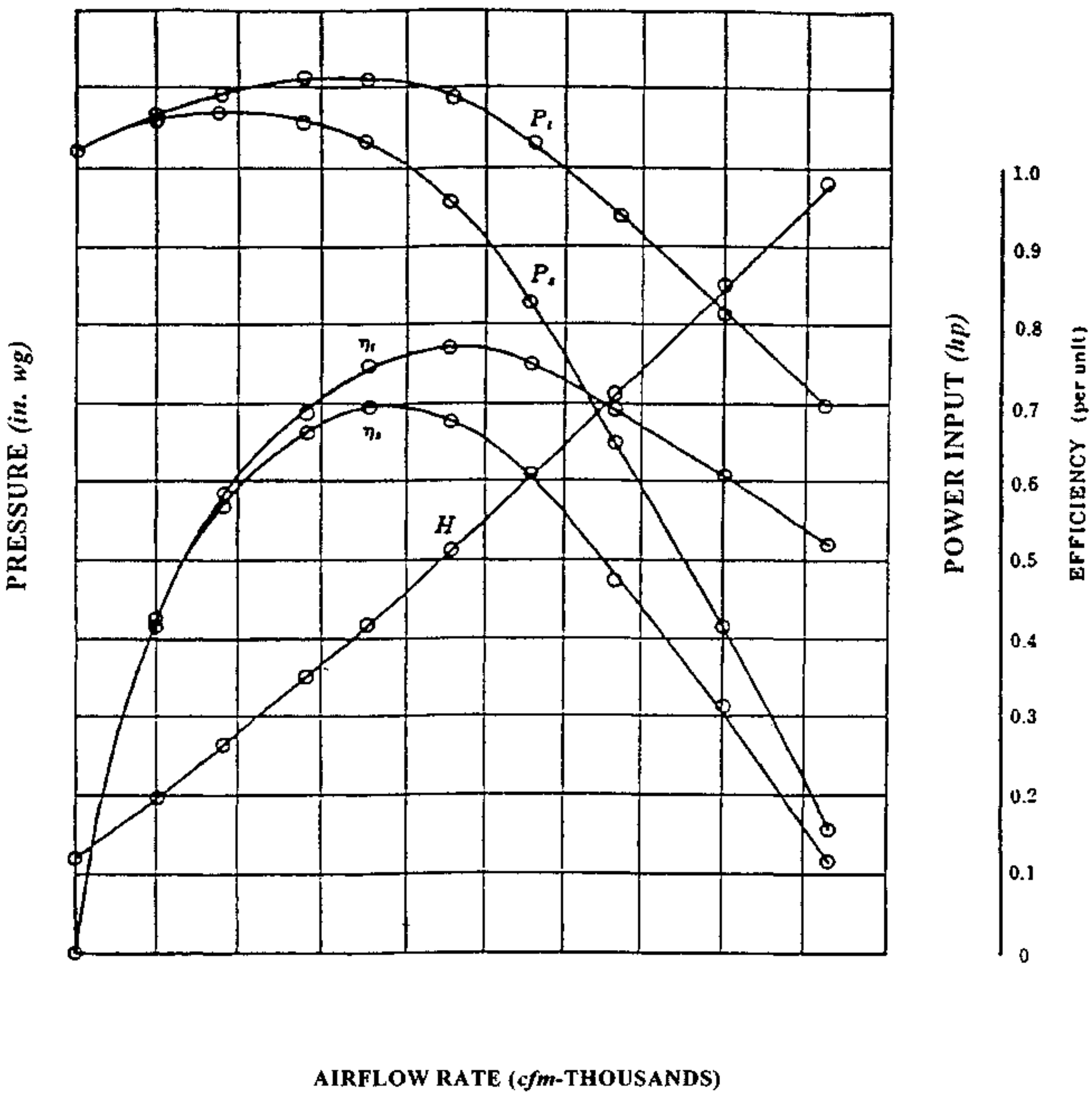
FAN AIR DENSITY__________ TEST SETUP__________

BAROMETRIC PRESSURE__________ FIGURE__________

TEST NUMBER__________ CURVE BY__________

LABORATORY__________ DATE__________

Figure 17 Typical Fan Performance Curve, SI



FAN__________ INSTALLATION TYPE__________FAN SPEED__________RPM

IMPELLER TIP DIAMETER (NOMINAL)__________

FAN AIR DENSITY__________TEST SETUP__________

BAROMETRIC PRESSURE__________FIGURE__________

TEST NUMBER__________CURVE BY__________

LABORATORY__________DATE__________

Figure 17 Typical Fan Performance Curve, I-P

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ ASHRAE Standard 51 but is included for information purposes.

APPENDIX A. Units of Measurement

A.1 System of Units. SI units (The International System of Units - Le Systéme International d'Unités)[24] are the primary units employed in this standard, with I-P units given as the secondary reference. SI units are based on the fundamental values of the International Bureau of Weights and Measures [25], and I-P values are based on the values of the National Institute of Standards and Technology which are, in turn, based on the values of the International Bureau. Appendix B provides conversion factors and coefficients for SI and other metric systems.

A.2 Basic Units. The unit of length is the meter *(m)* or millimeter *(mm)*; I-P units are the foot *(ft)* or inch *(in.)*. The unit of mass is the kilogram *(kg)*; the I-P unit is the pound-mass *(lbm)*. The unit of time is either the minute *(min)* or the second *(s)*. The unit of temperature is either the kelvin *(K)* or the degree Celsius *(°C)*; I-P units are the degree Rankine *(°R)* or the degree Fahrenheit *(°F)*. The unit of force is the newton *(N)*; the I-P unit is the pound *(lbf)*.

A.3 Airflow Rate and Velocity. The unit of airflow rate is the cubic meter per second *(m^3/s)*; the I-P unit is the cubic foot per minute *(cfm)*. The unit of velocity is the meter per second *(m/s)*; the I-P unit is the foot per minute *(fpm)*.

A.4 Pressure. The unit of pressure is either the Pascal *(Pa)* or the millimeter of mercury *(mm Hg)*; the I-P unit is either the inch water gauge *(in. wg)*, or the inch mercury column *(in. Hg)*. Values in *mm Hg* or in *in. Hg* shall be used only for barometric pressure measurements. The inch water gauge shall be based on a 1 inch column of distilled water at 68°*F* under standard gravity and a gas column balancing effect based on standard air. The millimeter of mercury shall be based on a 1 *mm* column of mercury at 0°*C* under standard gravity in vacuo. The inch of mercury shall be based on a 1 inch column of mercury at 32°*F* under standard gravity in vacuo.

A.5 Power, Energy, and Torque. The unit of power is the watt *(W)*; the I-P unit is the horsepower *(hp)*. The unit of energy is the joule *(J)*; the I-P unit is the foot pound *(ft•lbf)*. The unit of torque is the newton-meter *(N•m)*; the I-P unit is the pound inch *(lbf•in.)*.

A.6 Efficiency. Efficiencies are expressed on a per unit basis. Percentage values can be obtained by multiplying by 100.

A.7 Speed. There is no unit of rotational speed as such in the SI system of units. The commonly used unit in both systems is the revolution per minute *(rpm)*.

A.8 Gas Properties. The unit of density is the kilogram per cubic meter *(kg/m^3)*; the I-P unit is the pound-mass per cubic foot *(lbm/ft^3)*. The unit of viscosity is the Pascal second *(Pa•s)*; the I-P unit is the pound-mass per foot-second *(lbm/ft•s)*. The unit of gas constant is the joule per kilogram kelvin *(J/(kg•K))*; the I-P unit is the foot pound per pound mass degree Rankine *(ft•lbf/lbm•°R)*.

A.9 Dimensionless Groups. Various dimensionless quantities appear in the text. Any consistent system of units may be employed to evaluate these quantities unless a numerical factor is included, in which case units must be as specified.

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ ASHRAE Standard 51 but is included for information purposes.

APPENDIX B. SI Conversions and Physical Constants

B.1 I-P Equivalents of SI Units [26].

Area

1 m^2	= 10.76 ft^2

Length

1 m (meter)	= 3.2808 ft

Mass

1 kg	= 2.2046 lbm

Temperature

1 K	= 1.8°R
t_c	= (t_F - 32)/1.8

Force

1 N	= 0.22481 lbf
1 kp (kilopond)	= 2.2046 lbf

Flow Rate

1 m^3/s	= 2118.9 cfm
1 m^3/h	= 0.58858 cfm

Velocity

1 m/s	= 196.85 fpm

Pressure

1 Pa (Pascal) at 20°C	= 0.0040264 $in.$ wg at 68°F
1 Pa (Pascal) at 0°C	= 0.00029530 $in.$ Hg at 32°F
1 Pa (Pascal) at 3.9°C	= 0.004015 $in.$ wg at 39°F

Power

1 W (watt)	= 0.0013410 hp

Energy

1 J (joule)	= 0.73756 $ft \cdot lbf$

Torque

1 $N \cdot m$	= 8.8507 $lbf \cdot in.$

Density

1 kg/m^3	= 0.062428 lbm/ft^3
1.200 kg/m^3 at 20°C	= 0.075 lbf/ft^3 at 68°F

Viscosity

1 $Pa \cdot s$	= 0.67197 $lbm/ft \cdot s$

Gas Constant

1 $J/(kg \cdot K)$	= 0.18586 $ft \cdot lbf/(lbm \cdot °R)$

Gravitational Acceleration

9.80665 m/s^2	= 32.174 ft/s^2

B.2 SI Equivalents of I-P Units [26].

Area

1 ft^2	= 0.0929 m^2

Length

1 ft	= 0.30480 m

Mass

1 lbm =	0.45359 kg

Temperature

1°R	= K/1.8
t_F	= 1.8 t_c + 32

Force

1 lbf	= 4.4482 N

Flow Rate

1 cfm	= 0.00047195 m^3/s
1 cfm	= 1.6990 m^3/hr

Velocity

1 fpm	= 0.0050800 m/s

Pressure

1 $in.$ wg at 68°F	= 248.36 Pa at 20 °C
1 $in.$ wg at 39 °F	= 249.1 Pa at 3.9 °C
1 $in.$ Hg at 32 °F	= 3386.4 Pa at 0 °C

Power

1 hp (horsepower)	= 0.74570 kW

Energy

1 ft lbf	= 1.3558 J

Torque

1 lbf $in.$	= 0.11298 $N \cdot m$

Density

1 lbm/ft^3	= 16.018 kg/m^3
0.075 lbm/ft^3 at 68 °F	= 1.2 kg/m^3 at 20°C

Viscosity

1 $lbm/ft \cdot s$	= 1.4882 $Pa \cdot s$

Gas Constant

1 $ft \cdot lbf/(lbm \cdot °R)$	= 5.3803 $J/(kg \cdot K)$

Gravitational Acceleration

32.174 ft/s^2	= 9.80665 m/s^2

B3. Physical Constants. The value of standard gravitational acceleration shall be taken as 9.80665 m/s^2 which corresponds to the value at mean sea level at 45° latitude; the I-P value is 32.1740 ft/s^2 [24]. The density of distilled water at saturation pressure shall be taken as 998.278 kg/m^3 at 20°C; the I-P value is 62.3205 lbm/ft^3 at 68°F [27]. The density of mercury at saturation pressure shall be taken as 13595.1 kg/m^3 at 0°C; the I-P value is 848.714 lbm/ft^3 at 32°F [27]. The specific weights in kg/m^3 (lbm/ft^3) of these fluids *in vacuo* under standard gravity are numerically equal to their densities at corresponding temperatures.

This Appendix is not part of ANSI/AMCA Standard 210 or ANSI/ ASHRAE Standard 51 but is included for information purposes.

APPENDIX C. Derivation of Equations

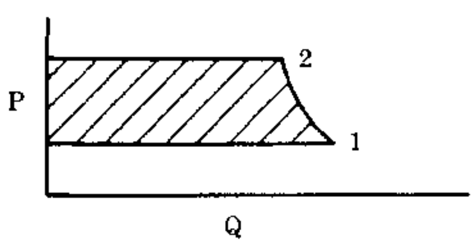
C.1 General. Various formulae appear in the standard. The origin of these formulae will be obvious to an engineer. Some, like the equations for α, β, P_t, P_s, and P_v, are algebraic expressions of fundamental definitions. Others, like the equations for p_e, μ, and C, are simply polynomials derived to fit the indicated data. Still others are derived from the equation of state, the Bernoulli equation, the equation of continuity, and other fundamental considerations. Only the less obvious formulae will be derived here.

C.2 Symbols. In the derivations which follow, certain symbols and notations are used in addition to those which are also used in the standard.

SYMBOL	DESCRIPTION	UNIT
H_i	Power Input to Impeller,	*W (hp)*
n	Polytropic Exponent	dimensionless
P	Absolute Total Pressure,	*Pa (in. wg)*

C.3 Fan Total Efficiency Equation. The values of the fan airflow rate, fan total pressure, and fan power input which are determined during a test are the compressible flow values for the fan speed and fan air density prevailing. A derivation of the fan total efficiency equation based on compressible flow values follows [23].

The process during compression may be plotted on a chart of absolute total pressure (P) versus airflow rate (Q). By using total pressure, all of the energy is accounted for including kinetic energy.



The fan power output (H_o) is proportional to the shaded area which leads to

$$H_o = \int_1^2 Q\,dP \qquad \text{Eq. C-1 SI}$$

$$H_o = \frac{1}{6362}\int_1^2 Q\,dP \qquad \text{Eq. C-1 I-P}$$

The compression process may be assumed to be polytropic for which, from thermodynamics,

$$Q = Q_1\left(\frac{P}{P_1}\right)^{-1/n} \qquad \text{Eq. C-2}$$

Substituting,

$$H_o = Q_1\int_1^2\left(\frac{P}{P_1}\right)^{-1/n} dP \qquad \text{Eq. C-3 SI}$$

$$H_o = \frac{Q_1}{6362}\int_1^2\left(\frac{P}{P_1}\right)^{-1/n} dP \qquad \text{Eq. C-3 I-P}$$

Integrating between limits,

$$H_o = Q_1 P_1\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right]$$

Eq. C-4 SI

$$H_o = \frac{Q_1 P_1}{6362}\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right]$$

Eq. C-4 I-P

But from the definition of fan total pressure (P_t),

$$P_1 = P_t / \left(\frac{P_2}{P_1} - 1\right) \qquad \text{Eq. C-5}$$

and the definition of fan total efficiency (η_t),

$$\eta_t = \frac{H_o}{H_i} \qquad \text{Eq. C-6}$$

it follows that

$$\eta_t = \frac{Q_1 P_t}{H_i}\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \Bigg/ \left(\frac{P_2}{P_1} - 1\right) \qquad \text{Eq. C-7 SI}$$

$$\eta_t = \frac{Q_1 P_t}{6362 H_i}\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \Bigg/ \left(\frac{P_2}{P_1} - 1\right) \qquad \text{Eq. C-7 I-P}$$

C.4 Compressibility Coefficient. The efficiency equation derived above can be rewritten

$$\eta_t = \frac{Q_1 P_t K_p}{H_i} \qquad \text{Eq. C-8 SI}$$

$$\eta_t = \frac{Q_1 P_t K_p}{6362\, H_i} \qquad \text{Eq. C-8 I-P}$$

where

$$K_p = \left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \Bigg/ \left(\frac{P_2}{P_1} - 1\right) \qquad \text{Eq. C-9}$$

This is one form of the compressibility coefficient.

C.5 Derivation of K_p in terms of x and z. The compressibility coefficient (K_p) was derived above in terms of the polytropic exponent (n) and the pressure ratio (P_2/P_1). The polytropic exponent can be evaluated from the isentropic exponent (γ) and the polytropic efficiency. The latter may be considered equal to the fan total efficiency for a fan without drive losses. From thermodynamics,

$$\left(\frac{n}{n-1}\right) = \eta_t\left(\frac{\gamma}{\gamma-1}\right) \qquad \text{Eq. C-10}$$

Two new coefficients (x and z), may be defined in terms of the information which is known from a fan test,

$$x = \frac{P_t}{P_1} \text{ and} \qquad \text{Eq. C-11}$$

$$z = \left(\frac{\gamma-1}{\gamma}\right)\left(\frac{H_i}{Q_1 P_1}\right) \qquad \text{Eq. C-12 SI}$$

$$z = \left(\frac{\gamma-1}{\gamma}\right)\left(\frac{6362 H_i}{Q_1 P_1}\right) \qquad \text{Eq. C-12 I-P}$$

Manipulating algebraically,

$$\left(\frac{\gamma}{\gamma-1}\right) = \frac{x}{z}\left(\frac{H_i}{Q_1 P_1}\right) \qquad \text{Eq. C-13 SI}$$

$$\left(\frac{\gamma}{\gamma-1}\right) = \frac{x}{z}\left(\frac{6362 H_i}{Q_1 P_1}\right) \text{ and} \qquad \text{Eq. C-13 I-P}$$

$$\frac{P_2}{P_1} = (1 + x) \qquad \text{Eq. C-14}$$

Substituting in the equation for K_p,

$$K_p = \frac{\eta_t \dfrac{x}{z}\left(\dfrac{H_i}{Q_1 P_t}\right)\left[(1+x)^{(\gamma-1)/\gamma\eta_t} - 1\right]}{(1+x) - 1} \qquad \text{Eq. C-15 SI}$$

$$K_p = \frac{\eta_t \dfrac{x}{z}\left(\dfrac{6362 H_i}{Q_1 P_t}\right)\left[(1+x)^{(\gamma-1)/\gamma\eta_t} - 1\right]}{(1+x) - 1} \qquad \text{Eq. C-15 I-P}$$

This reduces to

$$(1 + z) = (1 + x)^{(\gamma-1)/\gamma\eta_t} \qquad \text{Eq. C-16}$$

Taking logarithms and rearranging

$$\eta_t = \frac{\gamma - 1}{\gamma}\,\frac{\ln(1 + x)}{\ln(1 + z)} \qquad \text{Eq. C-17}$$

Substituting

$$\eta_t = \left(\frac{Q_1 P_t}{H_i} \right) \frac{z}{x} \frac{\ln(1+x)}{\ln(1+x)} \qquad \text{Eq. C-18 SI}$$

$$\eta_t = \left(\frac{Q_1 P_t}{6362 H_i} \right) \frac{z}{x} \frac{\ln(1+x)}{\ln(1+x)} \quad \text{and}$$

Eq. C-18 I-P

$$K_p = \left(\frac{z}{x} \right) \frac{\ln(1+x)}{\ln(1+z)} \qquad \text{Eq. C-19}$$

Since the coefficients x and z have been defined in terms of test quantities, direct solutions of K_p and η_t can be obtained for a test situation. An examination of x and z will reveal that x is the ratio of the total-pressure rise to the absolute total pressure at the inlet, and that z is the ratio of the total-temperature rise to the absolute total temperature at the inlet. If the total-temperature rise could be measured with sufficient accuracy, it could be used to determine z, but in most cases better accuracy is obtained from the other measurements.

C.6 Conversion Equations. The conversion equations which appear in Section 8.9.2 of the standard are simplified versions of the fan laws which are derived in Appendix D. Diameter ratio has been omitted in Section 8.9.2 because there is no need for size conversions in a test standard.

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ ASHRAE Standard 51 but is included for information purposes.

APPENDIX D. Similarity and Fan Laws

D.1 Similarity. Two fans which are similar and have similar airflow conditions will have similar performance characteristics. The degree of similarity of the performance characteristics will depend on the degree of similarity of the fans and of the airflow through the fans.

D.1.1 Geometric Similarity. Complete geometric similarity requires that the ratios of all corresponding dimensions for the two fans be equal. This includes ratios of thicknesses, clearances, and roughnesses as well as all the other linear dimensions of the airflow passages. All corresponding angles must be equal.

D.1.2 Kinematic Similarity. Complete kinematic similarity requires that the ratios of all corresponding velocities for the two fans be equal. This includes the ratios of the magnitudes of corresponding velocities of the air and corresponding peripheral velocities of the impeller. The directions and points of application of all corresponding vectors must be the same.

D.1.3 Dynamic Similarity. Complete dynamic similarity requires that the ratios of all corresponding forces in the two fans be equal. This includes ratios of forces due to elasticity, dynamic viscosity, gravity, surface tension, and inertia as well as the pressure force. The directions and points of application of all corresponding vectors must be the same.

D.2 Symbols. In the derivations which follow, certain symbols and notations are used in addition to those which are used in the standard.

SYMBOL	DESCRIPTION	UNIT
n	Polytropic Exponent	dimensionless
P	Absolute Total Pressure,	*Pa (in. wg)*
$\overline{Q}$	Mean Airflow rate,	*m^3/s (cfm)*
(Prime)	Incompressible Value	--------------

D.3 Fan Laws for Incompressible Flow. The fan laws are the mathematical expressions of the similarity of performance for similar fans at similar flow conditions. These laws may be deduced from similarity considerations, dimensional analysis, or various other lines of reasoning (JORGENSEN, R., *Fan Engineering*, Buffalo Forge Company, Buffalo, New York, 1983, Chapter 12.)

D.3.1 Fan Total Efficiency. The efficiencies of completely similar fans at completely similar flow conditions are equal. This is the fundamental relationship of the fan laws. It emphasizes the fact that the fan laws can be applied only if the points of operation are similarly situated for the two fans being compared. The fan law equation for fan total efficiency (η_t) is, therefore,

$$\eta_{tc} = \eta_t \qquad \text{Eq. D-1}$$

D.3.2 Fan Airflow Rate. The requirements of kinematic similarity lead directly to the airflow rate relationships expressed by the fan laws. Air velocities must be proportional to peripheral velocities. Since airflow rate is proportional to air velocity times flow area, and since area is proportional to the square of any dimension, say impeller diameter (D), it follows that the fan law equation for fan airflow rate (Q) is

$$Q_c = Q\left(\frac{D_c}{D}\right)^3\left(\frac{N_c}{N}\right) \qquad \text{Eq. D-2}$$

D.3.3 Fan Total Pressure. The requirements of dynamic similarity lead directly to the pressure relationships expressed by the fan laws. Pressure forces must be proportional to inertia forces. Since inertia force per unit area is proportional to air density (ρ) and air velocity squared and since air velocity is proportional to peripheral speed, it follows that the fan law equation for fan total pressure (P_t) which is also force per unit area is

$$P_{tc} = P_t\left(\frac{D_c}{D}\right)^2\left(\frac{N_c}{N}\right)^2\left(\frac{\rho_c}{\rho}\right) \qquad \text{Eq. D-3}$$

D.3.4 Fan Power Input. For incompressible flow, the compressibility coefficient is unity and power input is proportional to airflow rate times pressure divided by efficiency. From the above fan law relationships for fan airflow rate, fan total pressure, and fan total efficiency, it follows that the fan law equation for fan power input (H) is

$$H_c = H\left(\frac{D_c}{D}\right)^5\left(\frac{N_c}{N}\right)^3\left(\frac{\rho_c}{\rho}\right) \qquad \text{Eq. D-4}$$

D.3.5 Fan Velocity Pressure. The fan law equation for fan velocity pressure (P_v) follows from that for fan total pressure,

$$P_{vc} = P_v\left(\frac{D_c}{D}\right)^2\left(\frac{N_c}{N}\right)^2\left(\frac{\rho_c}{\rho}\right) \qquad \text{Eq. D-5}$$

D.3.6 Fan Static Pressure. By definition,

$$P_{sc} = P_{tc} - P_{vc} \qquad \text{Eq. D-6}$$

D.3.7 Fan Static Efficiency. By definition,

$$\eta_{sc} = \eta_{tc}\left(\frac{P_{sc}}{P_{tc}}\right) \qquad \text{Eq. D-7}$$

D.4 Fan Laws for Compressible Flow. More general versions of the fan laws, which recognize the compressibility of air, can also be deduced from similarity considerations [23].

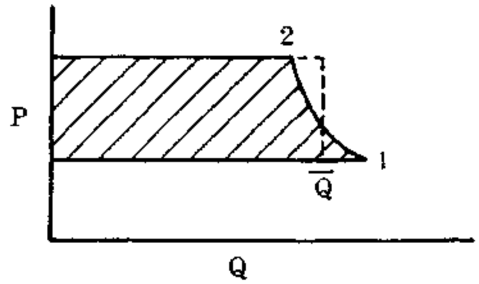
D.4.1 Fan Total Efficiency. It is obvious that airflow conditions can never be completely similar, even for two completely similar fans, if the degree of compression varies. Nevertheless, it is useful and convenient to assume that the fan law equation for fan total efficiency (η_t) need not be modified,

$$\eta_{tc} = \eta_t \qquad \text{Eq. D-8}$$

D.4.2 Fan Airflow Rate. Continuity requires that the mass airflow rate at the fan outlet equal that at the fan inlet. If the volumetric airflow rate at the inlet (Q_1) is proportional to peripheral speed, the volumetric airflow rate at the outlet (Q_2) cannot be proportional to peripheral speed or vice versa except for the same degree of compression. There is some average airflow rate which is proportional to peripheral speed and flow area. Since for a polytropic process, the airflow rate is an exponential function of pressure, the geometric mean of the airflow rates at the inlet and outlet will be a very close approximation of the average airflow rate($\overline{Q}$). The geometric mean is the square root of the product of the two end values,

$$\overline{Q} \approx \left(Q_1 Q_2\right)^{1/2} \qquad \text{Eq. D-9}$$

the value ($\overline{Q}$) illustrated in the following diagram is the average airflow rate based on power output. This value yields the same power output as the polytropic process over the same range of pressures.



For the polytropic process,

$$H_o = \frac{Q_1 P_t K_p}{1} \qquad \text{Eq. D-10 SI}$$

$$H_o = \frac{Q_1 P_t K_p}{6362} \qquad \text{Eq. D-10 I-P}$$

For the rectangle,

$$H_o = \frac{\overline{Q} P_t}{1} \qquad \text{Eq. D-11 SI}$$

$$H_o = \frac{\overline{Q} P_t}{6362} \qquad \text{Eq. D-11 I-P}$$

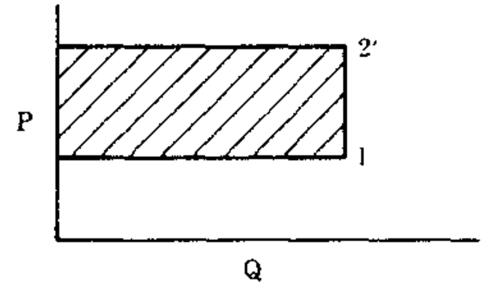
Therefore,

$$\overline{Q} = Q_1 K_p = Q K_p \qquad \text{Eq. D-12}$$

This average airflow rate can be substituted in D-2 to give the compressible flow fan law equation for fan airflow rate,

$$Q_c = Q\left(\frac{D_c}{D}\right)^3\left(\frac{N_c}{N}\right)\left(\frac{K_p}{K_{pc}}\right) \qquad \text{Eq. D-13}$$

D.4.3 Fan Total Pressure. The incompressible flow fan laws are based on a process which can be diagramed as shown below.



The fan power output is proportional to the shaded area which leads to

$$H_o = Q_1 (P_2' - P_1) / 1 \quad \text{Eq. D-14 SI}$$

$$H_o = Q_1 (P_2' - P_1) / 6362 \quad \text{Eq. D-14 I-P}$$

Extending the definition of fan total pressure to the incompressible case

$$P_t' = (P_2' - P_1) \quad \text{Eq. D-15}$$

Therefore,

$$H_o = Q_1 P_t' / 1 \quad \text{Eq. D-16 SI}$$

$$H_o = Q_1 P_t' / 6362 \quad \text{Eq. D-16 I-P}$$

For the same airflow rate (Q_1), absolute inlet pressure (P_1), and power output (H_o), the corresponding equation for compressible flow is

$$H_o = Q_1 P_1 K_p / 1 \quad \text{Eq. D-17 SI}$$

$$H_o = Q_1 P_1 K_p / 6362 \quad \text{Eq. D-17 I-P}$$

It follows that,

$$P_t' = P_1 K_p \quad \text{Eq. D-18}$$

The compressible flow fan law equation for fan total pressure can, therefore, be obtained by substitution,

$$P_{tc} = P_t \left(\frac{D_c}{D}\right)^2 \left(\frac{N_c}{N}\right)^2 \left(\frac{\rho_c}{\rho}\right) \left(\frac{K_p}{K_{pc}}\right) \quad \text{Eq. D-19}$$

D.4.4 Fan Power Input. The equation for efficiency may be rearranged to give either

$$H = \frac{Q P_t K_p}{1 \eta_t} \quad \text{Eq.D-20 SI}$$

$$H = \frac{Q P_t K_p}{6362 \eta_t} \text{ or} \quad \text{Eq. D-20 I-P}$$

$$H_c = \frac{Q_c P_{tc} K_{pc}}{1 \eta_{tc}} \quad \text{Eq. D-21 SI}$$

$$H_c = \frac{Q_c P_{tc} K_{pc}}{6362 \eta_{tc}} \quad \text{Eq. D-21 I-P}$$

Combining and using the compressible flow fan law relationships for fan airflow rate, fan total pressure, and fan total efficiency, it follows that the compressible flow fan law equation for fan power input is,

$$H_c = H \left(\frac{D_c}{D}\right)^5 \left(\frac{N_c}{N}\right)^3 \left(\frac{\rho_c}{\rho}\right) \left(\frac{K_p}{K_{pc}}\right) \quad \text{Eq. D-22}$$

D.4.5 Fan Velocity Pressure. By definition,

$$P_v = P_{v2} = \left(Q_2 / \sqrt{2}\, A_2\right)^2 \rho_2 \quad \text{Eq. D-23 SI}$$

$$P_v = P_{v2} = \left(Q_2 / 1097\, A_2\right)^2 \rho_2 \quad \text{Eq. D-23 I-P}$$

But from continuity,

$$\rho_2 Q_2 = \rho_1 Q_1 = \rho Q_1 \quad \text{Eq. D-24}$$

Therefore,

$$P_v = \rho Q_1 Q_2 / (\sqrt{2}\, A_2)^2 \quad \text{Eq. D-25 SI}$$

$$P_v = \rho Q_1 Q_2 / (1097\, A_2)^2 \quad \text{Eq. D-25 I-P}$$

But from D-9 and D-12,

$$\overline{Q^2} = Q^2 K_p^2 \approx Q_1 Q_2 \quad \text{Eq. D-26}$$

It follows that

$$P_v = \rho Q^2 K_p^2 / \left(\sqrt{2}\, A_2\right)^2 \quad \text{Eq. D-27 SI}$$

$$P_v = \rho Q^2 K_p^2 / \left(1097\, A_2\right)^2 \quad \text{Eq. D-27 I-P}$$

By similar reasoning,

$$P_{vc} = \rho_c Q_c^{\,2} K_{pc}^{\,2} \Big/ \left(\sqrt{2}\ A_{2c}\right)^2 \qquad \text{Eq. D-28 SI}$$

$$P_{vc} = \rho_c Q_c^{\,2} K_{pc}^{\,2} \Big/ \left(1097\ A_{2c}\right)^2 \qquad \text{Eq. D-28 I-P}$$

By using the compressible flow fan law relationships for fan airflow rate and the proportionality of outlet area to diameter squared, it follows that the compressible flow fan law equation for fan velocity pressure is

$$P_{vc} = P_v \left(\frac{D_c}{D}\right)^2 \left(\frac{N_c}{N}\right)^2 \left(\frac{\rho_c}{\rho}\right) \qquad \text{Eq. D-29}$$

D.4.6 Fan Static Pressure. By definition,

$$P_{sc} = P_{tc} - P_{vc} \qquad \text{Eq. D-30}$$

D.4.7 Fan Static Efficiency. By definition,

$$\eta_{sc} = \eta_{tc} \left(\frac{P_{sc}}{P_{tc}}\right) \qquad \text{Eq. D-31}$$

D.5 Fan Law deviations. Among the requirements for complete similarity are those for equal force ratios that lead to Reynolds and Mach number considerations.

D.5.1 Reynolds Number. There is some evidence that efficiency improves with an increase in Reynolds number (NIXON, R. A., *Examination of the Problem of Pump Scale Laws*, National Engineering Laboratory, Glasgow, Scotland, U.K., Paper 2D-1, 1967. (AMCA #1161)). However, that evidence is not considered to be sufficiently well documented to incorporate any rules in this Appendix. There is also some evidence that performance drops off with a significant decrease in Reynolds number. The fan laws should not be employed if it is suspected that the airflow regimes are significantly different because of a difference in Reynolds number.

D.5.2 Mach Number. There is evidence that choking occurs when the Mach number at any point in the flow passages approaches unity. Obviously, the fan laws should not be employed if this condition is suspected.

D.5.3 Bearing and Drive Losses. While there may be other similarity laws covering bearings and other drive elements, the fan laws cannot be used to predict bearing or drive losses. The correct procedure is to subtract the losses for the first condition, make fan law projections of power input for the corrected first condition to the second condition, and then add the bearing and drive losses for the second condition.

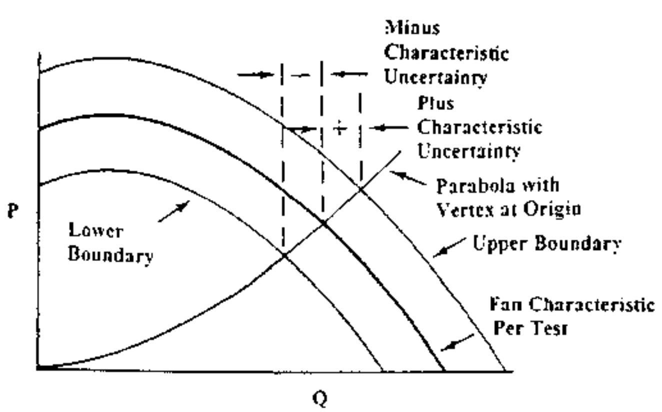
This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ ASHRAE Standard 51 but is included for information purposes.

APPENDIX E. Uncertainties Analysis [9]

E.1 General. This analysis is based on the assumption that fan performance can be treated as a statistical quantity and that the performances derived from repeated tests would have a normal distribution. The best estimate of the true performance would, therefore, be the mean results based on repeated observations at each point of operation. Since only one set of observations is specified in the standard, this analysis must deal with the uncertainties in the results obtained from a single set of observations.

The results of a fan test are a complex combination of variables which must be presented graphically according to the standard. In order to simplify this analysis, test results will be considered to be the curves of fan static pressure versus fan airflow rate and fan static efficiency versus fan airflow rate. Analysis of fan power input is unnecessary since it is a part of efficiency analysis. The findings from a total pressure analysis would be similar to those of a static pressure analysis.

The uncertainty in the results will be expressed in two parts, both of which will be based on the uncertainties in various measurements. That part dealing with the pressure versus airflow rate curve will be called the characteristic uncertainty and that dealing with the efficiency versus airflow rate curve will be called the efficiency uncertainty. The characteristic uncertainty can be defined with reference to the following diagram:



The diagram shows a plot of the fan static pressure versus fan airflow rate as determined by test per this standard. Surrounding this curve is a band of uncertainties, the boundaries of which are roughly parallel to the test curve. Also shown is a parabola with vertex at the origin that intersects the fan curve and both of the boundaries. The characteristic uncertainty is defined as the difference in airflow rate between the intersection of the parabola with the test curve and the intersections of the parabola with the boundaries. Typically, the absolute value of the characteristic uncertainty would be ± a certain number of *m^3/s (cfm)*. The relative characteristic uncertainty would be the absolute characteristic uncertainty divided by the airflow rate at the intersection with the test curve.

The absolute efficiency uncertainty is defined as the difference in efficiency between that at points corresponding to the above mentioned intersections with the boundaries and that at the above mentioned intersection with the fan test curve. Typically, this would be expressed as ± so many percent. The relative efficiency uncertainty would be the absolute efficiency uncertainty divided by the efficiency at the point corresponding to the above mentioned intersection with the test curve.

The accuracies specified in the standard are based on two standard deviations. This means that there should be a 95% probability that the uncertainty in any measurement will be less than the specified value. Since the characteristic uncertainty and the efficiency uncertainty are based on these measurements, there will be a 95% probability that these uncertainties will be less than the calculated value.

E.2 Symbols. In the analysis which follows, certain symbols and notations are used in addition to those which are used in the standard.

SYMBOL	QUANTITY
dP/dQ	Slope of Fan Characteristic
e_x	Per Unit Uncertainty in X
ΔX	Absolute Uncertainty in X
F_x	Correlation Factor for X

SUBSCRIPT	DESCRIPTION
A	Area
b	Barometric Pressure
C	Nozzle Discharge Coefficient
d	Dry-bulb Temperature
f	Pressure for Airflow Rate
g	Pressure for Fan Pressure
H	Fan Power Input
K	Character
m	Maximum

N	Fan Speed
o	Fan Power Output
P	Fan Pressure
Q	Fan Airflow Rate
T	Torque
V	Variable as Defined in Equation D-11
w	For Wet-bulb Depression, in t_w
X	Generalized Quantity (A, b, ...ρ)
η	Fan Efficiency
ρ	Fan Air Density

E.3 Measurement Uncertainties. The various measurement uncertainties which are permitted in the standard are listed below with some of the considerations that led to their adoption.

(1) Barometric pressure is easily measured within the ±170 *Pa* (±0.05 *in. Hg)* specified,

$$e_b = 1.70 / p_b \qquad \text{Eq. E-1 SI}$$

$$e_b = 0.05 / p_b \qquad \text{Eq. E-1 I-P}$$

(2) Dry-bulb temperature is easily measured within the ±1°C (±2.0°*F*) specified if there are no significant radiation sources,

$$e_d = 1.0 / (t_d + 273.15) \qquad \text{Eq. E-2 SI}$$

$$e_d = 2.0 / (t_d + 459.67) \qquad \text{Eq. E-2 I-P}$$

(3) Wet-bulb depression is easily measured within 3°C (5.0°*F*) if temperature measurements are within 1°C (2.0°*F*) and if air velocity is maintained in the specified range,

$$e_w = 3 / (t_d - t_w) \qquad \text{Eq. E-3 SI}$$

$$e_w = 5.0 / (t_d - t_w) \qquad \text{Eq. E-3 I-P}$$

(4) Fan speed requires careful measurement to hold the 0.5% tolerance specified,

$$e_N = 0.005 \qquad \text{Eq. E-4}$$

(5) Torque requires careful measurement to hold the 2.0% tolerance specified,

$$e_T = 0.02 \qquad \text{Eq. E-5}$$

(6) Nozzle discharge coefficients given in the standard have been obtained from ISO data and nozzles made to specifications should perform within a tolerance of 1.2% according to that data. A properly performed laboratory traverse is assumed to have equal accuracy,

$$e_c = 0.012 \qquad \text{Eq. E-6}$$

(7) The area at the flow measuring station will be within 0.5% when the diameter measurements are within the 0.2% specified,

$$e_A = 0.005 \qquad \text{Eq. E-7}$$

(8) The tolerance on the pressure measurement for determining flow rate is specified as 1% of the maximum reading during the test. This is easily obtained by using the specified calibration procedures. In addition, an allowance must be made for the mental averaging which is performed on fluctuating readings. This is estimated to be 1% of the reading. Using the subscript *m* to denote the condition for the maximum reading, a combined uncertainty can be written,

$$e_f = \left\{ (0.01)^2 + \left[0.01 \left(\frac{Q_m}{Q} \right)^2 \right]^2 \right\}^{1/2} \qquad \text{Eq. E-8}$$

(9) The pressure measurement for determining fan pressure is also subject to an instrument tolerance of 1% of the maximum reading and an averaging tolerance of 1% of the reading. In addition, there are various uncertainties which are related to the velocity pressure. A tolerance of 10% of the fan velocity pressure should cover the influence of yaw on pressure sensors, friction factor variances, and other possible effects,

$$e_g = \left\{ (0.01)^2 + \left[0.01 \left(\frac{P_m}{P} \right) \right]^2 + \left[0.1 \left(\frac{P_v}{P} \right) \right]^2 \right\}^{1/2}$$

Eq. E-9

E.4 Combined Uncertainties. The uncertainties in the test performance are the result of using various values each of which is associated with an uncertainty. The combined uncertainty for each of the fan performance variables is given below. The characteristic uncertainty and the efficiency uncertainty are also given.

1) Fan air density involves the various psychrometric

measurements and the approximate formula

$$\rho = \frac{p_b V}{R\left(t_d + 273.15\right)} \qquad \text{Eq. E-10 SI}$$

$$\rho = \frac{70.73\, p_b V}{R\left(t_d + 459.67\right)} \qquad \text{Eq. E-10 I-P}$$

where

$$V = \left\{1.0 - 0.378\left[\frac{p_e}{p_b} - \frac{\left(t_d - t_w\right)}{1500}\right]\right\} \qquad \text{Eq. E-11 SI}$$

$$V = \left\{1.0 - 0.378\left[\frac{p_e}{p_b} - \frac{\left(t_d - t_w\right)}{2700}\right]\right\} \qquad \text{Eq. E-11 I-P}$$

For random and independent uncertainties in products, the combined uncertainty is determined as follows:

$$\frac{\Delta\rho}{\rho} = \left\{\left(\frac{\Delta 1.0}{1}\right)^2 + \left(\frac{\Delta p_b}{p_b}\right)^2 + \left(\frac{\Delta V}{V}\right)^2 + \left(\frac{\Delta R}{R}\right)^2 + \left(\frac{\Delta t_d}{T_d + 273.15}\right)^2\right\}^{1/2} \qquad \text{Eq. E-12 SI}$$

$$\frac{\Delta\rho}{\rho} = \left\{\left(\frac{\Delta 70.73}{70.73}\right)^2 + \left(\frac{\Delta p_b}{p_b}\right)^2 + \left(\frac{\Delta V}{V}\right)^2 + \left(\frac{\Delta R}{R}\right)^2 + \left(\frac{\Delta t_d}{T_d + 459.67}\right)^2\right\}^{1/2} \qquad \text{Eq. E-12 I-P}$$

Assuming $\Delta 1.0$ ($\Delta 70.73$) and ΔR are both zero,

$$e_\rho = (e_b^2 + e_V^2 + e_d^2)^{1/2} \qquad \text{Eq. E-13}$$

It can be shown that

$$e_V^2 = [(0.00002349\, t_W - 0.0003204)\, \Delta(t_d - t_w)]^2 \qquad \text{Eq. E-14 SI}$$

$$e_V^2 = [(0.00000725\, t_W - 0.0000542)\, \Delta(t_d - t_w)]^2 \qquad \text{Eq. E-14 I-P}$$

(2) Fan airflow rate directly involves the area at the flow measuring station, the nozzle discharge coefficient, the square root of the pressure measurement for flow, and the square root of the air density. When making fan law conversions, fan speed has a first power effect on airflow rate. The effects of uncertainties in each of these variables can be expressed mathematically as follows, where e_{QX} is the uncertainty in airflow rate due to the uncertainty in X.

$$\left.\begin{array}{ll} e_{QA} = e_A & e_{QN} = e_N \\ e_{QC} = e_C & e_{Q\rho} = \dfrac{e_\rho}{2} \\ e_{Qf} = \dfrac{e_f}{2} & e_{QT} = 0 \\ e_{Qg} = 0 & \end{array}\right\} \qquad \text{Eq. E-15}$$

The uncertainty in the airflow rate only can be determined from the above uncertainties by combining,

$$e_Q = \left[e_c^2 + e_A^2 + \left(\frac{e_f}{2}\right)^2 + \left(\frac{e_\rho}{2}\right)^2 + e_N^2\right]^{1/2} \qquad \text{Eq. E-15A}$$

(3) Fan pressure directly involves the pressure measurement for fan pressure. In addition, when making fan law conversions, air density has a first power effect on fan pressure while fan speed produces a second power effect. Mathematically,

$$\left.\begin{array}{ll} e_{PA} = 0 & e_{PN} = 2\, e_N \\ e_{PC} = 0 & e_{P\rho} = e_\rho \\ e_{Pf} = 0 & e_{PT} = 0 \\ e_{Pg} = e_g & \end{array}\right\} \qquad \text{Eq. E-16}$$

The uncertainty in the fan pressure only can be determined from the above uncertainties by combining,

$$e_P = \left[e_g^2 + e_\rho^2 + \left(2e_N\right)^2\right]^{1/2} \qquad \text{Eq. E-16A}$$

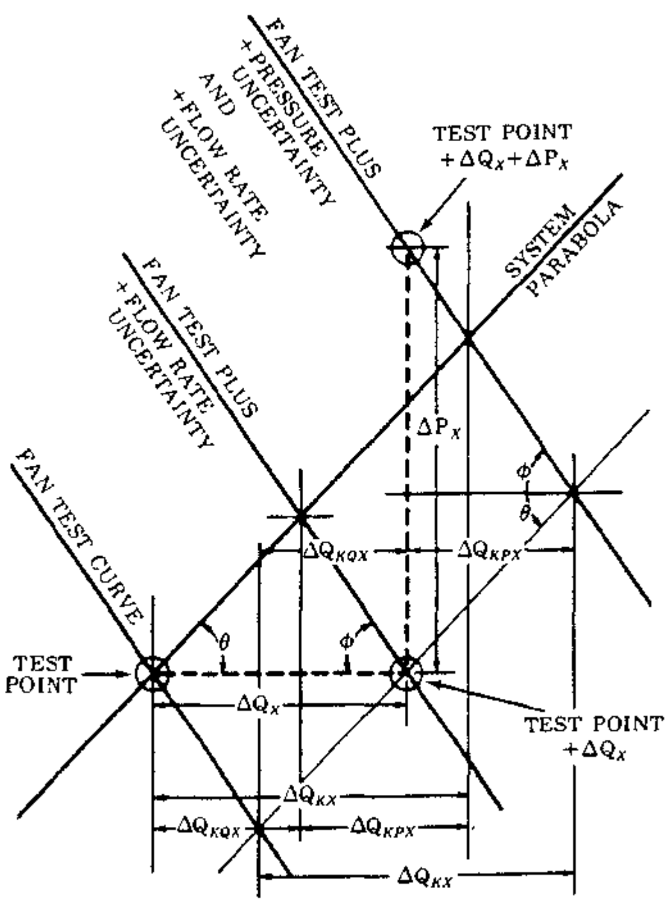
(4) Fan power input directly involves the torque and speed measurements. In addition, when making fan law conversions, density has a first power effect and speed a third power effect on fan power input. The net effect with respect to speed is second power. Mathematically,

$$\left.\begin{array}{ll} e_{HA} = 0 & e_{HN} = 2\,e_N \\ e_{HC} = 0 & e_{H\rho} = e_\rho \\ e_{Hf} = 0 & e_{HT} = e_T \\ e_{Hg} = 0 & \end{array}\right\} \quad \text{Eq. E-17}$$

The uncertainty in the fan power input only can be determined from the above uncertainties by combining,

$$e_H = \left[e_T^{\,2} + e_\rho^{\,2} + \left(2e_N\right)^2\right]^{1/2} \quad \text{Eq. E-17A}$$

(5) The uncertainties in the measurements for fan airflow rate and fan pressure create the characteristic uncertainty as defined in E.1. Assuming the uncertainties are small, the characteristic curves and parabola can be replaced by their tangents, and the effects of uncertainty in each measurement, (X), on the characteristic uncertainty can be determined. At a point (Q,P), the uncertainty in measurement (X) results in an uncertainty in Q and P of ΔQ_X and ΔP_X. For ΔQ_X,



$$\Delta Q_{KQX} \tan\theta = \left(\Delta Q_X - \Delta Q_{KQX}\right)\tan\phi \quad \text{Eq. E-18}$$

$$\Delta Q_{KQX} = \Delta Q_X \left[\frac{\tan\phi}{\tan\theta + \tan\phi}\right] \quad \text{Eq. E-19}$$

For ΔP_X,

$$\Delta Q_{KPX}\left(\tan\theta + \tan\phi\right) = \Delta P_X \quad \text{Eq. E-20}$$

$$\Delta Q_{KPX} = \Delta P_X \left[\frac{1}{\tan\theta + \tan\phi}\right] \quad \text{Eq. E-21}$$

Summing and simplifying by relating the tangents to the slopes of the parabola and the fan characteristic curve.

$$\Delta Q_{KX} = \Delta Q_{KQX} + \Delta Q_{KPX} \quad \text{Eq. E-22}$$

$$\tan\theta = 2\left(\frac{P}{Q}\right) \text{ and} \quad \text{Eq. E-23}$$

$$\tan\phi = -\left(\frac{dP}{dQ}\right) \quad \text{Eq. E-24}$$

$$\Delta Q_{KX} = \Delta Q_X \left[\frac{-\left(\frac{dP}{dQ}\right)}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)}\right] + \Delta P_X \left[\frac{1}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)}\right] \quad \text{Eq. E-25}$$

$$e_{KX} = e_{QX} \left[\frac{-\left(\frac{dP}{dQ}\right)}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)}\right] + \frac{e_{PPX}}{2} \left[\frac{2\left(\frac{P}{Q}\right)}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)}\right] \quad \text{Eq. E-26}$$

Introducing correlation factors

$$F_Q = \left[\frac{-\left(\frac{dP}{dQ}\right)}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)} \right] \quad \text{and} \qquad \text{Eq. E-27}$$

$$F_P = \left[\frac{2\left(\frac{P}{Q}\right)}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)} \right] \qquad \text{Eq. E-28}$$

$$e_{KX} = e_{QX} F_Q + \left(\frac{e_{PX}}{2}\right) F_P \qquad \text{Eq. E-29}$$

Combining equations E-15, E-16, and E-29,

$$e_{KA} = e_A F_Q,\ e_{Kg} = \left(\frac{e_g}{2}\right) F_P$$

$$e_{KC} = e_C F_Q,\ e_{KN} = e_N \left(F_Q + F_P\right) \qquad \text{Eq. E-30}$$

$$e_{Kf} = \left(\frac{e_f}{2}\right) F_Q,\ e_{K\rho} = \frac{e_\rho}{2}\left(F_Q + F_P\right)$$

Assuming these uncertainties are independent, they can be combined for the characteristic uncertainty as follows, noting that $F_Q + F_p = 1$,

$$e_K = \left\{ \left(\frac{e_\rho}{2}\right)^2 + e_N^{\ 2} + F_P^{\ 2}\left(\frac{e_g}{2}\right)^2 + F_Q^{\ 2}\left[e_C^{\ 2} + e_A^{\ 2} + \left(\frac{e_f}{2}\right)^2\right]\right\}^{1/2} \qquad \text{Eq. E-31}$$

(6) Fan power output is proportional to the third power of airflow rate along a system characteristic. Therefore,

$$e_o = 3\ e_K \qquad \text{Eq. E-32}$$

(7) Fan efficiency uncertainty was defined in E.1. Using the above noted correlation factors and recombining the components,

$$e_\eta = \left\{ \left(\frac{e_\rho}{2}\right)^2 + e_N^{\ 2} + e_T^{\ 2} + 9\left[F_P^{\ 2}\left(\frac{e_g}{2}\right)^2 + F_Q^{\ 2}\left(e_C^{\ 2} + e_A^{\ 2} + \left\{\frac{e_f}{2}\right\}^2\right)\right]\right\}^{1/2} \qquad \text{Eq. E-33}$$

E.5 Example. The characteristic test curve for a typical backward-curve centrifugal fan was normalized on the basis of shut-off pressure and free-delivery airflow rate. The resultant curve is shown in Figure E-1.

An uncertainty analysis based on this curve and the maximum allowable measurement tolerances follows:

(1) The maximum allowable measurement tolerances can be determined using the information from Section E.3. Where appropriate, lowest expected barometer and temperature for a laboratory at sea level are assumed.

Per unit uncertainties are:

$e_b = [0.2/96.5] = 0.0021$ Eq. E-34 SI
$e_b = [0.05/28.5] = 0.0018$ Eq. E-34 I-P

$e_d = [1.0/(15.5 + 273.2)] = 0.0035$ Eq. E-35 SI
$e_d = [2.0/(60 + 459.7)] = 0.0038$ Eq. E-35 I-P

$e_w = [3.0/(15.5 - 10)] = 0.545$ Eq. E-36 SI
$e_w = [5.0/(60 - 50)] = 0.5$ Eq. E-36 I-P

$e_N = 0.005$

$e_T = 0.02$

$e_C = 0.012$

$e_A = 0.005$

$$e_f = \left\{(0.01)^2 + \left[0.01\left(\frac{Q_m}{Q}\right)^2\right]^2\right\}^{1/2} \quad \text{and}$$

$$e_g = \left\{(0.01)^2 + \left[0.01\left(\frac{P_m}{P}\right)\right]^2 + \left[0.1\left(\frac{P_v}{P}\right)\right]^2\right\}^{1/2}$$

Eq. E-37

Note that e_f and e_g vary with point of operation. In this example, the values of Q_m, Q, P_m, and P are taken from Figure E-1. The velocity pressure at free delivery is taken to be 20% of the maximum static pressure.

(2) The various combined uncertainties and factors can be determined using the information from Section E.4. To illustrate, the per unit uncertainty in air density will be calculated:

$$e_\rho = (e_b^2 + e_V^2 + e_d^2)^{1/2}$$

$e_b^2 = [0.2/96.5]^2 = 0.0000043$ Eq. E-38 SI

$e_b^2 = [0.05/28.5]^2 = 0.00000308$ Eq. E-38 I-P

$e_V^2 = [(0.00002349 \times 10 + 0.0003204)\ (3.0)]^2 = 0.0000028$ Eq. E-39 SI

$e_V^2 = [(0.00000725 \times 50 + 0.0000542)\ (5.0)]^2 = 0.00000238$ Eq. E-39 I-P

$e_d^2 = [1.0/(15.5 + 273.2)]^2 = 0.000012$ Eq. E-40 SI

$e_d^2 = [2.0/(60 + 459.7)]^2 = 0.0000148$ Eq. E-40 I-P

and

$e_\rho = 0.0045$

This is the expected accuracy for a laboratory at sea level. For extremes of altitude and wet-bulb temperatures, the limit is $e_\rho = 0.005$.

(3) The characteristic uncertainty and the efficiency uncertainty can be calculated for various points of operation as indicated in Table E-1.

The values of Q, P, and $-(dP/dQ)$ have been read directly from the normalized fan curve. The results have been plotted as curves of per unit uncertainty versus airflow rate in Figure E-2.

E.6 Summary. The example is based on uncertainties which, in turn, are based on 95% confidence limits. Accordingly, the results of 95% of all tests will be better than indicated. Per unit uncertainties of one half those indicated will be achieved in 68% of all tests while indicated per unit uncertainties will be exceeded in 5% of all tests. The following conclusions can be drawn from the above example.

(1) The characteristic uncertainty for the specified tolerances is about 1% near the best efficiency point and approaches 2% at free delivery. The uncertainty also increases rapidly as shutoff is approached.

(2) The fan efficiency uncertainty is about 3% near the best efficiency point and exceeds 5% at free delivery. The uncertainty increases rapidly near shutoff.

(3) Psychrometric measurement uncertainties have very little effect on overall accuracy. Calibration corrections are unnecessary in most cases.

(4) The nozzle discharge coefficient uncertainty has a very significant effect on overall accuracy. The 1.2% tolerance specified was based on the current state of the art. Any significant improvement in the accuracy of test results will depend on further work to reduce the uncertainty of this quantity.

(5) While the example was based on a typical characteristic for a backward-curve centrifugal fan, analyses of different characteristics for other fan types will yield sufficiently similar results that the same conclusion can be drawn.

(6) This analysis has been limited to a study of measurement uncertainties in laboratory setups. Other factors may have an equal or greater effect on fan performance. The results of an on-site test may deviate from predicted values because of additional uncertainties in measurements such as poor approach conditions to measuring stations. Deviations may also be due to conditions affecting the airflow into or out of the fan which, in turn, affects the ability of the fan to perform. Differences in construction, which arise from manufacturing tolerances, may cause full-scale test performance to deviate from model performance.

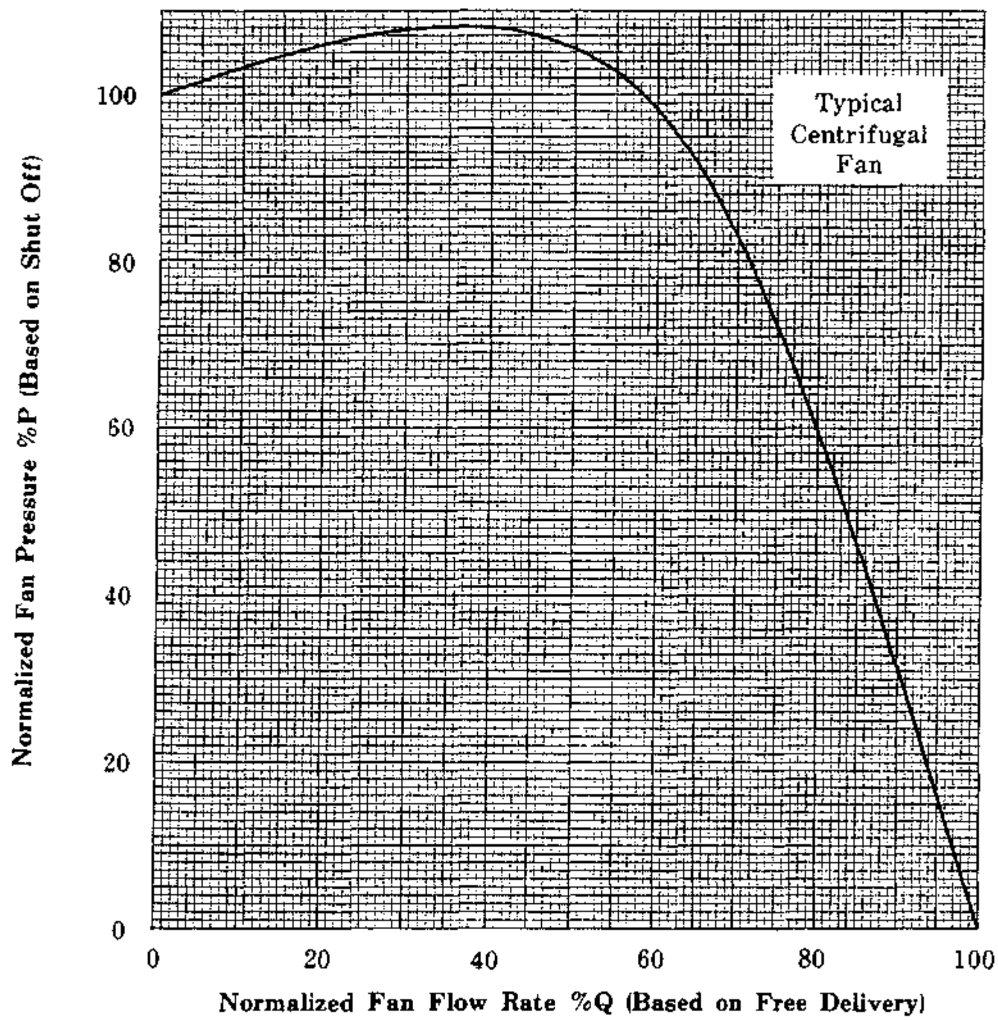


Figure E-1 Normalized Fan Performance Curve

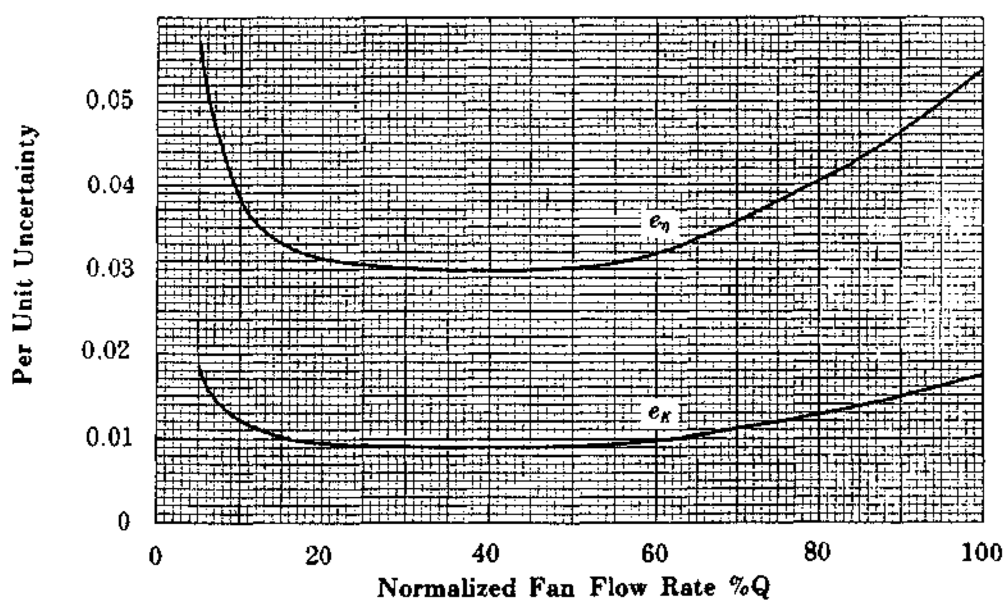


Figure E-2 Normalized Test Results Uncertainties

TABLE E-1 TABULATION FOR UNCERTAINTY ANALYSIS OF FIGURE E-1

$\%Q$	$\%P$	$\left(\frac{dP}{dQ}\right)$	F_P	F_Q	$\left[\left(\frac{e_\rho}{2}\right)^2 + e_N^2\right]$	$\left[F_P^2\left(\frac{e_g}{2}\right)^2\right]$	$\left[F_Q^2\left(e_c^2 + e_A^2 + \frac{e_f^2}{4}\right)\right]$	e_K	e_η
99	3.2	3.215	.01971	.98029	31.2 E-06	53.5 E-06	211.4 E-06	.0172	.0531
95	16.0	3.075	.09873	.90127	31.2 E-06	47.5 E-06	182.5 E-06	.0162	.0500
90	31.5	2.900	.19444	.80556	31.2 E-06	41.2 E-06	150.6 E-06	.0149	.0464
85	46.0	2.700	.28616	.71384	31.2 E-06	36.8 E-06	123.2 E-06	.0138	.0433
80	59.5	2.500	.37304	.62696	31.2 E-06	33.7 E-06	100.2 E-06	.0129	.0405
75	72.0	2.275	.45769	.54231	31.2 E-06	31.9 E-06	80.2 E-06	.0120	.0379
70	82.7	1.950	.54786	.45214	31.2 E-06	32.5 E-06	60.9 E-06	.0112	.0357
65	91.2	1.575	.64051	.35949	31.2 E-06	34.9 E-06	43.1 E-06	.0105	.0337
60	98.0	1.150	.73962	.26038	31.2 E-06	38.8 E-06	26.2 E-06	.0098	.0319
55	102.6	.800	.82343	.17657	31.2 E-06	42.6 E-06	14.5 E-06	.0094	.0307
50	105.3	.500	.89389	.10611	31.2 E-06	46.2 E-06	6.6 E-06	.0092	.0301
45	107.0	.250	.95006	.04994	31.2 E-06	49.3 E-06	2.0 E-06	.0091	.0299
40	107.9	.050	.99082	.00918	31.2 E-06	51.6 E-06	0 E-06	.0091	.0299
35	108.0	-.025	1.00407	-.00407	31.2 E-06	51.9 E-06	0 E-06	.0091	.0300
30	107.6	-.100	1.01414	-.01414	31.2 E-06	52.4 E-06	0.6 E-06	.0092	.0301
25	107.0	-.175	1.02087	-.02087	31.2 E-06	53.0 E-06	2.8 E-06	.0093	.0306
20	106.0	-.225	1.02169	-.02169	31.2 E-06	53.5 E-06	7.4 E-06	.0096	.0313
15	104.7	-.275	1.02009	-.02009	31.2 E-06	53.7 E-06	20.0 E-06	.0102	.0331
10	103.2	-.325	1.01600	-.01600	31.2 E-06	54.0 E-06	64.0 E-06	.0122	.0386
5	101.6	-.325	1.00806	-.00806	31.2 E-06	54.1 E-06	259.8 E-06	.0186	.0571

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ASHRAE Standard 51 but is included for information purposes.

APPENDIX F. ITERATIVE PROCEDURE

To obtain the value of C to be used in calculating the chamber nozzle airflow rate in 9.3.2.7, an iteration process is used. A calculated value of Re is made and using an estimated value of C. The calculated value of Re is then used to recalculate C until the difference between two successive trial values of C is < 0.001, at which point the last trial value of C is taken as the value to be used in calculating chamber nozzle volume. In the following example, the first estimate of Re is made using an estimated value of Ce = 0.99. It is suggested that calculations be carried out to at least 5 decimal places.

EXAMPLE ITERATION
F.1 Example Iteration (SI System of Units)

Iteration 1:

Step 1-1: Calculate Re, using

$$Re = \frac{\sqrt{2}}{\mu_6} Ce\ D_6\ Y \sqrt{\frac{\Delta P\ \rho_5}{1 - E\beta^4}} \qquad \text{Eq. F-1 SI}$$

$$? = \frac{1097}{60\ \mu_6} Ce\ D_6\ Y \sqrt{\left(\frac{\Delta P\ \rho_5}{[1 - E\ \beta^4}\right.} \qquad \text{Eq. F-1 I-P}$$

where:

μ_6 = 1.819 E-05 *Pa•s* (1.222E-05 *lbm/ft•s*)
Ce = 0.95 (estimated) (0.95 estimated)
D_6 = 0.15 *m* (6 *in.* = 0.5 *ft*)
Y = 0.998 (calculate per Section 9.3.2)
ΔP = 250 *Pa* (1.005 *in. wg*)
ρ_5 = 1.14 *kg/m³* (0.0711 *lbm/ft³*)
$(1\text{-}E\ \beta^4)$ = 1 for iteration purposes

Then:

$$Re_1 = \frac{\sqrt{2}}{\mu_6} Ce\ D_6\ Y \sqrt{\Delta P\ \rho_5}$$

$$= \frac{\sqrt{2}}{1.819\text{E}-05} (0.95)\ (0.15)\ (0.998) \sqrt{(250)\ (1.14)}$$

$$= 186\ 660$$

$$Re_1 = \frac{1097}{60\ \mu_6} Ce\ D_6\ Y \sqrt{\Delta P \rho_5}$$

$$= \frac{1097}{(60)\,1.222E-05} (0.95)(0.5)(0.998) \sqrt{(1.005)(0.0711)}$$

$$= 189\ 595$$

Step 1-2: Using Eq. F-2 (Eq. 8.19), calculate Ce_1, using Re_1 from the previous step, assuming that L/D = 0.6:

$$Ce_1 = 0.9986 - \frac{7.006}{\sqrt{Re}} + \frac{134.6}{Re}$$

$$= 0.9986 - \frac{7.006}{\sqrt{186660}} + \frac{134.6}{186660}$$

$$= 0.98311 \qquad \text{Eq. F-2 SI}$$

Check: $C_l - Ce$ = 0.98311 - 0.95
= 0.03311
Since 0.0331 > 0.001, another iteration is required.

$$Ce_1 = 0.9986 - \frac{7.006}{\sqrt{Re}} + \frac{134.6}{Re}$$

$$= 0.9986 - \frac{7.006}{\sqrt{195595}} + \frac{134.6}{189595}$$

$$= 0.98322 \qquad \text{Eq. F-2 I-P}$$

Check: $Ce_1 - Ce$ = 0.98322-0.95
= 0.03322

Since 0.03322 > 0.001, another iteration is required.

Iteration 2:

Step 2-1: Re-estimate Re, using Ce_1:

$$Re_2 = \frac{\sqrt{2}}{\mu_6} Ce_1\ D_6\ Y \sqrt{\Delta P \rho_5}$$

$$= \frac{\sqrt{2}}{1.819E-05} (0.98311)(0.15)(0.998) \sqrt{(250)(1.14)}$$

$$= 193\ 165 \qquad \text{Eq. F-3 SI}$$

ERRATA SHEET FOR
ANSI/ASHRAE 51-1999 (ANSI/AMCA 210-1999)
Laboratory Methods of Testing Fans for
Aerodynamic Performance Rating

February 15, 2006

The corrections listed in this errata sheet apply to all printings of ANSI/ASHRAE Standard 51-1999 (ANSI/AMCA 210-99). The shaded items have been added since the previously published errata sheet dated January 10, 2006 was distributed.

Page	Erratum
56	**Appendix F. Iterative Procedure:** In the first paragraph, first sentence, change the section referenced from "9.3.2.7" to "8.3.2.7".
56	**Appendix F. Iterative Procedure:** In Equation F-1 (first column of page 56) the value of Y (expansion factor) should be calculated per Section 8.3.2.3, not Section 9.3.2 as indicated. Change "Y = 0.998 (calculate per Section 9.3.2)" to read "Y = 0.998 (calculate per Section 8.3.2.3)".
56	**Appendix F. Iterative Procedure:** In Equation F-2 I-P (second column of page 56) change the number in the square root in the denominator from "195595" to "189595". The calculated value of Ce_1 (= 0.98322) is correct.

$Re_2 = 199{,}397\ Ce_1$

$= 199{,}397\ 0.984$

$$Re_2 = \frac{1097}{60\ \mu_6}\ Ce_1\ D_6\ Y\ \sqrt{\Delta P\,\rho_5}$$

$$= \frac{1097}{(60)\,1.22E{-}05}(0.98322)(0.5)(0.998)$$

$$\sqrt{(1.005)(0.0711)}$$

$$= 196\ 225 \qquad \text{Eq. F-3 I-P}$$

Step 2-2: Recalculate C, using Re_2:

$$Ce_2 = 0.9986 - \frac{7.006}{\sqrt{Re_2}} + \frac{134.6}{Re_2}$$

$$= 0.9986 - \frac{7.006}{\sqrt{193\,165}} + \frac{134.6}{193\,165}$$

$$= 0.98336 \qquad \text{Eq. F-4 SI}$$

Check: $C_2 - C_1 = 0.98336 - 0.98311$

$= 0.00025$

Since $0.00025 < .001$, $C = C_2 = 0.98336$

$$Ce_2 = 0.9986 - \frac{7.006}{\sqrt{Re_2}} + \frac{134.6}{Re_2}$$

$$= 0.9986 - \frac{7.006}{\sqrt{196\ 225}} + \frac{134.6}{196\ 225}$$

$$= 0.98347 \qquad \text{Eq. F-4 I-P}$$

Check: $Ce_2 - C_l = 0.98344 - 0.98322$

$= 0.00025$

Since $0.00025 < 0.001$, $C = C_2 = 0.98347$

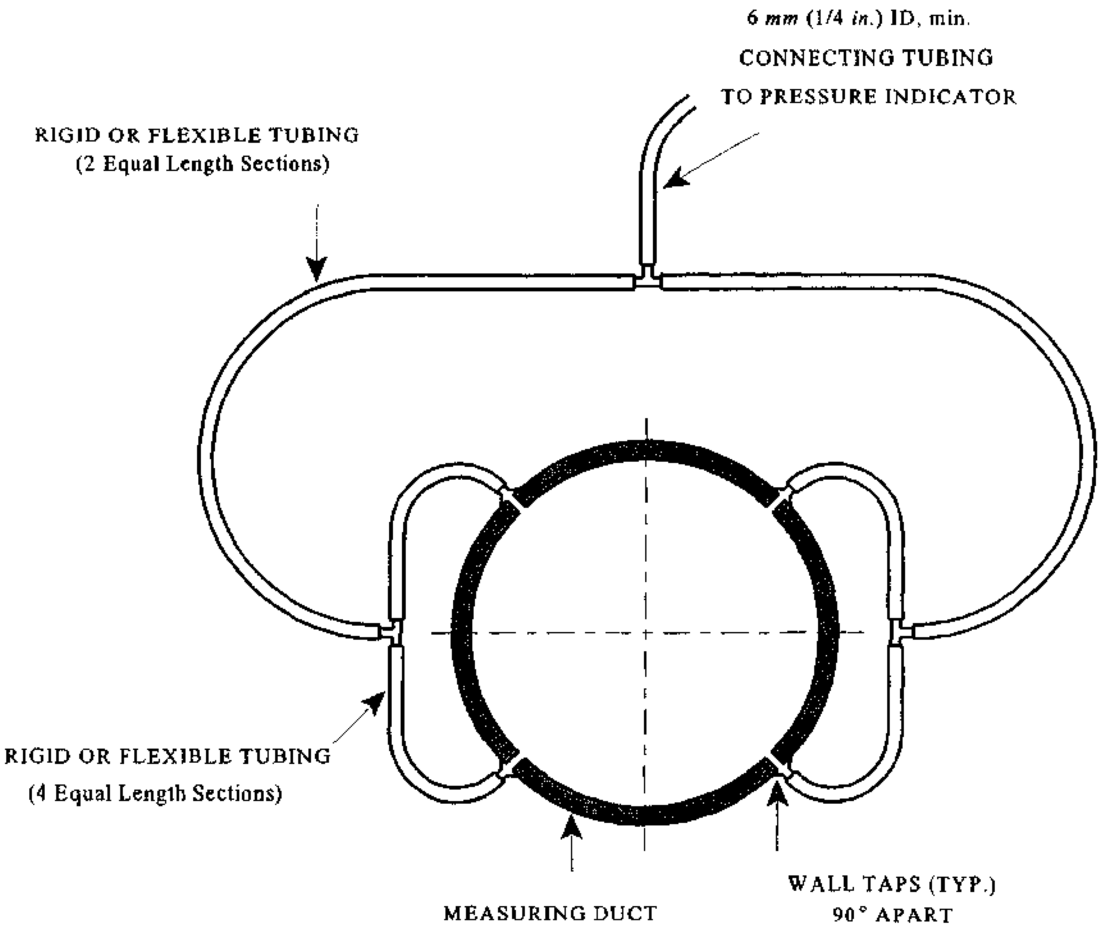
APPENDIX G

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ASHRAE Standard 51 but is included for information purposes.

APPENDIX G. TUBING

Large tubing should be used to help prevent blockage from dust, water, ice, etc. Accumulations of dirt are especially noticeable in the bottom of round ducts; it is recommended that duct piezometer fittings be located at 45° from the horizontal. Tubing longer than 1.5 *m* (5 *ft*) should be a minimum of 6 *mm* (1/4 *in.*) inside diameter to avoid long pressure response times. When pressure response times are long, inspect for possible blockage. Hollow flexible tubing used to connect measurement devices to measurement locations should be of relatively large inside diameter. The larger size is helpful in preventing blockage due to dust, water, ice, etc.

Piezometer connections to a round duct are recommended to be made at points 45° away from the vertical centerline of the duct. See Figure G-1 for an example.



NOTES:

1. Manifold tubing internal area shall be at least 4 times that of a wall tap.
2. Connecting tubing to pressure indicator shall be 6 *mm* (1/4 *in.*) or larger in ID.
3. Taps shall be within ± 13 *mm* (½ *in.*) in the longitudinal direction.

Figure G-1 Piezometer Ring Manifolding

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ASHRAE Standard 51 but is included for information purposes.

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This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ASHRAE Standard 51 but is included for information purposes.

APPENDIX I. HISTORY AND AUTHORITY OF ANSI/AMCA 210 - ANSI/ASHRAE 51, *LABORATORY METHODS OF TESTING FANS FOR AERODYNAMIC PERFORMANCE RATINGS*

This tenth edition of AMCA 210-ASHRAE 51 is the latest of a long series of Fan Test Codes that started with the first edition issued in 1923. Tradition has it that the first test code was developed as a result of problems encountered by the U. S. Navy with the performance ratings of fans being procured during World War 1. To resolve the issue of variations in testing methods which led to variations in performance ratings, a joint committee of the National Association of Fan Manufacturers (NAFM) and the American Society of Heating and Ventilating Engineers (ASHVE) was formed to develop a Standard Test Code for Fans.

The test code has been periodically reviewed and revised as problems arose and improvements to instrumentation and airflow measuring technology occurred. Over the years many people have made significant contributions to the development of this standard, some of whom are recognized in this history where appropriate. A brief outline of the significant changes that were made in each edition follows along with the complete Preface to the first edition.

Preface to the 1923 Edition of Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers

There are many instruments and methods existing today for use in testing of propeller fans, disc fans, and centrifugal fans and blowers. The different types of instruments used and the various methods or "setups" used have resulted in a wide variation in the data obtained.

Due to this lack of uniformity the Performance Tables and Characteristic Curves of fans and blowers have lacked uniformity and have not been comparable on a uniform basis.

Recognizing not only the desirability, but the necessity, of a standard method of testing fans and blowers in order to give them a proper rating, a Joint Committee was selected by the American Society of Heating and Ventilating Engineers and the National Association of Fan Manufacturers to prepare a Standard Test Code.

The Joint Committee was assisted by a sub-committee, composed of the Research Engineers of the member companies of the National Association of Fan Manufacturers.

The Joint Committee also cooperated with Committee No. 10 of the Power Test Code Committee of the American Society of Mechanical Engineers to the end that the respective codes would coincide where they covered the same ground.

The Standard Test Code for rating fans and blowers is offered to you with the belief that its use will be of marked benefit to both the manufacturer and user of fan and blower apparatus.

Joint Committee:

H. W. Page	B. F. Sturtevent Company
Prof. F. Paul Anderson (Secretary)	Director-Research Laboratory, American Society of Heating and Ventilating Engineers; Dean, University of Kentucky, College of Engineering
F. R. Still	American Blower Co.
C. A. Booth	Buffalo Forge Co.
E. M. Bassler	Bayler Manufacturing Co.
A. K. Owens	Tenny & Ohmes

Sub-Committee:

W. A. Rowe	American Blower Co.
J. C. Wolf	Bayley Mfg. Company
A. A. Criqui	Buffalo Forge Company
L. O. Monroe	Clarage Fan Company
E. D. Green	Garden City Fan Company
C. S. Messler	Green Fuel Economizer Co.
A. G. Sutcliffe	Ilg Electric Ventilating Co.
C. W. Rodgers	New York Blower Co.
H. F. Hagen	B. F. Sturtevent Company

The Test Code was revised in 1932 with the addition of provisions for testing fans with inlet boxes. In 1938 the Test Code was revised on the basis of research made with the help of various colleges of engineering, consulting engineers and other organizations. Flow straighteners were added to the test ducts, and the allowance for skin friction was reduced. Nomenclature changes relating to fan types and usage were made in the 1949 edition, which was published as NAFM Bulletin No. 110.

APPENDIX I

In 1955 NAFM was combined with the Power Fan Manufacturers Association (PFMA) and the Industrial Unit Heater Association (IUHA) to form the Air Moving and Conditioning Association (AMCA). One of the major concerns of this new organization was the accuracy and practicality of the Pitot traverse method of testing, and a committee was formed to study various test methods and to develop a new test code. To aid in this study AMCA sponsored research by the Battelle Memorial Institute to compare the test results using the Pitot tube test methods and nozzle test methods. The result of this effort was a new revision of the test code which was published in 1960 by AMCA as the AMCA Standard Test Code for Air Moving Devices, Bulletin 210. This fifth edition of the test code represented a major step forward in standard methods for testing fans and provides the underlying basis for all subsequent editions. The nozzle method of measuring air flow was recognized in this edition and the chamber-nozzle "setup" was developed and incorporated in the test code. Provision was also included for the effects of compressibility on fan performance.

The engineering committee that produced the fifth edition was composed of:
Tom Walters - American Standard Industrial Division
Bob Parker - Ilg Electric Ventilating Company
D. D. Herrman - Hartzell Propeller Fan Company
Hoy Bohanon - Acme Engineering & Manufacturing Co.

AMCA revised the Standard in 1960 to show arrangements for testing multiple outlet units. These changes, while minor, resulted in the sixth edition.

The seventh edition was published by AMCA in 1967 incorporating a number of changes in general format. International Standards were used as the basic units of measurement, and the symbols used throughout the code were consolidated into one table. As a result of AMCA sponsored research into the effectiveness of the Flow Straightener at unusual flow conditions, the cell size and length were changed, and a new derivation of the Compressibility Factor was added as an appendix.

AMCA Standard 210 became widely accepted and was virtually the only standard used in the United States and Canada since 1960. It had been widely accepted by producers, customers, and general interest groups, but no national consensus had ever been recorded. The Air Movement and Control Association (AMCA) asked The American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) to form a joint committee to facilitate obtaining national consensus. The Joint Committee first met on February 10, 1971 and decided that AMCA Standard 210-67 would be the starting point for development of a national consensus Standard. The following excerpts from the Foreword to the eighth edition published as AMCA Standard 210-74/ASHRAE Standard 51-75 provides an overview of the detailed review and revision of AMCA Standard 210-67.

Excerpts from the Foreword to AMCA Standard 210-74-/ASHRAE Standard 51-75:

The provisions of AMCA Standard 210-67 were subjected to critical review. The progress of the TC 117 Committee of the International Organization for Standardization (ISO) on methods of testing industrial fans, was monitored. The literature, particularly that on flow and pressure measurement, was searched. The Joint Committee recommended experimental work which was conducted in the AMCA Laboratory. All of this influenced the final content of this standard.

Some of the more significant differences between this standard and its predecessor, AMCA Standard 210-67, are:

(1) The style was changed to reflect American National Standards Institute (ANSI) recommendations regarding page format, abbreviations, symbols and subscripts, and general arrangement.

(2) The content of the standard is limited to matters directly related to testing. Other information, including the application of the fan laws for rating purposes, is contained in the appendices.

(3) The scope has been broadened by eliminating the upper limit on compression ratio. The scope has been narrowed by limiting the test gas to air.

(4) The units of measurement for gas properties are based on mass rather than weight. Water gauge is based on 68°F and includes a gas column balancing effect.

(5) The definitions have been expanded to include total temperature, head and compressibility coefficient.

(6) Test setups have been given new numerical designations.

(7) Performance specifications, as well as equipment specifications, are given for instruments and methods of measurements.

(8) A log-linear Pitot traverse method has been substituted for the equal area method.

(9) Data for nozzles have been expanded.

(10) Specifications for chamber size and chamber settling means have been changed.

(11) Calculation methods have been revised with respect to duct friction, straightener loss, compressibility coefficient, and conversion formulae. Calculation methods are presented in a manner to facilitate either manual or automatic data processing.

(12) Both the International System of Units (SI) and other metric units are treated in an appendix.

(13) The fan total efficiency equation for compressible flow is derived in an appendix. This is done in terms which eliminate the need for an iteration procedure.

(14) New compressible-flow fan laws are derived in an appendix.

(15) An error analysis method is derived in an appendix.

While each of the above changes is significant, the basic procedures of AMCA Standard 210-67 have been retained.

The Joint ASHRAE-AMCA Project Committee was composed of the following members:

Robert Jorgensen, Chairman	ASHRAE-AMCA
Kenneth W. Burkhardt, Secretary	AMCA
Nestor Brown, Jr.	ASHRAE
Hoy R. Bohanon	ASHRAE-AMCA
Harold F. Farquhar	ASHRAE-AMCA
Linn Helander	CONSULTANT
Donald D. Herman	AMCA
John G. Muirheid	ASHRAE
Allen C. Potter	ASHRAE
Wendell C. Zeluff	AMCA

E. A. Cruse, ASHRAE Standards Committee Liaison

In 1977 AMCA Standard 210-74/ASHRAE Standard 51-75 was granted American National Standards Institute status becoming ANSI/AMCA 210/ASHRAE 51-75.

The standard was reviewed by another Joint Committee starting in 1982 and was reaffirmed with minor revisions. Installation types corresponding with those in the British Standard BS848: Part 1:1980 and the draft ISO Standard (5801) were included, along with slight changes to the Inlet Bell and Settling Means. The appendix was changed to reflect the more acceptable term "uncertainty" as opposed to "error" and portions of the derivation were modified.

The ninth edition of the standard was published as ANSI/AMCA Standard 210-85 - ANSI/ASHRAE Standard 51-1985.

The Joint ASHRAE-AMCA Project Committee was composed of the following members:

Robert Jorgensen, Chairman	ASHRAE-AMCA
Gordon V.R. Holness, Vice Chairman	ASHRAE
Hoy R. Bohanon	ASHRAE-AMCA
A. Michael Emyanitoff	ASHRAE
Daniel Fragnito	ASHRAE
James W. Schwier	ASHRAE-AMCA
Thomas A. Hirsbrunner	ASHRAE-AMCA
Gerald P. Jolette	ASHRAE-AMCA

Since AMCA 210-67 was published, this standard has grown in recognition not only in the United States and Canada but also internationally. The provisions of this standard were included almost in its entirely in the International Standard, ISO 5801. The Foreword to this the tenth edition outlines the minor changes made to this edition. In the coming years the standard will again be reviewed and revised to reflect improvement to the technology of flow and pressure measurement, but it is always well to understand the foundation on which this standard was built and the people who contributed to its construction.

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Interpretation IC-51-1999-1 of
ANSI/ASHRAE Standard 51-1999/AMCA Standard 210-1999
Laboratory Methods for Testing Fans for Aerodynamic Performance Rating

February 18, 2000

Reference: This request refers to ANSI/ASHRAE 51-1999/AMCA 210-1999, Sub-clause 6.3.3

Background: The second paragraph of 6.3.3 reads:

"When a measuring plane is located downstream of the settling means, the settling means is provided to ensure a substantially uniform flow ahead of the measuring plane. In this case, the maximum velocity at a distance 0.1M downstream of the screen shall not exceed the average velocity by more than 25% unless the maximum velocity is less than 400 feet per minute."

Interpretation No. 1: Requestor's letter opines that this check is required at the design airflow (400 fpm x cross sectional area of the chamber) rating of the chamber.

Question No. 1: Is Interpretation No. 1 correct?

Answer: No.

Comment: The requirement must be met at all airflows used during the test.

Interpretation No. 2: Requestor's letter opines that this check is not required at intermediate air quantities if the check passes at the design airflow rating of the chamber.

Question No. 2: Is Interpretation No 2 correct?

Answer: No.

Comment: The check is required at all airflows used during the test.

Question No. 3: If the answer to Question No. 2 is NO and the intermediate air quality check is required, must more than one combination of nozzles be checked?

Answer: Yes.

Comment: Any combination of nozzles used during the test must be checked.

Interpretation No. 3: Requestor's letter opines that 30 points spaced in accordance with the log Tchebycheff rule for rectangular ducts (1993 ASHRAE Handbook, Figure 6, page 13.15) is adequate for this check.

Question No. 4: Is Interpretation No. 3 correct?

Answer: Yes.

Interpretation No. 4: Requestor's letter opines that an Electronic Air Velocity Indicator with an accuracy of ±3% (of reading) or ±10 fpm, whichever is greater, meets the requirements of this standard.

Question No. 5: Is Interpretation No. 4 correct?

Answer: Provided calibration records are in order.

Interpretation No. 5: Requestor's letter states that the local velocity at each point in the measurement fluctuates and opines that the velocity to be recorded is the average velocity, not the maximum velocity.

Question No. 6: Is Interpretation No. 5 correct?

Answer: Yes.

Comment: All local velocities have to be averaged in time as described in 6.2.1.2 of the standard.

Interpretation No. 6: Requestor's letter opines that the check is done with the control fan only (without a test fan).

Question No. 7: Is Interpretation No. 6 correct?

Answer: No.

Comment: The requirements must be met with the test fan operating. The control fan may or may not be operating.

Interpretation No. 7: Requestor's letter opines that the average velocity in 7.3.3 is that calculated by $V\ (fpm) = Q\ (cfm) / A$ *(cross sectional area of chamber,* ft^2*)*

Question No. 8: Is Interpretation No. 7 correct?

Answer: Yes.

General Comment: The standard is written to cover the requirements for a test. Since a test must cover a range of airflow rates, any requirement predicated on airflow rate, such as average velocity or maximum velocity, must be met for the full range of airflow rates. For instance, if there are eight determinations made during a test, the requirement must be satisfied for each determination. Any test utilizing equipment based on Figures 9, 10, 11 or 15 must meet the requirements of the paragraph cited in this interpretation at any and all airflow rates being measured. There is no single design airflow rate for the apparatus according to this standard. The maximum velocity referred to in sub-clause 7.3.3 is the maximum of the various readings across the face of the settling means during a determination.

Interpretation IC 51-1999-2 - September 12, 2000
ANSI/ASHRAE STANDARD 51-1999 and ANSI/AMCA Standard 210-99
Laboratory Methods of Testing Fans for Aerodynamic

Reference: This request refers to Interpretation IC 51-1999-01 of ANSI/ASHRAE 51-1999 and ANSI/AMCA 210-99 dated February 18, 2000.

Background: Requestor cites the second paragraph of subclause 6.3.3,

> "When a measuring plane is located downstream of the settling means, the settling means is provided to insure a substantially uniform flow ahead of the measuring plane. In this case, the maximum velocity at a distance 0.1M downstream of the screen shall not exceed the average velocity by more than 25% unless the maximum velocity is less than 400 feet per minute."

and the following interpretations of this paragraph, provided in IC 51-1999-01:

> "(a) The requirement must be met at all airflows during a test."
>
> "(b) The check is required at all airflows used during the test."
>
> "(c) Any combination of nozzles used during the test must be checked."
>
> "(d) The requirements must be met with the test fan operating. The control fan may or may not be operating."
>
> "(e) The standard is written to cover the requirements for a test. Since a test must cover a range of airflow rates, any requirement predicated on airflow rate, such as average velocity or maximum velocity, must be met for the full range of airflow rates. For instance, if there are eight determinations made during a test, the requirement must be satisfied for each determination. Any test utilizing equipment based on Figures 9, 10, 11 or 15 must meet the requirements for the paragraph cited in this interpretation at any and all airflow rates being measured. There is no single design airflow rate for the apparatus according to this standard. The maximum velocity referred to in subclause 6.3.3 is the maximum of the various readings across the face of the settling means during a determination."

Question No. 1: If the above requirement is to be met at all airflow determinations, is it required to conduct a qualification test for **every fan test** to verify compliance with 6.3.3?

Answer: Yes, except when a qualification procedure has been agreed to as noted in the General Comment, below.

Question No. 2: If the above requirement is met by conducting tests at all airflows to verify compliance with 6.3.3, is the qualification data to be reported with every fan test report?

Answer: Yes, except when a qualification procedure has been agreed to as noted in the General Comment, below.

Question No. 3: Requestor opines that a laboratory pre-qualification test using a test fan having sufficient airflow capacity could be used as a basis for all other fans to be tested, thereby eliminating the check-test requirement for every fan at all airflow determinations. This way the total time to test fans would be within reasonable time periods. Are one-time pre-qualification tests acceptable to meet the requirements of subclause 6.3.3?

Answer: No, except when a qualification procedure has been agreed to as noted in the General Comment, below.

General Comment:

There is no provision in the Standard for general qualification of a test facility to meet the requirements of subclause 6.3.3.

The parties to a test may agree on a procedure to qualify the test facility over a range of airflows, nozzle combinations, outlet velocities of the test fan, supply fan and nozzles that meet the requirements of the standard. The parties to a test may further agree that any test performed within the qualification range of the test facility would be deemed to have met the requirements of the Standard and would not require additional checks.

Addendum a-2001
to
ASHRAE 51-1999 / AMCA 210-99

Approved by American National Standards Institute 21 August 2001

In revising the 1985 edition, one revision produced Note 4 of Figure 4A. The intended result of the revision was to specify a dimensional measurement location that would be suitable for checking by a state-of-the-art measurement device. The unintended effect of this revision was the tightening of the tolerance on that dimension beyond that which is possible to achieve by existing and customary commercial fabrication processes. This went unnoticed during the review process. To rectify this oversight, it is necessary to revert to the requirement as given in the 1985 edition:

"4. The nozzle throat shall be measured (to an accuracy of $0.001D$) at the minor axis of the ellipse and the nozzle exit. At each place, four diameters — approximately 45° apart must be within $\pm 0.002D$ of the mean. At the entrance to the throat the mean may be $0.002D$ greater, but no less than, the mean at the nozzle exit."