TESTING OF FANS, FAN CLASSIFICATION, PERFORMANCE & SELECTION

A Dissertation Submitted in Partial Fulfilment of the Requirements
for the Degree of
MASTER OF ENGINEERING
IN
(APPLIED THERMODYNAMICS (Refrigeration and Air Conditioning)

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C E R T I F I C A T E

CERTIFIED that the Dissertation entitled "TESTING OF FANS, FAN CLASSIFICATION, PERFORMANCE AND SELECTION" which is being submitted by Shri Ajmer Singh as a partial fulfilment of the requirement for the Degree of Master of Engineering of the University of Roorkee is a record of bonafide work carried out by him under my supervision and guidance. The results embodied in this Thesis have not been submitted for award of any other Degree or Diploma.

This is to certify further that he has worked for a period of six months for the Master of Engineering Degree Thesis at the University of Roorkee.

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Dated: Jan. 13, 1964.

A C K N O W L E D G E M E N T

I am indebted to Shri J. Prasad, Reader in Mechanical Engineering Department, University of Roorkee, for his valuable guidance, full co-operation and encouragement in the preparation of this thesis.

My thanks are also due to the Mechanical Engineering and Electrical Engineering Departments for co-operation in the manufacture of apparatus and the facilities provided during the experimentation.

C O N T E N T S

PREFACE.

Theoretical knowledge is incomplete without practical background. The main object of the dissertation work is the testing of fans for their performance. But the study would have remained incomplete without including some other topics associated with it. Therefore chapters of fan classification and performance, and fan selection have been included. Some information which was necessary, but could not be included in the main body of the text has been given in Appendix.

LIST OF SYMBOLS

Q = Volume flow rate.

H = Pressure head.

P = Power input to fan.

p = Pressure.

L = Length.

D = Diameter.

R = Radius.

A = Cross-sectional area.

N = Fan R.P.M.

V = Velocity of fluid.

t = Temperature.

T = Absolute temperature.

ρ = Density of fluid.

η = Efficiency.

C_d = Coefficient of discharge.

m = Ratio of orifice area to duct area.

INTRODUCTION

Moving air or gas from one place to another place is very important engineering problem. The machine which performs this duty is the fan. For moving a fluid creation of pressure difference is necessary and the fan raises the pressure of air or gas as it passes through it. There are other machines like blowers and compressors which also perform the same duty. The name fan applies to cases when the pressure rise of air or gas is equal to or less than one pound per square inch. This study relates only to fans.

CLASSIFICATION

Basically the fans can be classified into two groups (1) Centrifugal and (2) Axial flow. The various types of centrifugal fans such as forward curved blade, radial blade, backward curved blade, radial tip blade and backward straight blade have been included in Chapter I. The various types of axial flow fans (1) Propeller fans, (2) Tube axial fans and (3) Vane axial fans have also been discussed.

Some special designs such as mixed flow impeller fan, and axial flow fans with airfoil blade section have been introduced.

FAN PERFORMANCE

The fan performance curves for some representative fan types have been given in figures 2 to 7, and the special features regarding them have been discussed at length in Chapter I.

FAN SELECTION

When selecting a fan two factors are to be considered (1) Type of the fan and (2) Size of the fan. The suitability of various types of fans for various applications has been discussed in Chapter II. Further the procedure for selecting the size of fan from rating tables has been explained. The rating tables are prepared from fan performance curves. The use of dimensionless groups in preparing the rating tables has been described in Chapter II. The rating tables are prepared for air having standard density (0.075 lbs./cu.ft.). The use of rating tables in selection of fan to deliver air or gas having non-standard density has been explained. Further the effect of change in air density and fan speed for a given fan and system has been explained.

TESTING OF FAN PERFORMANCE

Tests are conducted on fans for two purposes (1) For improvement in design and (2) For issuing the certificates of their performance. The fan should be tested under similar conditions in which it is to be installed in the field. The various test set-ups (as recommended by A.S.M.E.) have been given in figures 11 to 15 and the various parts of equipment have been described in Chapter III. The various definitions and the procedure of calculations has also been included in Chapter III. According to various test codes the air flow measurement is made by pitot tube traverses. The use of orifices and nozzles in air flow measurement has also been discussed. The use of a special circular disc slide rule for calculating air density has been described in Chapter III. Reference has also been made to an automatic plotter (used by vicking air products) which plots the fan performance curves automatically. The methods of testing the fan performance in the field have also been incorporated in Chapter III.

EXPERIMENTAL SET-UP

The experiment was set up for testing an axial flow fan to draw its performance curves. Figures 21-26 include the various sketches of the equipment, ~~and~~ the description of the equipment, procedure of test, observations required, and method of calculations ~~all~~ are given in Chapter IV.

APPENDIX

It consists of the various readings taken by the author in his experimentation and some specimens of calculations. It also includes the procedure ~~of~~ for testing the axial flow type A.C. ventilating fans as prescribed by Indian Standards. The discussion about the standardization of pitot tube has also been included in the end.

CHAPTER I

CHAPTER - I.

FANS CLASSIFICATION AND PERFORMANCE

I.(a) CLASSIFICATION

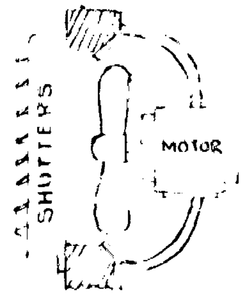
The fans can be classified into two main groups (1) Centrifugal or radial flow and (2) Axial flow.

CENTRIFUGAL FANS

The centrifugal fans consist of a fan wheel rotating within a scroll shaped housing. They are further sub-divided according to the relative position and shape of blades in the rotor wheel. The different types of centrifugal fan blades are (1) Forward curved fig. 1(d), (2) Straight or radial fig. 1(e), (3) Backward curved fig. 1(f), (4) Radial tip fig.1(g), (5) Backward flat fig. 1(h) etc. The centrifugal fans are designed to deliver air or gas over wide range of volume. Usually the static pressure developed is upto 25 inches of water but in specially designed fans it goes upto 90 inches of water. The fan housing may be sheet or cast metal, with or without any protective coat of rubber, lead, enamal etc. The drive may be through a belt or directly. (Ref. 1)

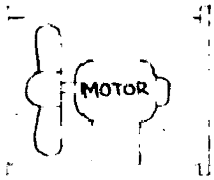
PROPELLER FANS

The axial flow fans consist of two or more blades mounted on a shaft. The flow of air is parallel to the wheel shaft axis. The basic type among axial flow fans is the propeller fan Fig. 1(a). In a propeller fan the blades may be spinning free



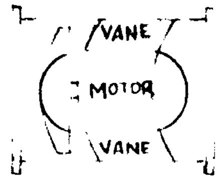
PROPELLER

FIG. 1 (a)



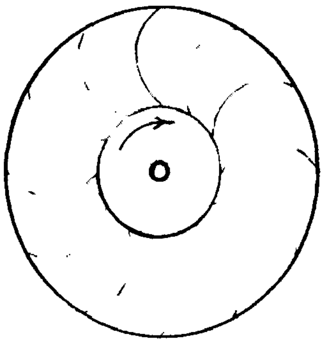
TUBE AXIAL

FIG. 1 (b)



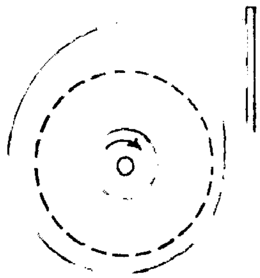
VANE AXIAL

FIG. 1 (c)



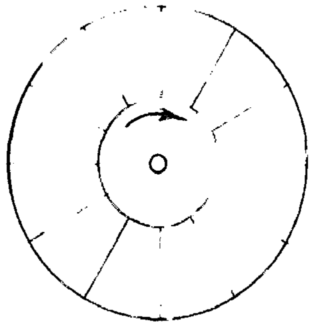
FORWARD CURVED

FIG. 1 (d)



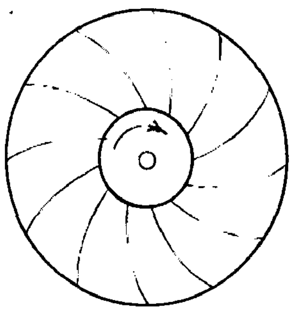
CENTRIFUGAL FAN

FIG. 1 (k)



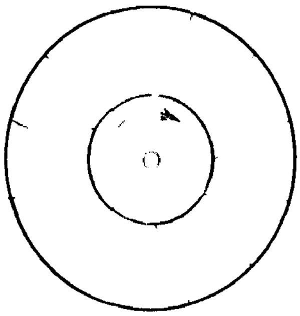
RADIAL BLADED

FIG. 1 (e)



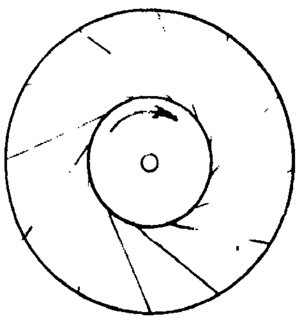
BACKWARD CURVED

FIG. 1 (f)



RADIAL TIP BLADED

FIG. 1 (g)



BACKWARD FLAT BLADED

FIG. 1 (h)

or are mounted ~~in~~ on a ring or plate which is generally attached to the wall. They are designed to move air from one large enclosed space to another or from indoors to outdoors or vice versa. They can deliver air ina wide range of volumes at low pressures (less than one inch water static). They may be provided with automatic shutters on discharge side, but the shutters is not a part of the fan. It protects fan from wind rain, snow etc. Mostly they use direct drive but belt drive is also used. (Ref. 1)

The propeller fan can deliver large volumes of air but cannot develop sufficient pressure. Hence it is not considered to be a true axial flow type fan. There are some propeller fans installed in ducts which give upto or above one inch static pressure. They are called semi-pressure or duct fans. They are in between a propeller fan and the true axial flow fan. (Ref. 2)

TUBE AXIAL FANS

The tube-axial fans consist of the fan placed in a cylindrical duct. The hub has relatively larger diameter and helical blades. The blades may be of uniform thickness or can have airfoil section. (Ref. 3).

The tube axial fans can move wide range of air or gas volumes at medium pressures (0.25 to 2.5 inches water static). The drive may be direct or by means of belt. (Ref. 1)

DISCUSSION

According to Ref. 2 the semi-pressure or duct fans are just propeller fans placed inside a duct. And according to Ref. 3 the tube axial fans also consist of the fan placed in the duct.

The actual difference between the two types is that of their construction. The duct fans are simple propeller fans placed in the duct but the tube axial fans have larger hub diameters and more inclination of blades.

VANE AXIAL FANS

The vane-axial fans are similar in construction to tube axial fans with only addition of guide vanes which are attached before or after the impeller. The addition of vanes eliminates spin of inlet or outlet air as the case may be and increases the fan static pressure. Thus the addition of vanes results in increased pressure and efficiency of fan. It delivers air over wide range of volumes and pressures. The common pressure range is from 0.5 to 6.0 inches of water. Special designs can develop pressures upto or above 60 inches of water. The wheel may be made from sheet metal or may be a cast. Both belt and direct drives are used. (Ref. 1)

AXIAL FLOW FANS (AIRFOIL BLADE SECTION)

The axial flow fans with airfoil section blades can be used to develop relatively higher pressures. The number of blades is generally two. Lionel S. Marks Professor of Mechanical Engineering, Harvard University had designed an axial flow fan of this type with 8 blades. The fan was able to develop 10 inches water static pressure at no delivery condition at 3600 R.P.M. The static pressure developed at the same speed in the working range was in between 5 and 6 inches of water. (Ref. 4)

A propeller fan with airfoil blade section was also designed by C.E. Peck (Engineer, Power Engineering Department, Westinghouse electric and Manufacturing Co.) and M.D. Ross (Design Engineer, Westinghouse Electric and Manufacturing Co.) for cooling electrical machines. Here space was a consideration. The fan consisted of six blades and was capable to develop 25 inches water static pressure at no delivery and about 15 inches water static pressure in the range of maximum efficiency. (Ref. 5)

MIXED FLOW IMPELLER FAN

The Torrington Manufacturing Company has produced a fan in which the intake edge of the impeller resembles propeller fan blades and remaining portion of the impeller resembles to that of centrifugal fans. This has been named as mixed flow impeller fan. Its operation is in between that of an axial flow fan and a centrifugal fan. It is particularly suitable for applications where the pressure requirement is too low from efficient range of centrifugal fans and too high from the efficient range of axial flow fans. Most of the air cooling and heating appliances operate in this range, Since the system will be operating at higher efficiency, hence there will be lesser noise. (Ref. 6)

I. (b) FAN PERFORMANCE

The fan performance as defined by A.S.H.R.A.E. is a statement of volume, total pressures, static pressures, speed, power input, mechanical efficiency and static efficiency at a stated density. (Ref. 3)

The various terms as used in fan performance have been defined by the A.S.M.E. as follows. The definitions by other associations are also similar.

Standard air has a density of 0.075 pounds per cubic foot. This is based on air at 68F, 29.92 inches Hg, and 50 percent relative humidity. (Ref. 7)

Capacity of a fan is the volume rate of flow of air at the fan inlet at any air density and at specified fan speed. (Ref. 7)

Discharge Velocity pressure is the pressure corresponding to the calculated average velocity at the fan outlet at specified inlet air density and fan speed. (Ref. 7)

Fan Total pressure is the rise in total pressure between fan inlet and fan outlet at specified inlet air density and fan speed. (Ref.7)

Fan static pressure is the fan total pressure minus the discharge velocity pressure at specified inlet air density and fan speed. (Ref.7)
~~(Ref. 7)~~

Power input is the power supplied to the fan shaft at specified inlet air density and fan speed. (Ref. 7)

Mechanical Efficiency (or total efficiency) is the ratio of the power output of the fan, based on capacity and fan total pressure, to the shaft power input. (Ref. 7)

Static Efficiency is the ratio of the power output of the fan, based on capacity and fan static pressure, to the shaft power input. (Ref. 7)

PLOTTING OF FAN PERFORMANCE CURVES

The fan characteristic curves show so as to how the fan will operate when delivering different volumes of air against different resistances to air flow. For plotting the fan characteristic curves, the volume delivered by the fan is plotted along x-axis, with the fan static pressure, total pressure, power input, static efficiency and total efficiency, all plotted along y-axis. There are two methods of plotting these values. According to first method, the volume delivered by the fan is plotted along x-axis represented as percentage of free delivery volume. The fan static pressure and the fan total pressure are plotted along y-axis represented as percentage of the fan pressure at no delivery. The Horse Power is plotted along y-axis as percentage of the maximum Horse Power in the range. The static and total efficiencies are also plotted along y-axis. The fan characteristic curves are also called fan performance curves. Figures 2 to 7 represent performance curves for few selected types of fans, plotted according to the above method.

According to second method the volume actually delivered by the fan is plotted along x-axis and the actual values of fan static pressure, total pressure, power input and efficiencies are plotted along y-axis.

When the performance curves are plotted according to first method then the same curves will represent the performance of all the different sizes of fans of a particular class. This is so, because all the fans of a particular class are made geometrically similar.

The curves plotted by the second method will only be for one particular fan. The advantage of plotting the curves in this manner will be that the actual values of the fan static pressure, total pressure, horse power and efficiencies corresponding to different volumes delivered by the fan will be available.

(1) STRAIGHT OR RADIAL BLADE

The discharge pressure rises continuously from free delivery end to a maximum near no delivery and falls off after it. The H.P. input increases continuously with increase in volume. The maximum static efficiency is nearly at maximum pressure. (1)

(2) FORWARD CURVED BLADE

The fan pressure rises from free delivery end, reaches a maximum value, then droops before reaching the no delivery end. The maximum efficiency occurs at about maximum pressure. Horse power input rises more rapidly with increase in capacity. (1)

BACKWARD CURVE BLADE

The fan pressure rises from free delivery end to nearly no delivery end with little drop near no delivery end. The horse power first rises with increase in volume, reaches a maximum and then starts decreasing. The maximum value of power is near the working range of fan. Thus the Horse Power characteristic is self limiting. The efficiency is also maximum near the maximum Horse Power. (1)

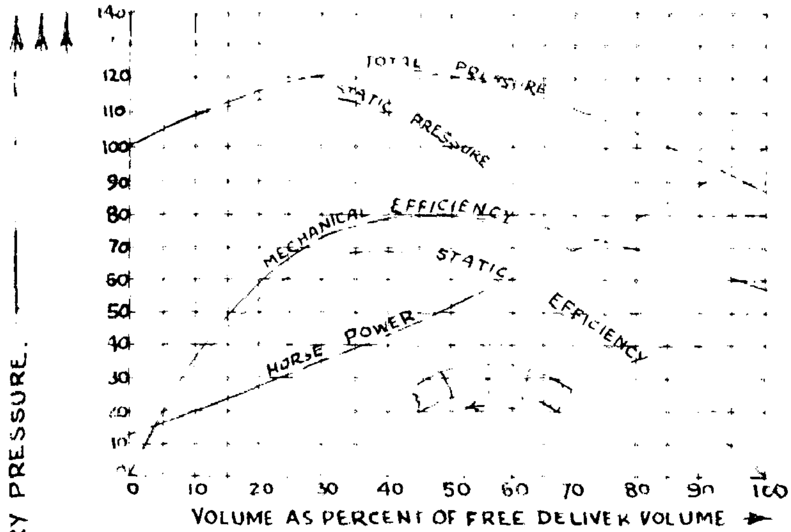


FIG.2 STRAIGHT BLADE

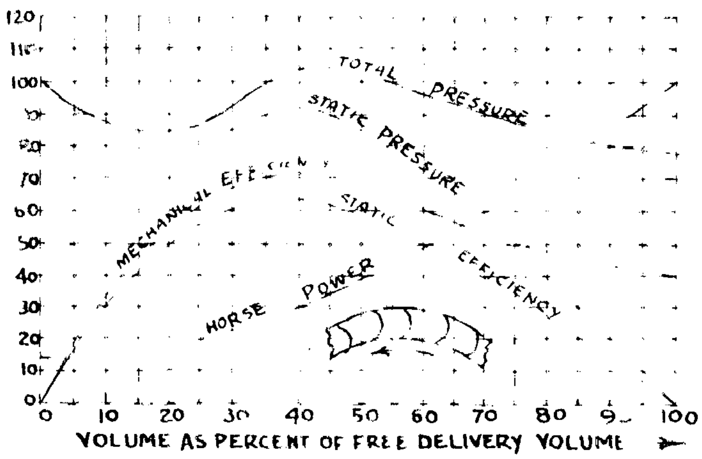


FIG.3 FORWARD CURVED BLADE

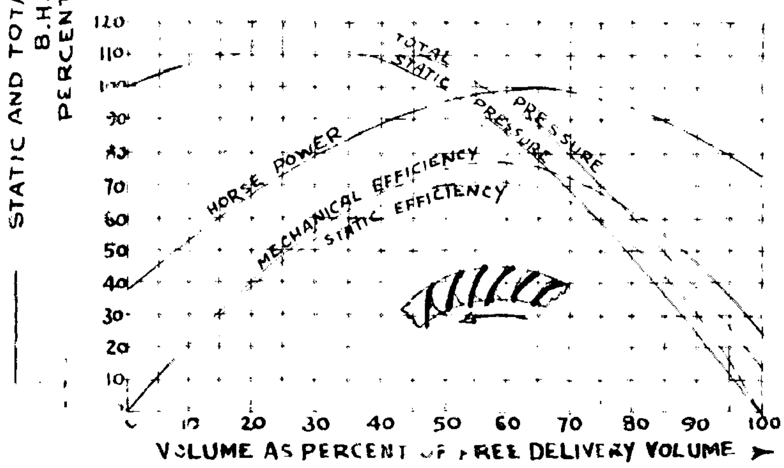


FIG.4 BACKWARD CURVED BLADE

~~The efficiency is also maximum near maximum Horse Power. (Ref. 1)~~

RADIAL TIP BLADE

The pressure characteristic and horse power characteristic is similar to radial or straight blade fans. But it has relatively higher efficiencies and permits the use of shorter blades. Shock losses at wheel entrance are also reduced. (Ref. 1)

BACKWARD FLAT BLADE

They are just a modification of backward curved blades. In this case the pressure characteristics become less steep and the maximum efficiencies are available over wider range. (Ref. 1)

AXIAL FLOW FANS

Basically the axial flow fans are a higher capacity type. They can be designed with widely varying characteristics. Generally the pressure rises continuously from free delivery end to no delivery end. In case of tube axial and vane axial fans there may be a drop in pressure characteristic. But the vane axial and tube axial fans can also be designed to give performance curves resembling somewhat to that of backward curved bladed centrifugal fans.

Depending upon design the H.P. curve may be flat with a self limiting characteristic as in backward bladed centrifugal fans, or it may have a general downward trend from no delivery to free delivery with the maximum at no delivery. In a vane axial fan the horse power curve also depends upon the shape of vanes.

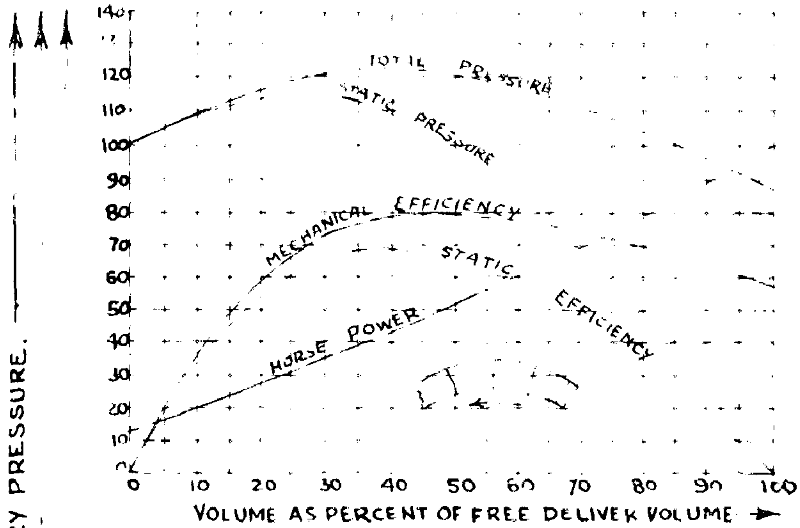


FIG.2 STRAIGHT BLADE

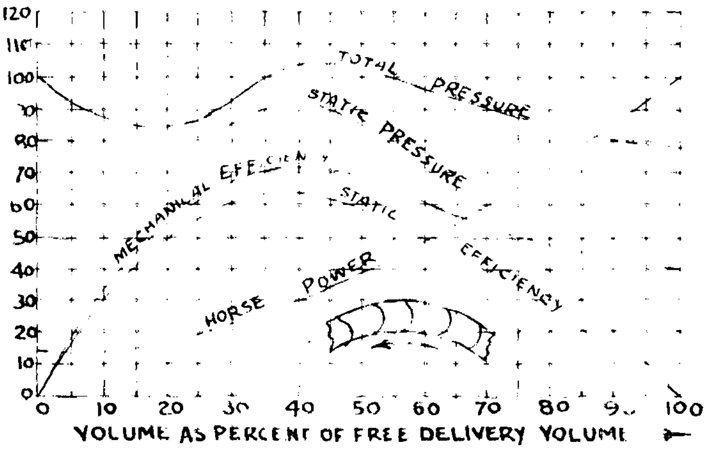


FIG.3 FORWARD CURVED BLADE

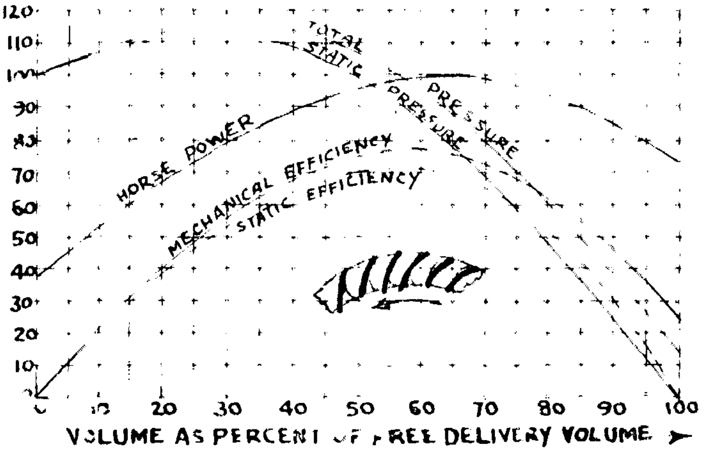


FIG.4 BACKWARD CURVED BLADE

STATIC AND TOTAL PRESSURES AS PERCENTAGE OF NO DELIVERY PRESSURE.
B.H.P. AS PERCENTAGE OF MAXIMUM B.H.P.
PERCENT STATIC AND TOTAL EFFICIENCY.

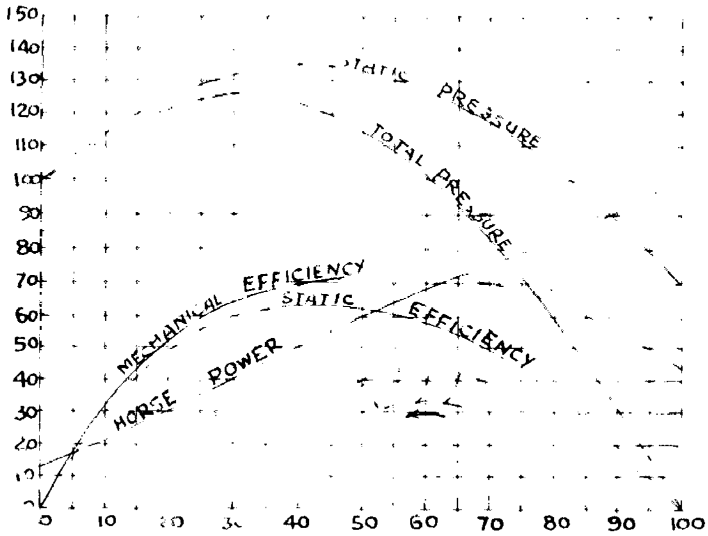


FIG.5 RADIAL TIP BLADE

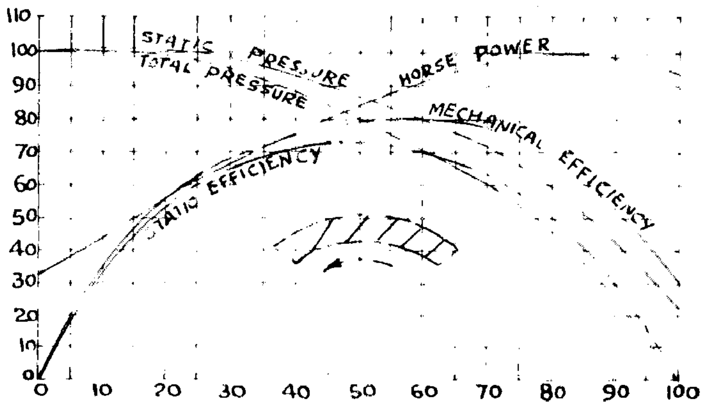


FIG.6 BACKWARD FLAT BLADE

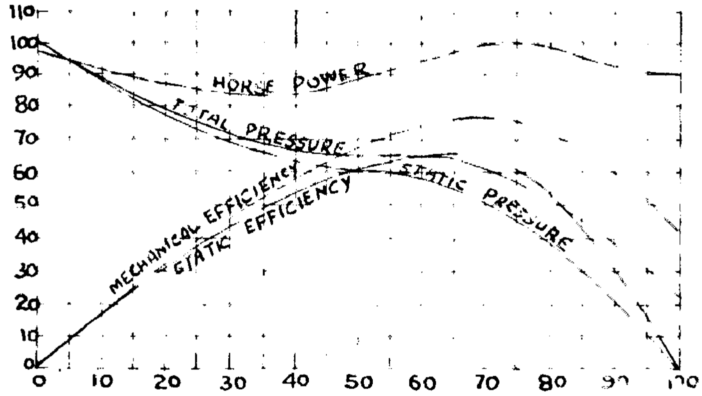


FIG.7 AXIAL FLOW

~~The efficiency is also maximum near maximum Horse Power. (Ref. 1)~~

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Depending upon design the H.P. curve may be flat with a self limiting characteristic as in backward bladed centrifugal fans, or it may have a general downward trend from no delivery to free delivery with the maximum at no delivery. In a vane axial fan the horse power curve also depends upon the shape of vanes.

The maximum efficiency tends to occur at a higher percentage of free delivery capacity than that for a centrifugal fan.

The propeller fans are generally designed for operation near free delivery. Generally the static pressure rises continuously from free delivery to no delivery. The Horse Power is highest near no delivery and falls off towards free delivery. The maximum total efficiency generally occurs at higher percentage of free delivery as compared to other types. (Ref. 3)

SUMMARY OF CHAPTER I.

(a) Fan Classification

The two main classes of fans are (1) Centrifugal and (2) Axial flow. The centrifugal fans have further been classified according to shape and position of blades as forward curved bladed, backward curved bladed, radial bladed etc. The axial flow fans have been further sub-divided as propeller fans, tube axial fans and vane axial fans. The propeller fans are without ducts and generally deliver air from one large space to another large space against low pressures (less than 1 inch water). The tube axial fans are installed in the ducts and have relatively large efficiency than propeller fans and can operate against higher pressures (0.25-2.5 inches water). Vane axial fans have higher efficiency and higher pressure range than tube axial fans. The pressure range is from (0.5 - 6.0 inches water). Propeller fans with airfoil blade section can develop larger pressures(The working pressure may be upto 15 inches of water)and have higher efficiency. Torrington Company has manufactured a fan which combines the features of both axial flow and centrifugal fans. This fan has been named as mixed flow impeller fan. This can be used for pressures too high from the efficient range of axial flow fans and too low from the efficient range of centrifugal fans.

(b) Fan Performance.

The fan performance may be defined as a statement of volume, static pressures, total pressures, speed, power input, mechanical efficiency and static efficiency at a stated density. The performance of the fans is shown by fan characteristics or fan performance curves. There are two methods of plotting the fan performance curves. According to first method the volume delivered by the fan is taken along x-axis and represented as percentage of of free delivery volume. The fan static and total pressures are taken along y-axis represented as percentage of no delivery pressure. The Horse Power is taken along y-axis and represented as percentage of maximum power. The static and total efficiencies are also plotted along y-axis. The fan performance curves plotted according to above method for few representative fan types have been given in figures 2-7, and the special features of curves of each type have been discussed.

According to second method the actual volume delivered by the fan is plotted along x-axis with the actual values of fan static pressure, total pressure, power input and efficiencies plotted along y-axis.

When plotted according to first method then the same curves represent the performance of all the different sizes of fans of a particular class. Performance curves plotted according to second method give the actual magnitudes of volume, pressures horse-power, and efficiencies given out by any one particular fan.

C H A P T E R - II.

FAN SELECTION

While selecting a fan two things are to be considered (1) type of fan and (2) size of fan. The selection of type of fan is a matter of experience. After the fan type has been decided the fan size can be selected from the fan rating tables published by the manufacturers. (Ref. 8)

SELECTION OF FAN TYPE

Generally the selection of fan type depends upon services to which the fan is to be put. When the delivery is against negligible resistance without a duct system then propeller fan is the selection for convenience and low cost. With low resistance the power consumption is also low and hence efficiency becomes of secondry importance. When the duct system is involved then there is selection between, tube axial, vane axial and centrifugal fans. (Ref. 3)

ADVANTAGES OF AXIAL FLOW FANS

The axial flow fans ~~have more simplicity~~ are simpler in construction, have lighter weight~~s~~, ~~hence~~ need more light supporting structures, occupy less space than centrifugal fans, & their cost is ~~very~~ lesser ~~as compared~~ to centrifugal fans of the same capacity. (Ref. 2) Out of the tube axial and vane axial fans, the vane axial fans have a higher efficiency, less noise but occupy more space and are more costly. (Ref. 3)

DISADVANTAGES OF AXIAL FLOW FANS

The disadvantages of axial flow fans are that they are very noisy (vane axial fan creates lesser noise) and cannot handle solid materials. In case of direct connected fans the difficulty arises, when the gas to be handled is corrosive or at high temperature. ~~For this,~~ In such cases, sometimes the motor is sometimes installed in an elbow to protect it from air stream. Some manufacturers install the motor outside the duct and provide V-belt drives. (2)

APPLICATIONS OF AXIAL FLOW FANS

In short it can be said that axial flow fans should be used where air is clean and noise is not a consideration. The typical applications include fume and vapour exhausts, process and made up air supply, industrial, mine and tunnel ventilation, paint spray booth exhaust, combustion air supply and product cooling and drying. (2)

APPLICATIONS OF RADIAL BLADED FANS

Radial bladed fans can be used for handling cold or hot air and gases carrying abrasive and sticky materials. (Ref. 3)

The self cleaning action due to air sweeping over the radial blades makes it more suitable for material handling and other applications where air is dirty. (Ref. 9)

Where worst conditions regarding wear are suspected, the wear resistance can be improved by increasing thickness of the blades and housing at important points ~~with~~ strips of metal.(Ref.8)

The flat pressure characteristic makes it more suitable to be used for compressor so that large range of volumes may be delivered at nearly constant pressure. (Ref. 9)

The radial bladed fans need deep blades for good efficiency and hence the housing needed will be of larger size. The shaft speeds are low requiring belt drive except for smaller sizes.(Ref.3)

APPLICATIONS OF FORWARDLY CURVED BLADED FANS

The advantages of forwardly curved bladed fans over other centrifugal types are their smaller size, lesser weight, lesser space, less cost, less speed and hence lesser noise. They can generate larger pressures at slow speed. They are more commonly used in heating and air conditioning applications. Generally the belt drive is required due to slow speed. (Ref. 9)

APPLICATIONS OF BACKWARD CURVED BLADED FANS

The backward curved bladed fans operate at higher shaft speeds and permit direct drive from standard speed electric motors. They have higher efficiencies, good pressure characteristics, and no difficulty in parallel operation. Due to higher speed, there is more noise. They are suitable for applications where noise is not a consideration such as manufacturing and power plants. Good designs with improved efficiency,& lesser noise are acceptable even for most low pressure ventilation systems. (Ref.10)

Backward curved bladed fans with airfoil blading have higher efficiency, higher pressure and quietness give quieter operation. (Ref. 9)

APPLICATIONS OF RADIAL TIP BLADED FANS

The radial tip bladed fans which have blades radial at the tip only and forwardly curved at the heel are medium speed fans. They are mainly used in induced draft service, and are rarely used for general fan service in industry or ventilation. (Ref. 10)

DISCUSSION

Axial flow fans have large use for low pressure systems.

The propeller fans are particularly suitable for delivering air from one large space to another large space. The tube axial and vane axial fans are particularly useful when the fan is to be installed directly inside the duct. The radial bladed centrifugal fans are particularly useful when the air to be delivered has dirt. Forward curved bladed fans have the main advantages of low speed and lesser noise and hence are particularly useful for air conditioning and ventilating systems. The backwardly curved bladed fans are particularly useful where noise is not a consideration as in production plants. They have the main advantage of wider high efficiency range.

After the type of the fan to be used has been decided, the next consideration is to select the size of the fan.

Before going to next discussion let us see so as to how the point of operation of a fan delivering air or gas against a given system is fixed.

TO FIND OUT THE POINT OF OPERATION FOR A GIVEN FAN AND SYSTEM

A system may be defined as a grouping of ducts, filters, registers, grilles, collectors, plenum chamber etc. through which the air is pushed or pulled. There will be frictional force in the system. Pressure is required to overcome this frictional resistance. This pressure required will be equal to the fan static pressure (S.P.). The graph showing the relationship between volume of air through a system and the corresponding static pressure required to overcome frictional resistance is called system curve. For a given fan operating at a given speed against a given system the point of operation will be where the

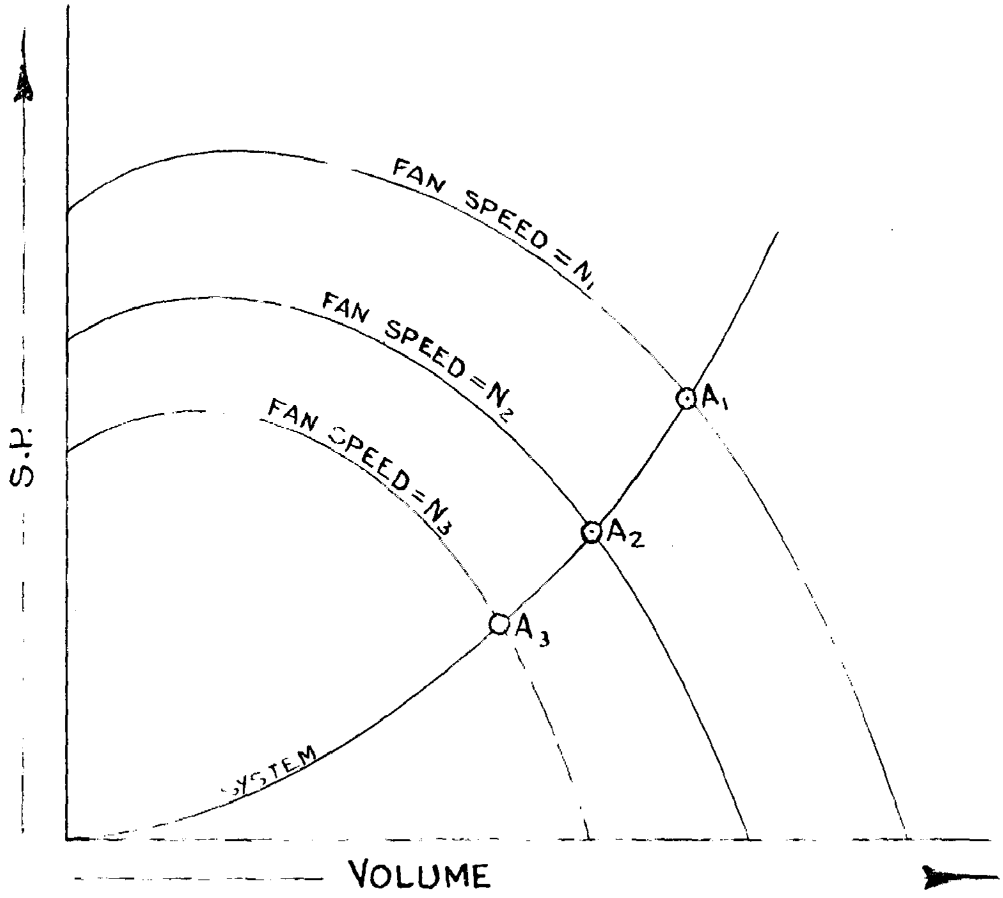
FAN SPEED = N_1
FAN SPEED = N_2
FAN SPEED = N_3
A_1
A_2
A_3
SYSTEM
S.P.
VOLUME

FIG. 8

system characteristics meets the fan static pressure characteristic when the two have been plotted in the same diagram against C.F.M. In figure 8 the system characteristic and the fan static pressure characteristics corresponding to three speeds are drawn. The points of operation for each speed are the points of intersection of fan characteristics with system curve. (Ref. 11)

USE OF RATING TABLES

The fan rating tables or fan performance tables are prepared from the fan performance curves. They are prepared for standard air density (0.075 lbs./cu.ft. for air and 0.078 for flue gases) and are published by the manufacturers for each type and size of fan. A portion of the rating table for some particular fan is given below.

TABLE 1 (Rating Table)

C.F.M.	Outlet Velocity	4½" S.P.		5" S.P.		5½" S.P.	
		R.P.M.	B.H.P.	R.P.M.	B.H.P.	R.P.M.	B.H.P.
2700	1200	817	3.09	860	3.44	889	3.74
3150	1400	824	3.50	864	3.92	894	4.19
3600	1600	832	4.02	870	4.46	901	4.83
4050	1800	840	4.48	877	4.96	908	5.44
4500	2000	845	5.02	884	5.50	916	5.98
4950	2200	854	5.58	892	6.06	925	6.66
5400	2400	864	6.13	901	6.70	936	7.22
5850	2600	873	6.74	911	7.36	945	7.98
6300	2800	887	7.42	923	8.06	958	8.70
6750	3000	897	8.08	936	8.82	971	9.52

EXAMPLE

Required:- To select the fan to deliver 5400 C.F.M. against 5 inches Static Pressure (S.P.).

Solution:- Suppose the selection is to be made for the fan for which rating table 1 is given. First of all find 5400 C.F.M. in the first column, and then proceed horizontally along that row and read the R.P.M. and B.H.P. against 5 inches S.P. which for this particular case comes out to be 901 R.P.M. and 6.70 H.P.

Further it is to be seen that the outlet velocity does not exceed the recommended value for that application. Higher outlet velocities give more noise.

DISCUSSION

To find out the size of the fan the rating tables for various sizes are consulted. Then the size of the fan is selected from these rating tables keeping in view the fan efficiency, noise, cost and space required. The points of operation corresponding to maximum efficiencies are generally shown to be bold faced. The minimum noise goes side by side along with maximum efficiency. The cost and space required depends upon the size of the fan. For selecting smaller size to give the required C.F.M. against required Static Pressure (S.P.) one will have to choose the point of operation with somewhat lower efficiency. Thus a compromise will have to be made between efficiency and cost of the fan. The cost of the fan should also include the cost of the driving unit and ducts etc. Where the fan is to operate for longer periods during the year or where low noise is a requirement the efficiency is given more consideration. When the fan is to operate for short periods and noise is not so important there a smaller size can be selected. A small size fan will have to run at a higher speed and hence will give more noise.

DIMENSIONLESS GROUPING

The arrangement of the various variables involved in the fan operation into dimensionless groups as given in Table 2, proves to be very helpful in fan selection and in preparing the fan rating tables.

Table No. 2

Grouping	Variables	Dimensionless grouping
1	Q, N, D, ρ	$\frac{Q}{D^3N}$
2	Q, N, H, ρ	$\frac{Q^2N^4\rho^3}{H^3}$
3	Q, N, P, ρ	$\frac{Q^5N^4\rho^3}{P^3}$
4	Q, H, P, ρ	$\frac{QH}{P}$
5	Q, H, D, ρ	$\frac{Q^2\rho}{D^4H}$
6	Q, P, D, ρ	$\frac{Q^3\rho}{D^4P}$
7	H, N, D, ρ	$\frac{H}{\rho D^2N^2}$
8	H, N, P, ρ	$\frac{P^2N^4\rho^3}{H^5}$
9	H, P, D, ρ	$\frac{P^2\rho}{D^4H^3}$
10	P, N, D, ρ	$\frac{P}{D^5N^3\rho}$

Here Q = flow rate

H = pressure head

P = Power required

D = Fan diameter

N = Fan R.P.M.

ρ = gas or air density.

For any particular solution the necessary dimensionless groups can be compared. (Ref. 13)

METHOD OF PREPARING RATING TABLES

For a given fan, handling a certain fluid of given density and viscosity, the variables speed N, rate of flow Q, Pressure H and efficiency η are dependent upon each other and are sufficient to describe the performance of that fan. Any two homologous fans have the same curve when plotted with percentage of free air delivery along one axis and the efficiency along the other axis. The dimensionless groups from the last table may be written as

$$K_1 = \frac{Q}{D^3N}$$

$$K_2 = \frac{H}{\rho D^2N^2}$$

$$K_3 = \frac{P}{N^3D^5\rho}$$

$$K_4 = \frac{Q^2\rho}{D^4H}$$

If the fluid be considered incompressible, so that its other ~~physical properties are~~ density remains constant, then the following grouping will be obtained from the above dimensionless groups.

$$K_5 = \frac{H}{D^2N^2}$$

$$K_6 = \frac{P}{N^3D^5}$$

$$K_7 = \frac{Q}{D^2 x \sqrt{H}}$$

$$K_8 = \frac{HQ}{P}$$

Except K_8 these groups are no longer dimensionless as the density has been assumed to be constant and has been excluded from the groups. These groups are very helpful in preparing the fan rating tables.

For preparing the fan rating tables, actual tests are conducted for a fan in accordance with the standard test code and then the rating tables can be prepared for the next larger fans of that group without conducting any test.

First of all the curves for H and P versus Q are plotted from the actual test data. Then the points are taken on these curves very near to each other and from the values of Q, H and P at those points the values of K_5, K_6, K_7 and K_8 are found out and tabulated.

Basically two types of rating tables are required. In one type Q and H may be specified and the values of N and P may be required. In the second type H and P may be specified and Q and N may be required.

For the case when Q and H are specified the following steps may be followed for preparing the rating tables.

(1) Compute the values of K_5, K_7, and K_8 from the original data and tabulate.

(2) Cpmpute K_7 for specified values of Q and H.

(3) Knowing K_7, go to step 1 and by interpolation find values of K_5 and K_8.

(4) Knowing K_8, Q and H solve for P.

(5) Knowing K_5 and D solve for N.

Repeat for as many values of Q, H and D as specified.

In second type of rating tables where H and P are specified and Q and N are required the procedure will be as follows.

(1) Compute values of K_5, K_6, K_7 and K_8 from the original data and tabulate.

(2) From step 1 choose the value of K_6 corresponding to the reading for maximum P.

(3) With K_6 known, P and D specified solve for N.

(4) With N known and H specified compute K_5.

(5) Knowing K_5 go to step 1 and by interpolation find K_7 and K_8.

(6) Knowing K_7, and with H and D specified solve for Q.

(7) With K_8 known, Q known and H specified solve for P.

For different values of H with the same size of motor the speed will remain to be the same as found in step No. 3.

(8) With N known and H specified compute the value of K_5.

(9) Knowing K_5, go to step 1 and, by interpolation, find K_7 and K_8.

(10) Knowing K_7 and with H and D specified solve for Q.

(11) Knowing K_8 and Q and with H specified solve for P.

Repeat for as many values of D, H and P as desired.

The fan speed N, as solved in step 3 of the second type of rating tables is the speed at which this particular fan can operate for all values of H without overloading the motor specified. It does not mean that the power input to fan will be the specified one for all values of H.

The accuracy of these methods depends upon the closeness in taking the values of K_5, K_6, K_7 and K_8. At Texas Engineering Research Station, the points are taken at quite close intervals to ensure accuracy of at least $\pm$ 1 percent. (Ref. 14)

SELECTION OF FAN SIZE FOR NON STANDARD DENSITY OF AIR

Fan catalogue ratings are for standard air with 0.075 lbs/cu.ft. density. When fan is to deliver air or gas with non standard density then the correction has to be applied for fan static pressure and B.H.P. before consulting the rating tables. The air density changes either due to temperature, or pressure or both.

For a given volume rate both the static pressure of the fan and the power requirement will be proportional to the density of the air or the gas to be handled. To select the fan from the standard catalogue to deliver a certain volume Q_1 against some static pressure H_1 under some non standard conditions, first of all find the static pressure H which the fan will develop when delivering standard air at same volume rate Q_1. This will be given by the equation $H = \frac{\rho}{\rho_1} H_1$

where ρ = standard air density

ρ_1 = actual density of air to be handled.

Then select the fan from the standard catalogue corresponding to volume Q_1 and pressure H.

The actual required power P_1 will be given by the equation

$$P_1 = P \times \frac{\rho_1}{\rho}$$

where P = Power given by rating tables.

EXAMPLE

Required: To select a fan to deliver 8000 C.F.M. of air at 600 deg. F and atmospheric pressure (29.92 inches Hg) against 8 inches static pressure.

Solution:

$$H = H_1 \times \frac{\rho}{\rho_1} = H_1 \left(\frac{460 + T_1}{460 + T} \right)$$

$$= 8 \left(\frac{460 + 600}{460 + 70} \right) = 8 \times \frac{1060}{530} = 16 \text{ inches.}$$

The fan corresponding to 8000 C.F.M. and 16 inches static pressure can be selected from the standard catalogue.

If the corresponding power from catalogue is say 32 H.P. then the actual power required will be $32 \times \frac{530}{1060} = 16$ H.P.

In this fan handles 8000 C.F.M. of air having standard density, then the pressure developed will be 16 inches of water and the power required will be again 32 H.P.

Therefore either a damper must be installed or 32 H.P. motor be installed if the fan is to deliver air sometimes at 600 deg. F and sometimes at 70 deg. F.

~~Next consider a fan to deliver 8000~~

EXAMPLE

Required: To select the fan to deliver 8000 C.F.M. of air at 70°F against 8 inches static pressure at an altitude of 7000 ft.

For this first of all find the ratio $\frac{\rho}{\rho_1}$ corresponding to 7000 ft. altitude from table 3 which for the above case is equal to 1.3.

$$\therefore \quad H = H_1 \times \frac{\rho}{\rho_1} = 8 \times 1.3 = 10.4 \text{ inches}$$

Therefore select a fan from standard catalogue corresponding to 8000 C.F.M. and 10.4 inches S.P.

Table 3.

Air Density Correction for Altitude

Altitude ft. above sea level	Factor	Altitude ft. above sea level	Factor
0	1.00	5000	1.20
500	1.02	5500	1.22
1000	1.04	6000	1.25
1500	1.06	6500	1.27
2000	1.08	7000	1.30
2500	1.10	7500	1.32
3000	1.12	8000	1.35
3500	1.14	8500	1.37
4000	1.15	9000	1.40
4500	1.18	10000	1.45

When the fan is to deliver air or gas at higher temperatures then there are so many other things which have to be considered. Consideration will have to be given whether the material of the fan is not overstressed, the bearings are able to operate properly under non-standard temperatures. It may be possible that the bearings may not be able to dissipate the required heat. (Ref. 15)

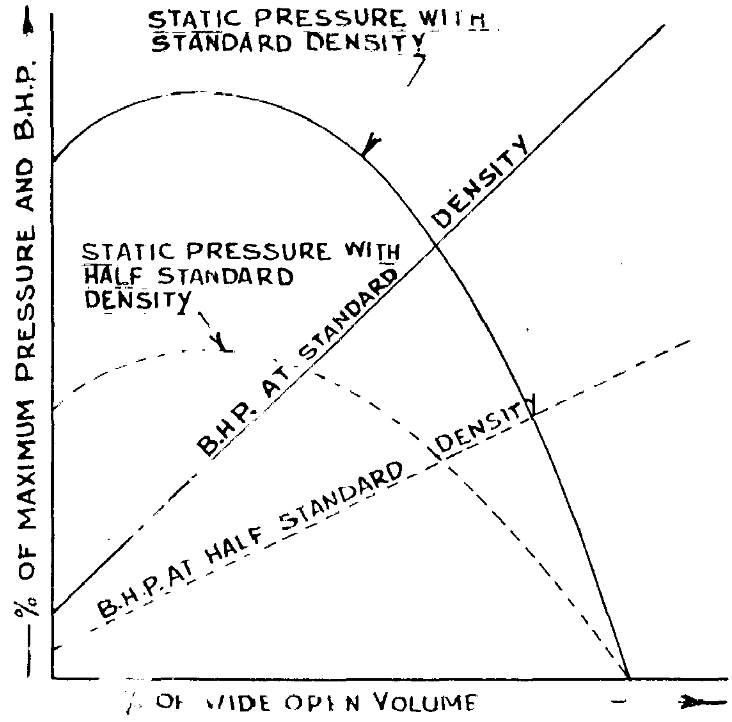
STATIC PRESSURE WITH STANDARD DENSITY
% OF MAXIMUM PRESSURE AND B.H.P.
STATIC PRESSURE WITH HALF STANDARD DENSITY
B.H.P. AT STANDARD DENSITY
B.H.P. AT HALF STANDARD DENSITY
% OF WIDE OPEN VOLUME

FIG. 9

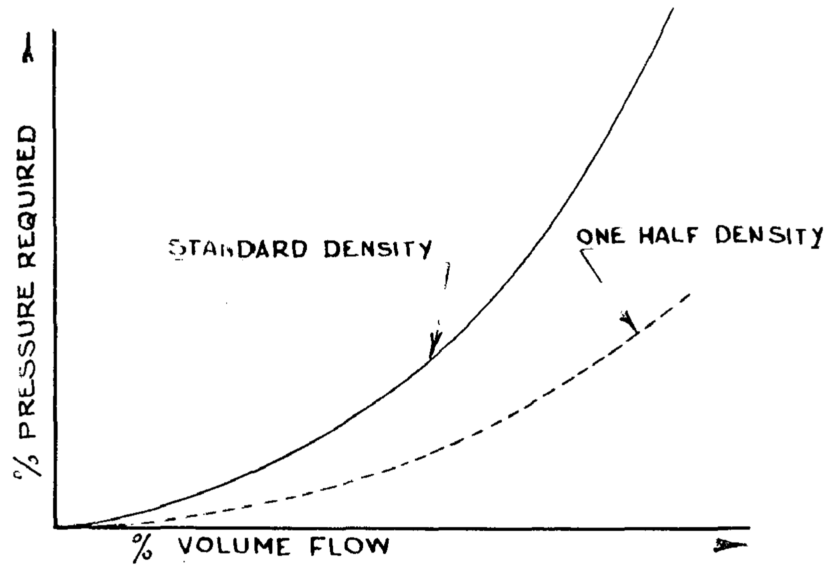
% PRESSURE REQUIRED
STANDARD DENSITY
ONE HALF DENSITY
% VOLUME FLOW

FIG. 10

EFFECT OF AIR DENSITY FOR A GIVEN FAN AND SYSTEM

It has already been discussed that for a given size and R.P.M. of fan the volume handled will remain constant irrespective of density. The static pressure will vary directly proportional to density and B.H.P. will also vary directly proportional to density.

Figure 9 shows this effect of variation in density of air on static pressure characteristic and power characteristic.

The pressure required to move air or some other gas through a system will be proportional to the square of volume. Hence the system curve will be a parabolic curve. Also the pressure required to move given volume of air or gas through the system varies directly proportional to density. Hence the system curve with half standard density will be as shown in figure 10.

From the fan characteristics and system characteristics it can be seen that the fan will deliver same volume against a given system irrespective of density. If the decrease in density decreases the pressure developed by the fan, then it reduces the pressure required by the system also. (Ref. 16)

EFFECT OF CHANGE IN SPEED FOR A GIVEN FAN AND SYSTEM

Whenever we want to handle more volume in air conditioning, ventilating and exhaust systems the remedy ~~after~~ often suggested is to increase the speed of the fan. How much to speed up fan is the question? The fan laws are well known but the system characteristics with ducts, coils and filters etc. is a matter of consideration.

Let Q_1 = initial volume handled by fan

H_1 = initial fan static pressure

P_1 = initial power required

H_3 = initial duct friction loss.

Let the speed of the fan be increased by ratio r.

Then according to fan laws

new volume $Q_2 = r\ Q_1$

new fan S.P. $H_2 = r^2\ H_1$

new fan power $P_2 = r^3\ P_1$

If a straight duct is attached to the fan and the duct resistance also increases by the law $H_4 = r^2\ H_3$ (where H_4 is the final duct friction loss), then there will be no difficulty. But actually the duct friction will increase by the law $H_4 = H_3\ r^{2-q}$ where q lies between 0.09 and 0.16. Thus the volume handled will be slightly larger than that given by fan laws. The percentage error will be = $(r^q - 1) \times 100$.

The pressure drop through the remaining equipment also varies approximately in the similar manner as that through the ducts (especially in case of items having greater resistance).

Thus when the speed of the fan is raised to increase volume the fan laws give lesser values of new volume delivered. For example if the speed is doubled the actual new volume may be over 10 percent in excess of the value given by the fan laws.(

(Ref. 17)

SUMMARY OF CHAPTER II

Fan Selection

Fan selection includes the selection of type and size. The type to be choosen, depends upon the application. The particular characteristics of the fans which help in selecting the type of fan have been discussed in the chapter. The fan size can be selected from the fan rating tables published by the manufacturers. The rating tables are prepared for standard density of air. For non standard air corrections have to be applied to fan static pressure before consulting the rating tables and to horse power after consulting the rating tables. The rating tables are prepared from fan performance curves. Use of dimensionless groups helps lot in fan selection and in preparing the fan rating tables.

At Texas Engineering Research Station the rating tables are prepared for different sizes of fans with the help of dimensionless groups, from the test data of only the fan of smallest size of that class.

The density of air or gas handled by the fan has no effect on volume handled, but the S.P. and Horse Power is directly proportional to density.

For a given fan and system when the speed is increased to increase volume then the actual new volume handled is somewhat greater than the volume given by fan laws.

CHAPTER III

CHAPTER - III.

TESTING OF FAN PERFORMANCE

SUMMARY OF CHAPTER III

The tests are conducted on fans for improvement in design and for issuing the certificates for their performance. The performance of the fan also depends upon their installation in the field. Basically the fan installations are of four types (1) Bare fan has no ducts (2) Blower has duct on discharge side only (3) Booster carries ducts on both sides and (4) Exhauster carries duct only on the inlet side.

The various test set-ups as recommended by A.S.M.E. are given in figures 11-15 and the description of the equipment used has also been included. The traverse method of air flow measurement has been fully discussed. The various definitions and the method of calculations as prescribed by A.S.M.E. ~~has~~ have been included in the chapter.

A special circular disc slide rule designed and constructed at Texas Engineering Experiment station has reduced the time of air density calculation to its one fourth and gives results with error within 0.7 percent.

The use of orifices and nozzles for air flow measurement at low volumes is more convenient.

The Vicking Air Products are using an automatic plotter, which plots the fan performance curves automatically. The error with this method is within 2 percent and complete test time at one constant speed is approximately one or two minutes.

The methods of testing fans in the field are more approximate and ~~quick~~ quicker than laboratory methods.

CHAPTER - III.

TESTING OF FAN PERFORMANCE.

The tests are conducted on the fans for their performance for two reasons.

(1) For improvement in design and

(2) For issuing the certificate for their performance.

To establish the agreement between the seller and the buyer, different associations have prescribed different standard methods of testing.

BASIC FAN INSTALLATION ARRANGEMENTS

The performance of a fan depends upon the different arrangements of its installation in various applications. The different applications are covered by four basic installation arrangements.

(1) Bare Fan:- There are no ducts on either side. The discharge is from one large space to another large space. In other words the fan is a wall installation.

(2) Blower:- Where the fan discharges from a large space to duct on discharge side.

(3) Booster:- Where the ducts are provided both on suction and discharge sides of the fan.

(4) Exhauster:- Where the fan intakes the air through duct but discharges into a large space.

The fan must be tested under the conditions similar to that in which it is to be installed in the field. (18)

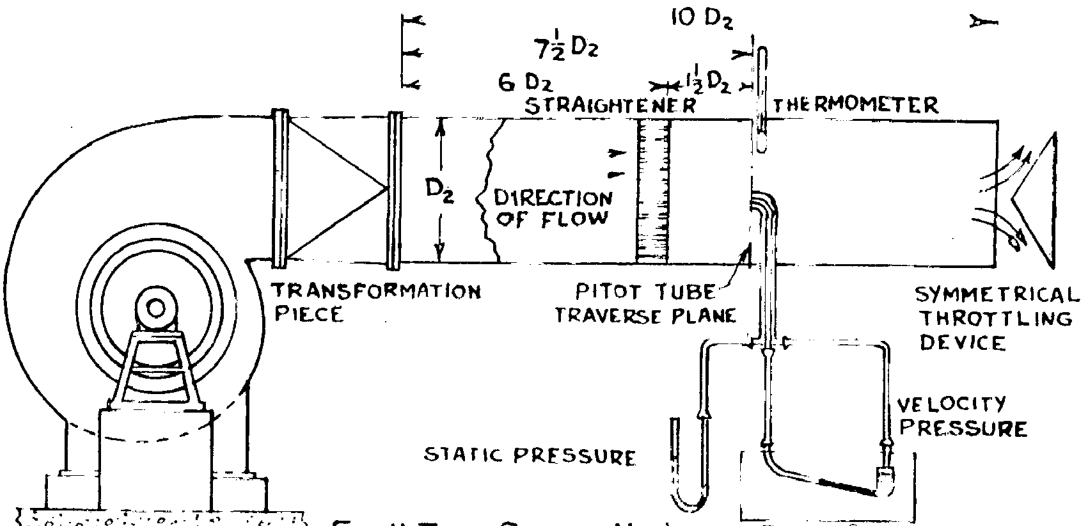

FIG. 11 TEST SETUP No. 1 WITH DISCHARGE DUCT ONLY.

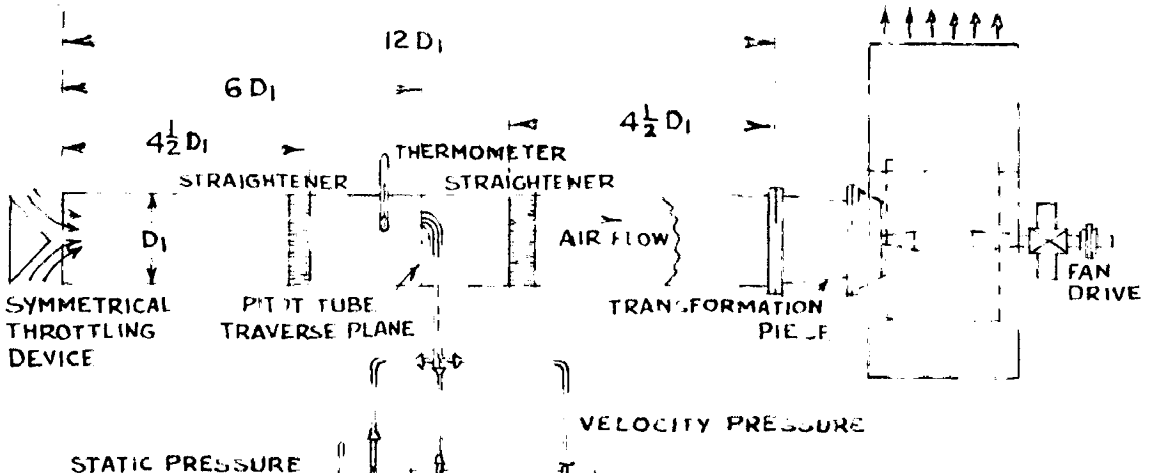

FIG. 12 TEST SETUP No. 2 WITH INLET DUCT ONLY.

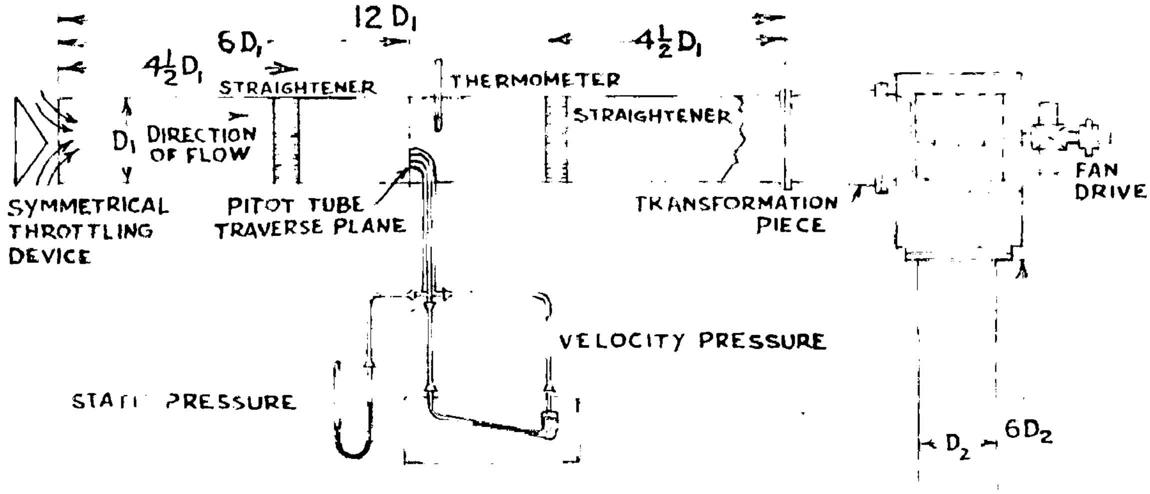

FIG. 13 TEST SETUP No. 3 WITH INLET AND DISCHARGE DUCTS.

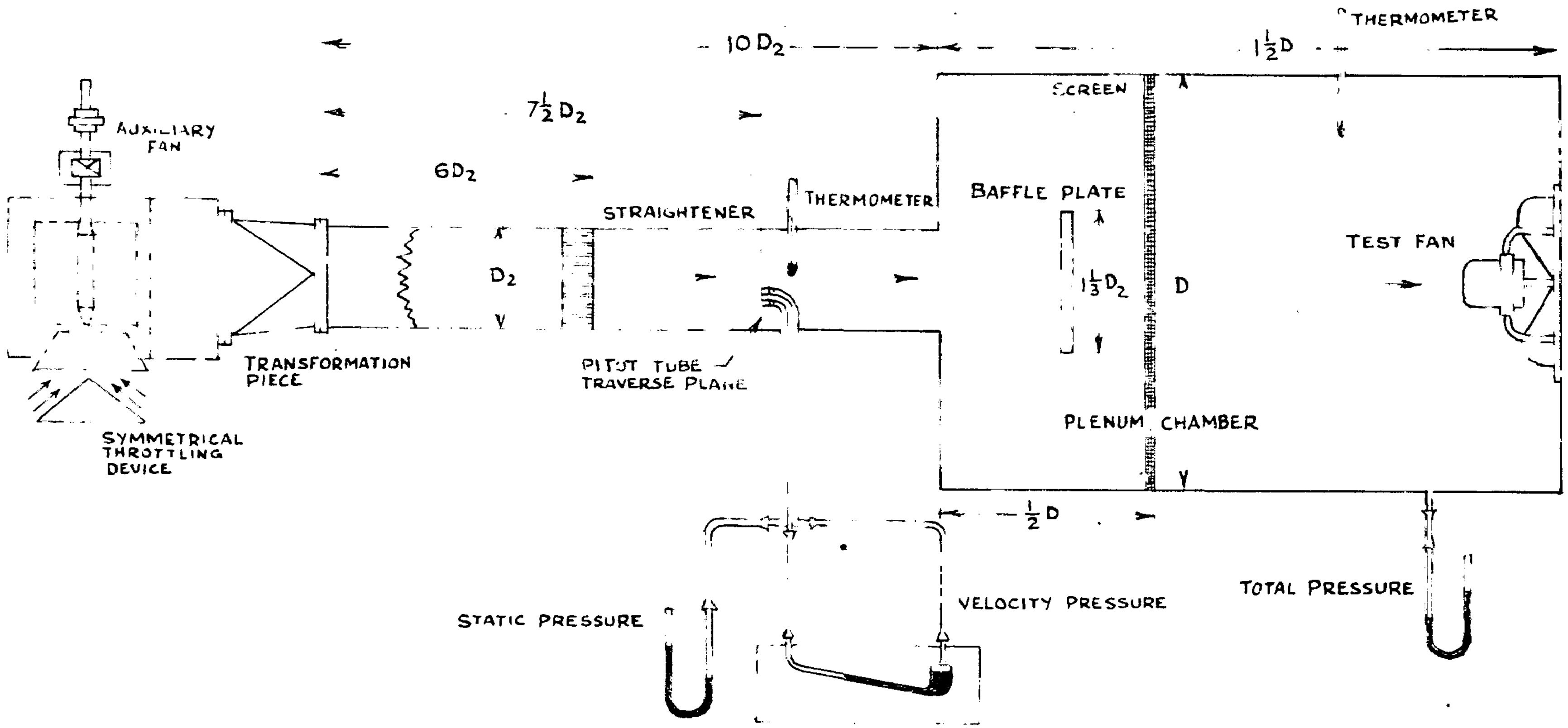

FIG. 14 TEST SETUP No. 4 WITH FAN IN WALL OF PLENUM CHAMBER

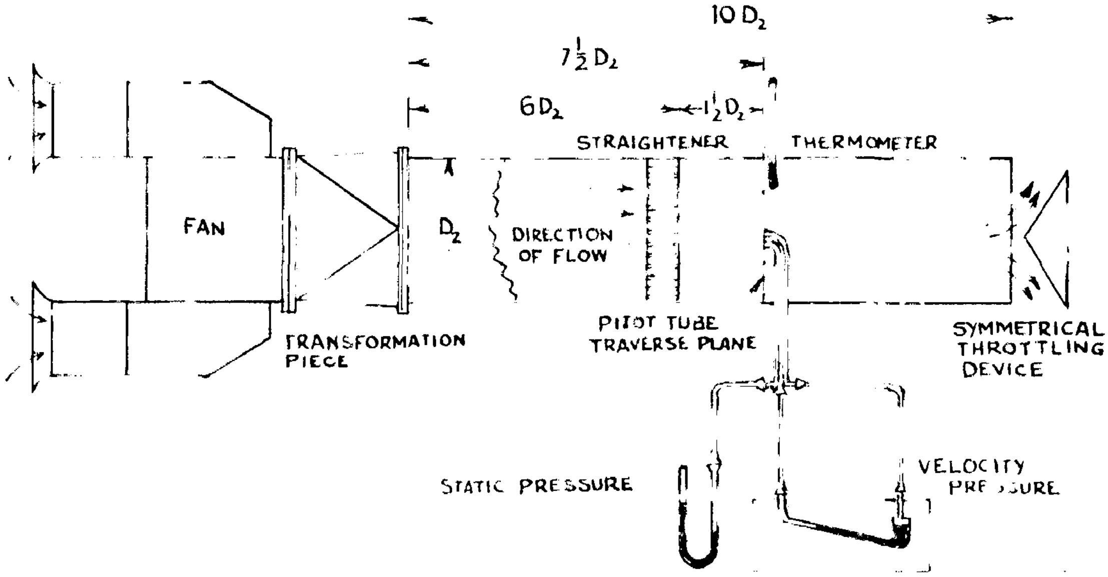

FIG. 15 TEST SETUP No 5 WITH DOUBLE INLET BOXES AND DISCHARGE DUCT.

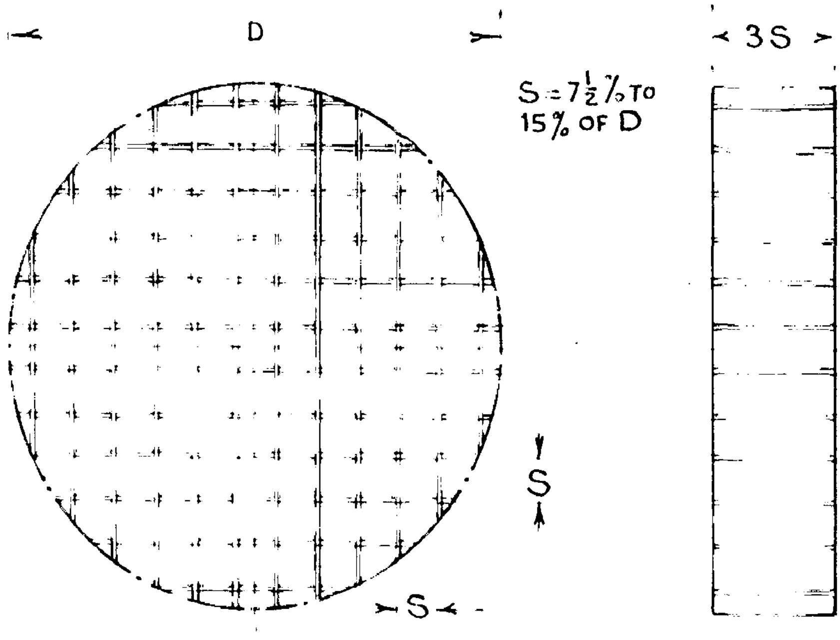

FIG.16 AIR STRAIGHTENER

DIFFERENT TEST SET-UPS

The A.S.M.E. has prescribed the five set-ups shown in figure 11-15 for testing of the fans to cover the various conditions under which they can be installed. Set-up No. 1 (fig. 11) is used when the fan is to be installed with discharge duct and without inlet duct. Set-up No. 2 (fig. 12) will be used for testing the fan when the fan is to be installed with inlet duct and without discharge duct. Set-up No. 3 (fig. 13) will be used when the fan is to be installed with inlet and discharge ducts both. Set-up No. 4 (fig. 14) will be used when the test fan is intended to be installed without ducts. Set-up No. 5 (Fig. 15) is used for a fan which is to be installed with inlet box or boxes and with discharge duct. (7)

THROTTLING DEVICES

In the testing of fan performance, the tests at different volume rates are conducted. The volume rate is varied by creating different resistances to air flow in the duct by some symmetrical throttling device. The devices for this purpose may be an orifice plate, perforated plate, adjustable cone etc.

TRANSFORMATION PIECES

The transformation pieces for changing the duct section from rectangular to round should be so constructed that the angle between any longitudinal element and the duct axis does not exceed 7 degrees. (7)

AUXILLIARY FAN OF SET-UP NO. 4.

It may be of any convenient size, provided it can produce

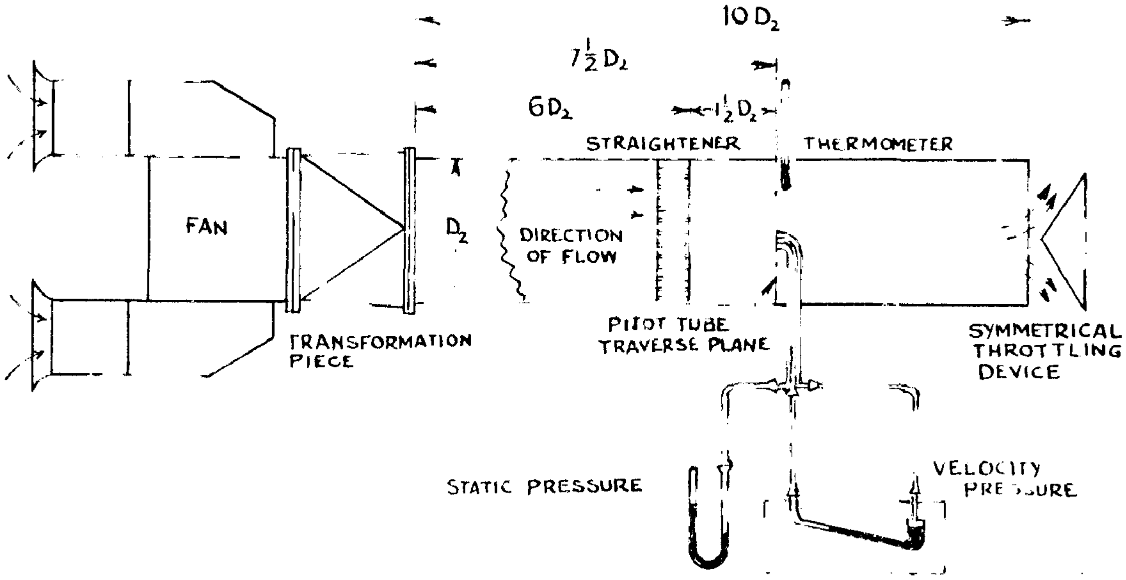

FIG. 15 TEST SETUP No 5 WITH DOUBLE INLET BOXE AND DISCHARGE DUCT.

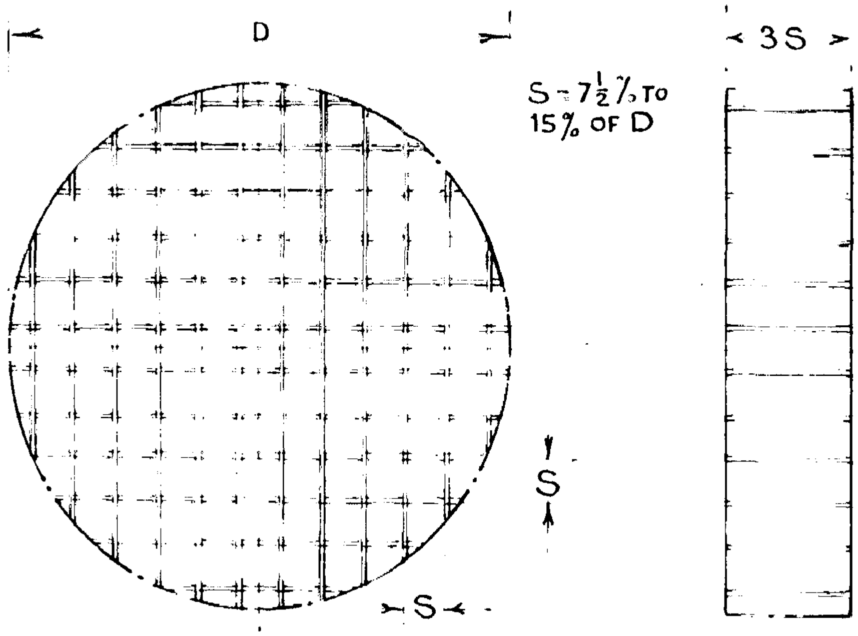

FIG.16 AIR STRAIGHTENER

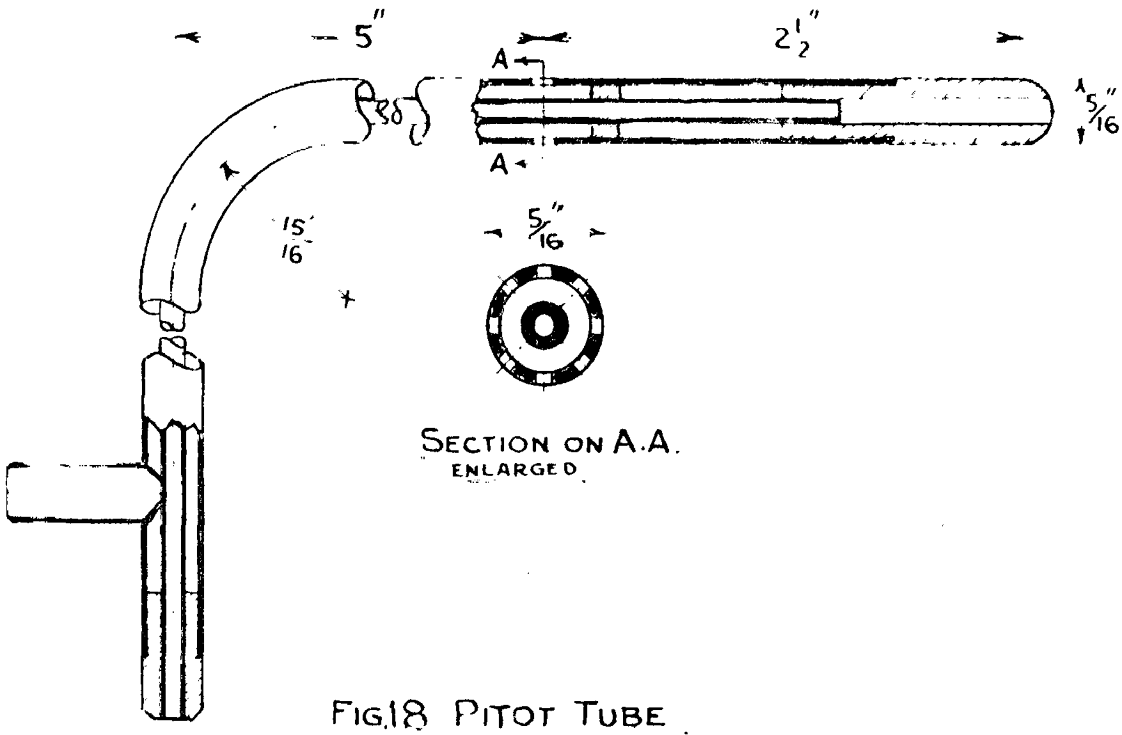

FIG. 18 PITOT TUBE.

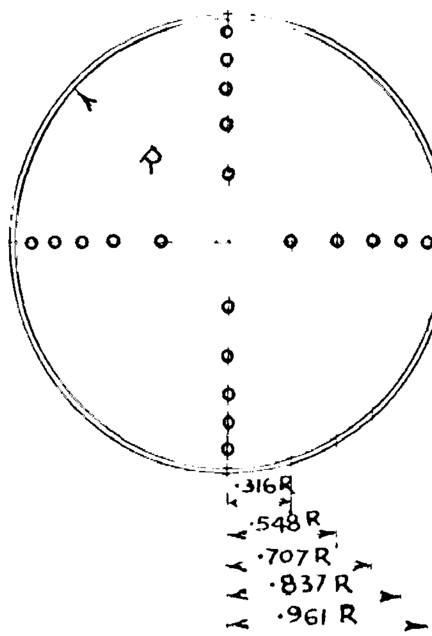

FIG. 17 TRAVERSE POINTS IN DUCTS.

DIFFERENT TEST SET-UPS

The A.S.M.E. has prescribed the five set-ups shown in figure 11-15 for testing of the fans to cover the various conditions under which they can be installed. Set-up No. 1 (fig. 11) is used when the fan is to be installed with discharge duct and without inlet duct. Set-up No. 2 (fig. 12) will be used for testing the fan when the fan is to be installed with inlet duct and without discharge duct. Set-up No. 3 (fig. 13) will be used when the fan is to be installed with inlet and discharge ducts both. Set-up No. 4 (fig. 14) will be used when the test fan is intended to be installed without ducts. Set-up No. 5 (Fig. 15) is used for a fan which is to be installed with inlet box or boxes and with discharge duct. (7)

THROTTLING DEVICES

In the testing of fan performance, the tests at different volume rates are conducted. The volume rate is varied by creating different resistances to air flow in the duct by some symmetrical throttling device. The devices for this purpose may be an orifice plate, perforated plate, adjustable cone etc.

TRANSFORMATION PIECES

The transformation pieces for changing the duct section from rectangular to round should be so constructed that the angle between any longitudinal element and the duct axis does not exceed 7 degrees. (7)

AUXILLIARY FAN OF SET-UP NO. 4.

It may be of any convenient size, provided it can produce atmospheric pressure in the plenum chamber whenever desired.

AIR STRAIGHTENERS

The air straighteners are provided to eliminate the rotational component of air, and to make the air velocity ~~to be~~ parallel to duct axis before it reaches the plane of traverse. The straighteners as prescribed by A.S.M.E. (fig. 16) should consist of square cells with sides 7.5 to 15 percent of the duct diameter. The prescribed length of the straighteners in the direction of air flow is three times that of the side of square cell. Further 22 U.S. gage sheet metal has been recommended for the straighteners. (7)

PITOT TUBE

The design of pitot tube as prescribed by A.S.M.E. is shown in figure 18. A similar tube was constructed in the Roorkee University and the details are given in Chapter IV.

MANOMETERS

The manometers are just liquid - filled glass tubes for indicating pressure differences. The tubes may be vertical or inclined. In case of inclined manometers the inclination should not be more than 84 degrees from the vertical. Inclined manometers give more accurate reading for small pressure differences (2 inches of water or less). For inclined manometers mineral oil, kerosene, or other liquids may be used in place of water to extend the scale and flatten the meniscus.

All manometers, other than vertical leg tubes filled with distilled water, should be calibrated in position before and after test with all tubing in place by a water-filled hook gauge or micromanometer. (7)

INDIAN STANDARDS FOR TESTING OF PROPELLER TYPE A.C. VENTILATING FANS.

The Indian Standards (I.S. 2312, 1963) have prescribed the test procedure for propeller type A.C. ventilating fans. The necessary details of the contents of the code concerning to this study are given in Appendix C.

The code covers only the free delivery test for the fan. It does not give any information regarding the performance of the fan when operating against some resistance. The method of calculating the air velocity with vane anemometer though simple, is less accurate. A vane anemometer does not give reliable results (particularly at too low and too high speeds). The anemometer gives reliable results when the velocities are of the order of 500 F.P.M. The power input test measures only the power input to fan motor and not the power input to fan.

A TRAVERSE

A traverse is a series of ten or more readings made with a pitot tube or a static tube taken along the diameter of a duct at a selected location. (7)

MEASUREMENT OF AIR FLOW BY TRAVERSE METHOD

For measuring the air flow in the circular ducts the pressure readings are taken with pitot tube at the locations specified in Fig. 17. Twenty locations in figure 17 represent two traverses along axes at right angles to each other. For verification the traverses can also be taken on other axes. The background for taking these points at prescribed distances from duct centre is as follows. (7)

METHOD OF FIXING TRAVERSE POINTS

The duct cross sectional area is divided into five equal concentric zones. The points for readings are selected at the radii which further sub-divide, the zones equally. Accordingly the positions of the pitot tube for different readings will be at $\sqrt{\frac{1}{10}}$ R, $\sqrt{\frac{3}{10}}$ R, $\sqrt{\frac{5}{10}}$ R, $\sqrt{\frac{7}{10}}$ R, and $\sqrt{\frac{9}{10}}$ R from the centre.

These distances represented in decimal form will be as 0.316R, 0.548R, 0.707R, 0.837R and 0.949R. This traverse method would have given correct results if velocity at each point of reading were the average velocity for the corresponding area. Near the duct wall the velocity change is very abrupt and consequently the velocity at the outermost readings is not the true average velocity for that zone and the error is introduced due to that. In the other cases it is nearly correct to assume the average velocity at the point of reading because the velocity change is gradual. When the last reading is taken at 0.961R instead of 0.949R then more accurate results are obtained. (19)

STATIC PRESSURE AT PLANE OF TRAVERSE is taken to be the arithmatic average of the static pressures taken at 20 similar fixed points at the plane of traverse. (7)

VELOCITY PRESSURE AT PLANE OF TRAVERSE is taken to be the square of the average of the square roots of the velocity pressures taken at 20 similar fixed points at the plane of traverse.

TOTAL PRESSURE AT PLANE OF TRAVERSE is the algebraic sum of the velocity pressure at the plane of traverse and the static pressure at the plane of traverse. (7)

FAN OUTLET AREA is determined from the inside dimensions of the fan outlet. The outlet area of a fan furnished with a diffuser is the area at outlet of the diffuser.

FAN INLET AREA is determined from the inside dimensions of the fan inlet. For a fan with inlet boxes, the inlet area is that of the box openings. (7)

A TEST RUN pertains to all readings and calculations at any one setting of the throttling device. (7)

DENSITY OF ATMOSPHERIC AIR (CALCULATIONS)

$$\rho_a = \frac{P_t - 0.38\ P_w}{0.754\ T_a}$$

where ρ_a = density of atmospheric air in lbs. per cu. ft.

T_a = absolute temperature of atmospheric air, F + 460

p_t = barometric pressure, inches of Hg at 32°F.

p_w = partial vapour pressure in atmosphere, inches of Hg.

and $$p_w = P_s - \frac{p_t(t_a - t_w)}{2700}$$

where p_s = sat vapour pressure at wet bulb temperature in inches of Hg.

t_a = dry bulb temperature in deg. F.

t_w = wet bulb temperature in deg. F. (7)

DENSITY AT PLANE OF TRAVERSE

$$\rho_d = \rho_a \frac{p_t + (p_s'/13.6)}{p_t} \quad \frac{T_a}{T_d}$$

where ρ_a = density of atmospheric air lbs/cu.ft.

ρ_d = density of air at plane of traverse lbs/cu.ft.

p_s' = measured static pressure at plane of traverse, inches water.

T_d = absolute temperature of air in duct at plane of traverse, deg. F + 460

T_a = absolute temperature of atmospheric air, deg.F+460

p_t = atmospheric pressure inches Hg. (7)

In the above equation 13.6 represents the ratio of the density of mercury to that of water.

VELOCITY AT PLANE OF TRAVERSE

For the design of pitot tube shown in figure 18, the A.S.M.E. has prescribed the formula:

$$V = 1096.2 \sqrt{\frac{p_v'}{\rho_d}}$$

where V = Average velocity in duct at plane of traverse in F.P.M.

p_v' = Average velocity pressure at plane of traverse in inches of water.

ρ_d = Density of air at plane of traverse, lbs. per cu. ft. (7)

VOLUME RATE OF FLOW AT PLANE OF TRAVERSE

$$Q_d = A\ V$$

where Q_d = volume rate of air flow at plane of traverse in C.F.M.

V = velocity at plane of traverse in feet per minute.

A = duct cross sectional area at plane of traverse in sq. ft.

VOLUME RATE OF FLOW CORRECTED TO FAN INLET

According to the definition of capacity of a fan, it is the volume rate of flow of air at the fan inlet. Therefore the volume rate of flow obtained at plane of traverse must be corrected to fan inlet conditions to obtain the capacity of the fan.

If Q_a = C.F.M. at fan inlet.

Q_d = C.F.M. in duct at plane of traverse.

ρ_d = air density in duct at plane of traverse.

and ρ_a = density of atmospheric air.

then $$Q_a = Q_d \times \frac{\rho_a}{\rho_d} \qquad (7)$$

(Note) This correction will only apply to set up No. 1 and 5. For set-ups Nos. 2, 3 and 4 the volume rate of flow at fan inlet will be nearly the same as that at the plane of traverse.

DUCT FRICTION LOSS

To calculate the friction loss for a round duct the following formula shall be used.

$$p_f = \frac{0.02\ L\ p_v^{I}}{D} \qquad (7)$$

where p_f = pressure loss in duct, in inches of water.

p_v' = average duct velocity pressure in inches of water.

L = distance from fan to pitot tube location in ft.

D = diameter of duct in feet.

(Note):- Loss in transformation piece is computed as part of the total length L and as if it were round duct of diameter D. Loss in a possible rectangular discharge duct is computed on basis of loss in a round duct of same hydraulic radius.

(7)

STRAIGHTENER LOSS

The loss of pressure due to friction of the straightener shall be considered as equivalent to the loss in a length equal to four diameters of the test duct as calculated by the above formula. This loss shall be applied to total pressure.

(7)

DISCUSSION

The application of these duct and straightener friction losses to the various set-ups will be as follows.

(1) Set-up No. 1.:- Inlet total pressure will be atmospheric while the fan discharge total pressure will be equal to the total pressure at plane of traverse plus duct friction and straightener losses, upto the traverse.

(2) Set-up No. 2:- Fan inlet pressure will be the total pressure at plane of traverse minus the duct friction and straightener loss between the plane of traverse and fan inlet. Total pressure at discharge is equal to atmospheric pressure plus discharge velocity pressure.

(3) Set-up No. 3:- Fan inlet pressure is equal to total pressure at plane of traverse minus duct friction and straightener losses between the plane of traverse and fan inlet. Total pressure at fan discharge will be equal to atmospheric pressure plus discharge duct losses plus discharge velocity pressure.

(4) Set-up No. 4:- Fan inlet pressure will be the total pressure in plenum chamber. The fan outlet total pressure will be the atmospheric pressure plus discharge velocity pressure. Duct friction losses and straightener loss can be avoided from the calculations.

(5) Set-up No. 5:- The procedure will be same as that for set-up No. 1.

DISCHARGE VELOCITY PRESSURE

The discharge velocity pressure will be given by the formula

$$p_v = p_v' \left(\frac{A}{A_o}\right)^2 x \left(\frac{\rho_o}{\rho_d}\right)$$

where p_v = discharge velocity pressure in inches of water.

p_v' = velocity pressure at plane of traverse in inches of water.

ρ_o = density of air at fan outlet.

ρ_d = density of air at plane of traverse.

A_o = fan outlet area in sq. ft.

A = cross-sectional area of duct at plane of traverse in sq. ft.

Where the velocity is measured at discharge duct the ratio $\frac{\rho_o}{\rho_d} = 1$ (7)

VOLUME RATE OF FLOW CORRECTED TO SPECIFIED SPEED

The volume of air delivered by a fan for a given setting of throttling device at a given plane varies directly as the fan speed. It is not at all affected by the density of air.

If Q = volume rate of flow at fan inlet corrected to specified speed.

Q_a = volume rate of flow at fan inlet at test speed.

N_s = specified speed of fan.

N_x = test speed of fan.

Then $Q = Q_a \times \frac{N_s}{N_x}$

The volume rate of flow Q at fan inlet thus corrected to specified speed will give the capacity of the fan for the given throttling device. (7)

PRESSURE CORRECTED TO SPECIFIED INLET AIR DENSITY AND SPEED

For a given setting of throttling device pressure at a point varies as the square of speed and directly as the density at that point.

$\therefore$ pressure correction factor = $\left(\frac{N_s}{N_x}\right)^2 \times \frac{\rho_s}{\rho_x}$

where N_s = specified speed

N_x = test speed

ρ_s = specified density of air

ρ_x = air density at fan inlet during test. (7)

EXAMPLE

If H_s' = Fan static pressure during test.

H_s = Fan static pressure at specified inlet air density and speed.

$$\text{Then } H_s = H_s' \times \left(\frac{N_s}{N_x}\right)^2 \times \frac{\rho_s}{\rho_x}$$

Similarly if H_t' = Fan total pressure during test.

H_t = Fan total pressure at specified inlet air density and speed.

$$\text{Then } H_t = H_t' \left(\frac{N_s}{N_x}\right)^2 \times \frac{\rho_s}{\rho_x}$$

POWER INPUT

The power input to the fan may be measured with a dynamometer. If the dynamometer is not available then the power input may be calculated from the electrical energy supplied to the motor multiplied by motor efficiency as determined by test.

POWER INPUT CORRECTION

The power input varies as the cube of the speed and directly as density

$$\therefore \text{ power input correction factor} = \left(\frac{N_s}{N_x}\right)^3 \times \frac{\rho_s}{\rho_x} \qquad (7)$$

EXAMPLE

If P_x = Power input to fan during test.

and P = Power input to fan at specified inlet air density and speed.

$$\text{Then} \quad P = P_x \left(\frac{N_s}{N_x}\right)^3 \times \frac{\rho_s}{\rho_x}$$

TOTAL EFFICIENCY (CALCULATION)

It is calculated as follows

$$\text{Total efficiency } \eta_t = \frac{Q \times H_t}{6356\ P}$$

where Q = Capacity of fan in C.F.M.

H_t = Fan total pressure in inches of water.

P = Power input, H.P.

The quantities Q, H_t and P are all as corrected to specified speed and density. (7)

STATIC EFFICIENCY (CALCULATION)

It is calculated as follows:

$$\text{Static efficiency} = \frac{Q \times H_s}{6356 \times P}$$

where Q = Capacity of fan in C.F.M.

H_s = Fan static pressure in inches of water.

P = Power input, H.P.

The quantities Q, H_s and P are all as corrected to specified speed and density.

Since same correction factors come in the numerator and denominator of efficiency terms hence the efficiencies calculated under test conditions will be same as that obtained after corrections. (7)

QUICK DETERMINATION OF AIR DENSITY (A disc slide rule)

The calculations of air density is a lengthy job. At Texas Engineering Experiment Station, thought was given to the various methods by which the air density could be calculated from a knowledge of barometric pressure, dry bulb temperature and wet bulb depression. As a result a circular slide rule calculation has been designed and constructed.

The equation satisfied by the calculations with this slide rule is

$$\rho_a = \frac{p_t - 0.375 \left[p_s - 0.011092\,(t_a - t_w)\right]}{R\,T_a}$$

where ρ_a = density of moist air, lb. per cu. ft.

p_t = atmospheric pressure, inches Hg.

t_a = dry bulb temperature deg. F.

T_a = dry bulb temperature deg. R.(Rankine)

t_w = wet bulb temperature, deg. F.

p_s = saturated vapour pressure inches of mercury at t_w.

R = gas constant for dry air, 0.75430 (cu. ft.) (inches of mercury)/(lb.) (deg. R), based on the work of John A. Goff and S. Gratch.

The constant 0.011092 involves an approximation for atmospheric pressure. A median value of 29.5 inches of mercury has been assumed. The maximum error with this approximation can be $\pm$ 11 x 10^{-6} lb. per cu. ft. at the extremes of the ranges of barometric pressures.

The use of slide rule is restricted to the following ranges, barometric pressure 28.5 to 31.0 inches of mercury; dry bulb temperature, 60 to 110 deg. F; and wet bulb depression, 0 to 20 deg. F. Within this range the error in density calculation will be approximately within 0.7 percent. The use of this slide rule reduces the time of air density calculation for a normal fan test from 40 minutes to 10 minutes. (20)

USE OF ORIFICES AND NOZZLES IN AIR FLOW MEASUREMENT

Though the pitot tube traverses can give satisfactory results but they demand much time for taking the observations and also when used with a strongly pulsating discharge, may give appreciable errors. (21)

The other devices commonly used for this purpose are nozzles and orifices. The advantage of nozzles is that their coefficient can be found out more accurately by means of a traverse with a small impact tube. Such a nozzle becomes primary standard for calibrating other devices such as orifices. The disadvantage of nozzles is their high cost and difficulty in accurately manufacturing. When the coefficient of an orifice has been accurately determined then it becomes as accurate as a nozzle. It is ~~more~~ cheaper and convenient. (23)

According to L.S. Marks a combination of pitot tube and square edged inlet and discharge orifices has been suggested for the air flow measurement in ducts in fan testing. The pitot tube be used at higher fan discharges, while the inlet or discharge orifices be used for lower fan discharges. It is suggested that the pitot tube be used down to capacities approximately upto capacity at maximum efficiency, and orifices be used for capacities lower than that. The pitot tube traverses should also be made when the two largest orifices are used to check the accuracy and accordance of the two methods.

The large sized square edged orifices are more accurate and give predictable coefficients of discharge. Generally orifices above 8 inches in diameter will give quite satisfactory results when machined with ordinary care and all burs removed. However, more care is necessary during manufacturing for smaller orifices. The orifice performance is mainly effected by (1) the Reynolds number and (2) the ratio of orifice area to duct area. (23)

Discharge coefficients for thin plate inlet orifices were determined by Ebaugh and Whit field. Ebaugh and Whitfield have found that in an inlet orifice, if the pressure tap be located 40 percent of the pipe diameter downstream, a constant coefficient of 0.601 is obtained for orifice area ratios 0.2 to 1.0. For orifice area ratios 0.2 to 0.8, the pressure differential can be measured with an accuracy within ± 0.5 percent. (21)

Stach has obtained the following results for discharge orifices (I.S.A. orifices).

m	0.05	0.10	0.15	0.20	0.25	0.30	0.35
Coeff.	0.598	0.602	0.608	0.615	0.624	0.636	0.651
m	0.40	0.45	0.50	0.55	0.60	0.65	0.70
Coeff.	0.666	0.682	0.701	0.724	0.751	0.784	0.820

The accuracy of these coefficients is given as = 0.8 per-cent. They apply to the simple flow equation $V^2 = 2\ gh$ and the correction for velocity of approach is included in the coefficient. (23)

DISCUSSION

From the results given above we see that the coefficients for inlet orifices are more constant then that for the discharge orifices.

DUCT ORIFICES

Though the coefficients are more well defined for the duct orifices but they are not recommended because of inconvenience in use. In case of inlet and discharge orifices a series of orifices can be used because there is no difficulty in changing the orifice. (23)

The nozzles and orifices give predictable and reproducable discharge coefficients at high Reynolds numbers but the discharge coefficients do not remain constant at low Reynolds numbers and are not completely reproducable even. Many investigators have tried with new designs, so that by an opportune compensation, a constant value of coefficient of discharge may be maintained at low Reynolds numbers. (22)

DISCUSSION

We see that at low Reynolds numbers the orifice coefficient is not constant. Error may be introduced when the orifice is used to measure flow at low volumes. But for the same flow the pressure differential reading at orifice tap will be sufficiently large than, that with pitot tube. Hence pressure differential reading with orifice can be taken more accurately. The error introduced due to change in orifice coefficient will not be so large as it may be due to inaccuracy in reading the velocity pressure with a pitot tube at very low volumes.

AUTOMATIC PLOTTER FOR PERFORMANCE CURVES

With manufacturers having experimental design apparatus which are frequently of a cut and try nature, the pitot tube traverse method for flow measurement does not prove to be suitable because of being too lengthy. Testing devices have been produced in which the fan performance curves are automatically plotted.

The Vicking Air Products, Div., National U.S. Radiator Corp. is using an equipment in which fan performance curves at various speeds are plotted automatically by computer in terms of B.H.P. or watt input (in case of directly connected motors), and static pressure vs. flow. (24)

The test fan can be driven at any desired constant speed within ± 1 R.P.M. The power input to the fan is measured by means of a Torquemeter. The output of the torquemeter in milivolts varies directly as the torque. Since the speed is constant therefore the torquemeter output varies directly proportional to Horse Power input. This output is fed to plotter to indicate shaft horse power directly. (24)

The static pressure is fed to a differential strain gage transducer and the output is further fed to the plotter.

The test apparatus consists of nozzles of 5, 6 and 8 inches diameter which can be used individually or in combination. The opening and closing of nozzles is done by basket-ball type bladders in the nozzle throat which can be inserted or retracted from the nozzles. The pressure differentials across the nozzles are also detected with strain gage transducer. Since the flow is proportional to square root of differential pressure at nozzle, hence the square root of the signal given out by the strain gage is extracted. The signal which now represents C.F.M. actuates the chart paper.

The probable error introduced by this method is within 2 percent. Complete test time at one constant speed is approximately 1 to 2 minutes. (24)

<u>TESTING OF FANS IN THE FIELD</u>

In testing of the fans in field, it is not possible to achieve so much accuracy as ~~it~~ can be obtained in the laboratory methods. There are three principal methods of testing the fans in the field.

Each of them has its own advantages and degree of accuracy.

<u>Method No. 1:-</u> Anemometers can be used to measure air flow through supply outlets and return inlets. A correction factor has to be applied depending upon the type of supply outlet or intake used. Generally the velocity reading is multiplied by 0.85 for return air intakes and by 1.03 for supply outlets. The anemometer needs caliberation after some usage. (25)

<u>Method No. 2:-</u> The velometer gives direct reading for velocity within 2 or 3 percent accuracy. It also needs frequent factory adjustment. When handled and used with care, it can suffice for most needs in the field. It should not be used on dust, acid or material handling systems.

<u>Method No. 3:-</u> Pitot tube is considered to be the basis for caliberation of other air flow measuring devices. It is more durable and accurate.

In the locations where the velocity pattern is uniform with highest velocity nearly at the centre, the velocity reading at the centre is taken and then multiplied by 0.8 to get the average velocity. Same will apply to anemometers and velometers when a single reading is to be taken with them in the duct. (25)

Where maximum accuracy is desired, a complete traverse, as in laboratory method can be taken. If possible the location of traverse should be such ~~so~~ that the static pressure at different points of traverse is constant. By doing so the necessary number of static pressure readings at the plane of traverse can be reduced. If some straight length of the duct several diameters

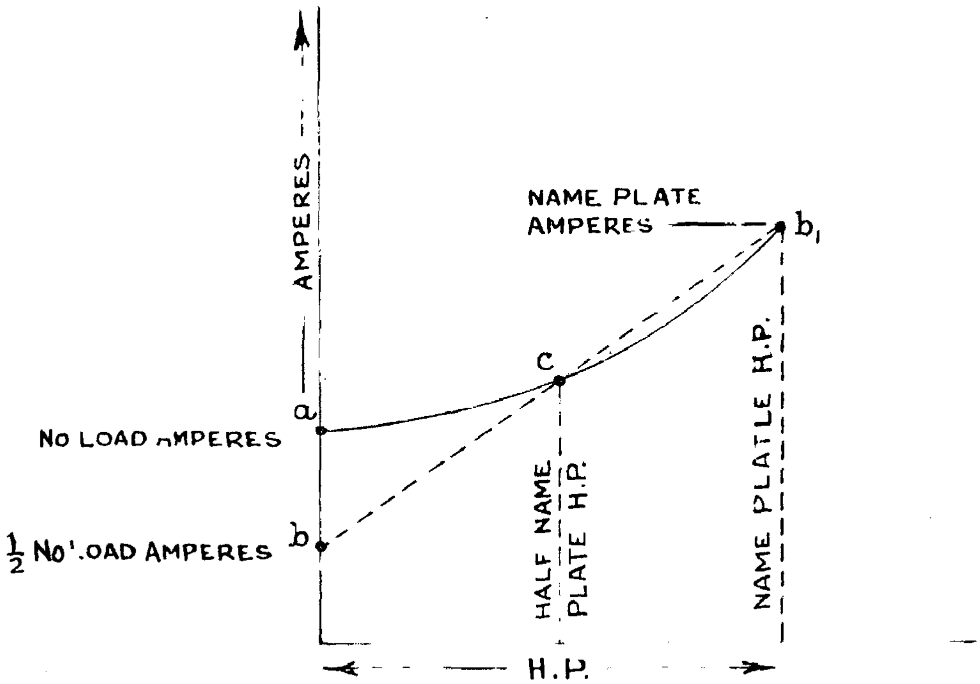
AMPERES
NAME PLATE AMPERES
b_1
c
a
NO LOAD AMPERES
b
$\frac{1}{2}$ NO LOAD AMPERES
HALF NAME PLATE H.P.
NAME PLATLE H.P.
H.P.

FIG. 19

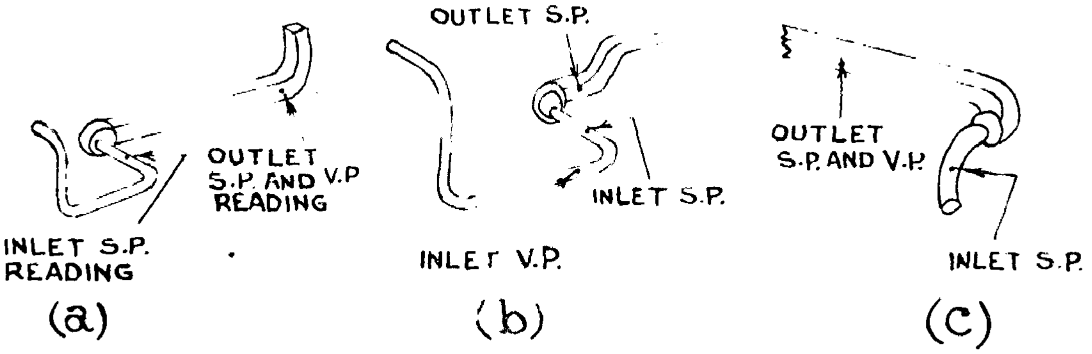
OUTLET S.P. AND V.P READING
INLET S.P. READING
(a)
OUTLET S.P.
INLET S.P.
INLET V.P.
(b)
OUTLET S.P. AND V.P.
INLET S.P.
(c)

FIG. 20

long is available, then the reading should be taken near the downstream end.

The static pressure can be measured with static connection of pitot tube if available otherwise a static tap has to be provided. (25)

APPROXIMATE METHOD OF DETERMINING HORSE POWER IN FIELD

To findk the H.P. input to the fan the following approximate method can be applied. Run the motor disconnected from the fan and find no load amperes. On the graph as in figure 19 locate point a corresponding to no load current. Locate point b corresponding to ½ no load current. Locate point b_1 corresponding to name plate amperes and name plate H.P. join b b_1 by a dotted line. On this line locate point c corresponding to half name plate H.P. Draw a smooth solid curve through the points a, c and b_1. (25)

The running H.P. can be read from this smooth solid curve corresponding to running amperes.

The barometric pressure may be obtained from local weather Bureau.

The wet and dry bulb readings should be taken in air stream.

The figures 20 (a, b and c) explain the choice of proper points in three given field systems. (25)

C H A P T E R IV

THE FAN TESTED

FAN UNDER TEST

THE COMPLETE TEST

CHAPTER - IV.

EXPERIMENT SET-UP

DESCRIPTION OF EXPERIMENTAL SET-UPS

The experimental apparatus was set up for testing an exhaust fan (Axial flow). The test was conducted for drawing the performance curves for the fan. The complete range of the test was obtained with three set-ups as shown in Figure 21.

The first set-up Fig. 21(a) was used for low volumes. The resistance to air flow was created with multi-hole perforated plates for which the arrangement of holes was as shown in Figures 22(c) and 22(d). The size of all the holes for any particular perforated plate was the same, but the different plates had holes of different diameters. The last test run made with this set-up was that with open duct. During each test run, the volume delivered by the fan was found out by pitot tube traverses. Test runs Nos. 1, 2, 3, 5 and 7 (see Appendix A) were made with this test set-up.

The second set up shown in figure 21(b) was made use for volumes which were too low for pitot tube measurements. In this set up the perforated plates (with smaller holes) were used as in first set up and a small duct piece carrying the thin rectangular orifice plate was further attached to it ~~further~~. The orifice tap was provided on this duct piece at a location 40 percent of duct diameter downstream from the orifice end (as recommended by Ebaugh and Whitefield for inlet orifices according to ref. 21). Test run No. 4 was made without any perforated plate to find out the

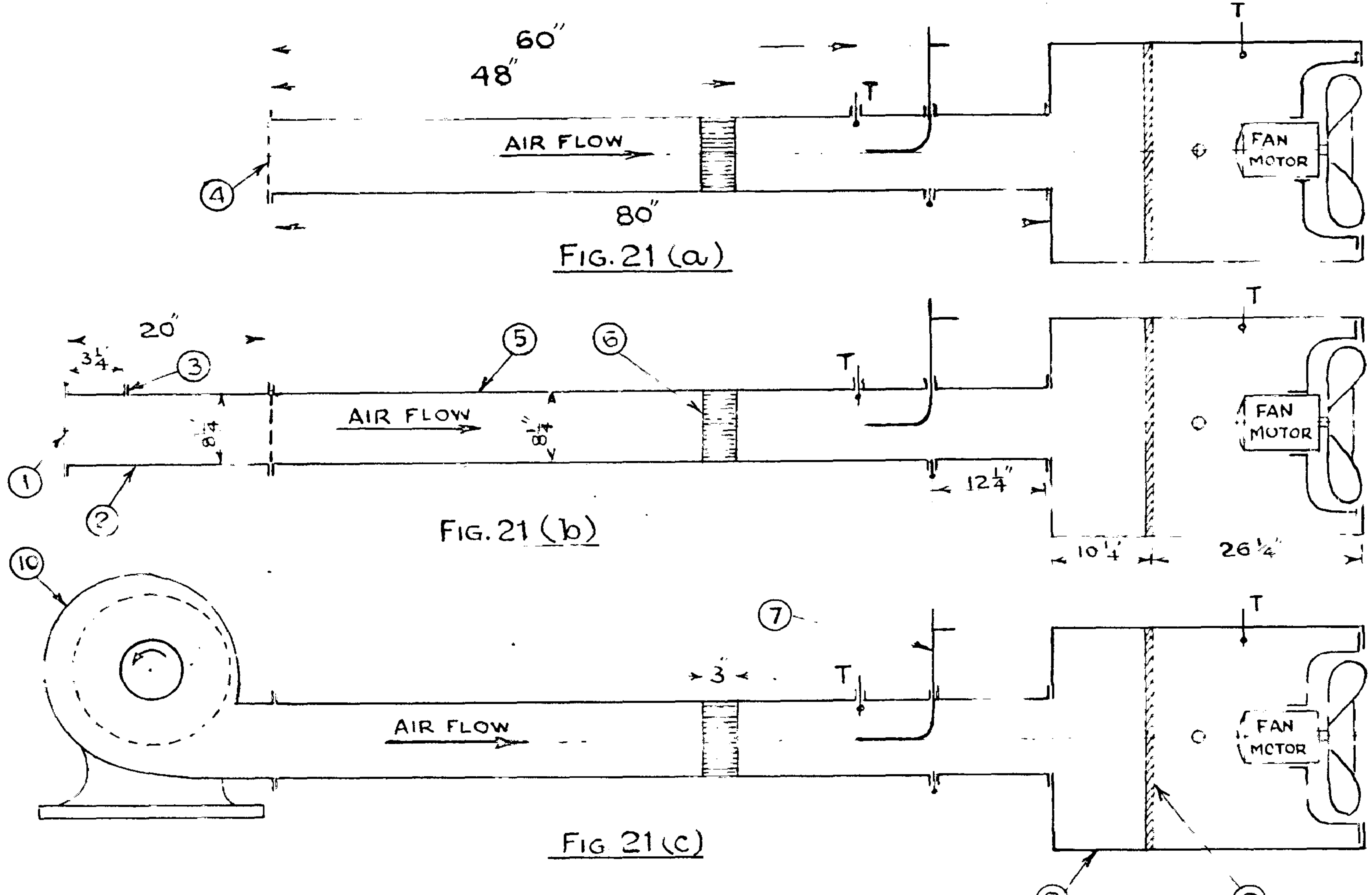

FIG. 21 TEST SET UPS

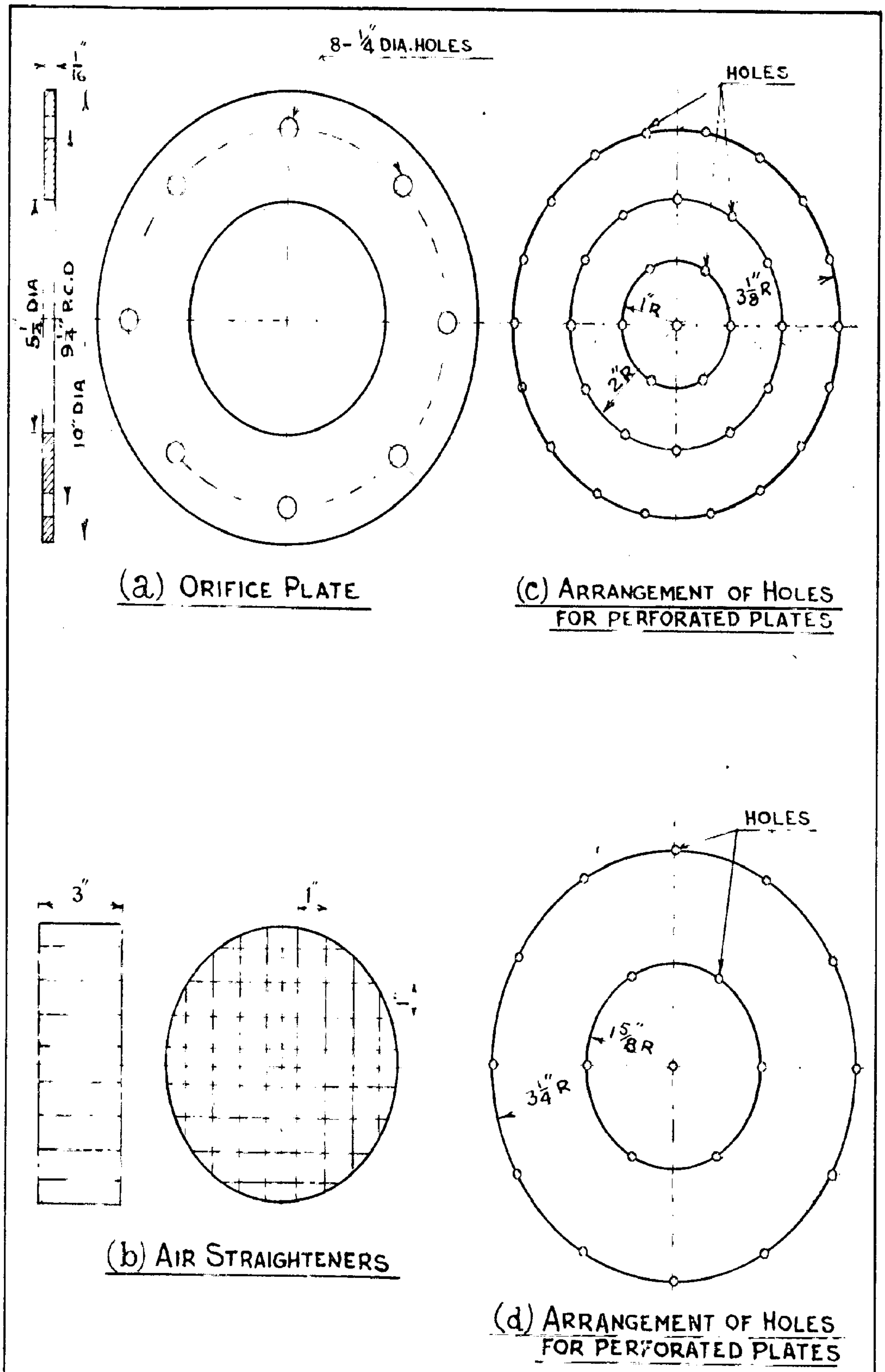
8- 1/4" DIA. HOLES
1/16"
5 1/4" DIA
9 1/4" P.C.D
10" DIA
(a) ORIFICE PLATE
HOLES
1" R
2" R
3 1/8" R
(c) ARRANGEMENT OF HOLES FOR PERFORATED PLATES
3"
1"
(b) AIR STRAIGHTENERS
HOLES
1 5/8" R
3 1/4" R
(d) ARRANGEMENT OF HOLES FOR PERFORATED PLATES

FIG. 22

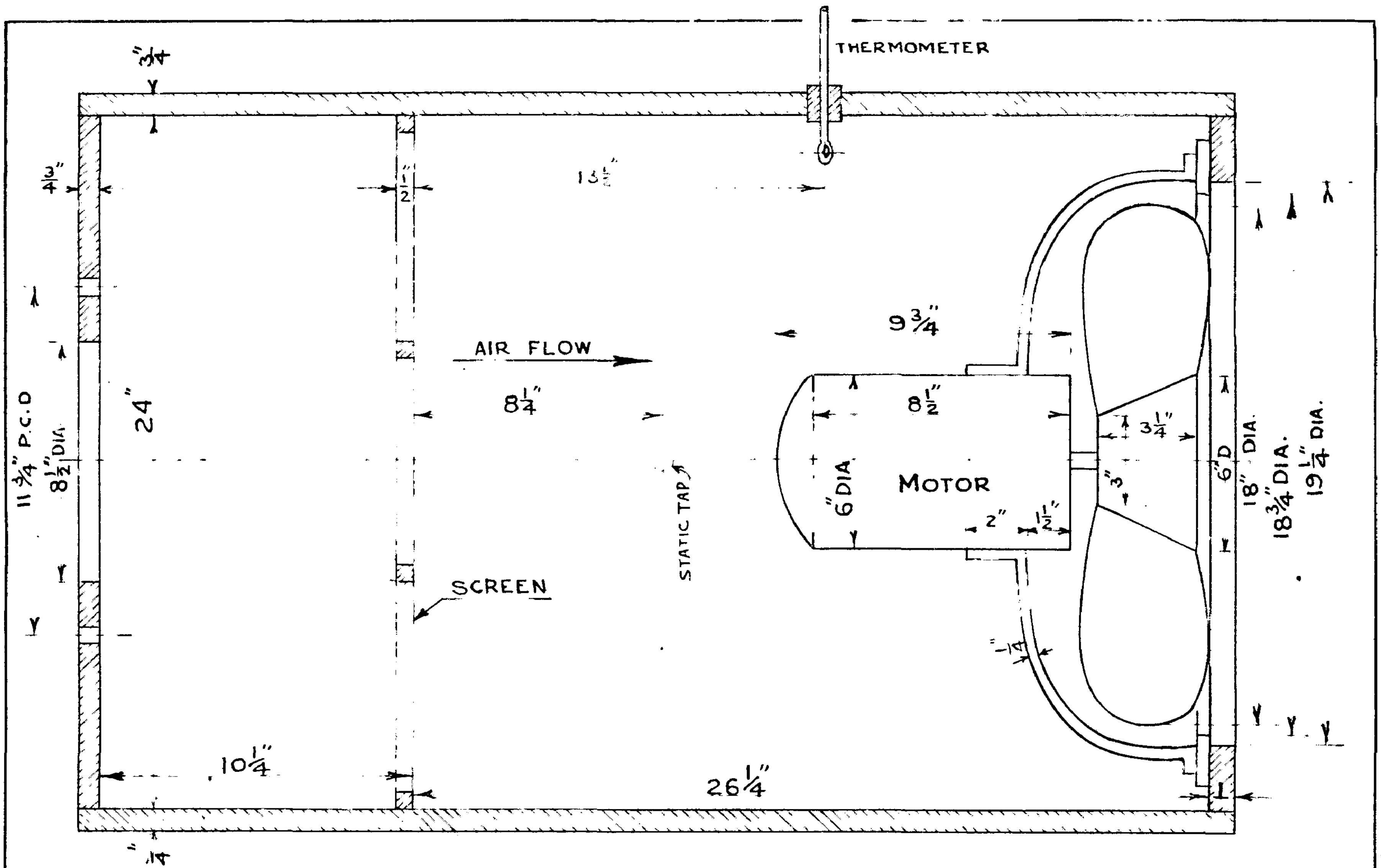

FIG. 23 PLENUM CHAMBER

discharge coefficient of the orifice. In this case the volume delivered was found out by making the pitot tube traverse. After finding out the discharge coefficient of the orifice, the next test runs Nos. 22 and 23 were made with perforated plates attached. The volume delivered by the fan was calculated from the differential pressure reading at orifice tap by using the value of discharge coefficient as previously found by test run No. 4.

The remaining of the test runs Nos. 6, 8 to 21, for higher volumes were made with the third set up as shown in figure 21(c). The auxilliary fan of this test set-up was driven by a D.C. motor. The electrical resistances were used both in the field and armature circuits of the motor to run it at variable speeds. For the first test run with this set-up the auxilliary fan motor was run at a slow speed. Then the motor speed was increased during each succeeding test run. This practice of increasing the speed of auxilliary fan was continued till it was felt that the speed be not increased further. Test runs Nos. 6, 8 to 12 were made thus.

The test fan was driven with an A.C. induction motor which was fed with stabilized voltage from a voltage stabilizer. In the test runs mentioned so far, the test fan motor was supplied with nearly 230 volts, (name plate voltage of motor).

Since the further increase in the speed of auxilliary fan was considered to be undesirable, therefore for the remaining range (test runs Nos. 13-21), the test fan was run at reduced speeds by supplying reduced voltage to the test fan motor through a variac. For all test runs of third set-up, the volume was measured by pitot tube traverses.

The no delivery test run was made with first set-up by attaching solid plate to the duct end. Although the apparatus was already made quite leak proof, but still to make it more air tight, corn flour paste was applied at the joints.

DESCRIPTION OF THE APPARATUS.

(1) Orifice Plate:- It was manufactured from mild steel sheet 1/16 inch thick. The finish of the orifice hole was obtained by turning it on the lathe machine. The edges on both sides were kept sharp.

(2) Smaller Duct:- It was made out of a mild steel sheet nearly 1/16 inches thick. The joints were obtained by welding to make it air tight. The flanges on the two sides were also welded to the straight length.

(3) Orifice Tap:- It was made out of a steel bar by turning it on the machine. It was welded to the smaller duct at a location 40 percent of the duct diameter downstream from the orifice plate end of the duct. Sufficient care was taken to remove the burrs around the hole, so that the flow pattern near the tap hole is not disturbed.

(4) Multi-hole perforated plates:- They were also constructed from mild steel sheets 1/16 inch thick. The arrangement of holes in one plate was as in Fig. 22(d) and the dia. of the holes was $\frac{7}{8}$ inch. The other four plates had the arrangement of holes as shown in Fig. 22(c) and the size of holes were $\frac{7}{32}$, $\frac{13}{32}$, $\frac{17}{32}$ and $\frac{3}{8}$ inches respectively.

(5) <u>Larger duct:-</u> It was also made from a mild steel sheet nearly $\frac{1}{16}$ inch thick. The joints ~~have been~~ were obtained by welding. It ~~is~~ was provided with holes $\frac{3}{8}$ inches diameter for inserting the pitot tube to take the readings. Small pieces of pipe were welded to the duct around these holes, so that the holes ~~may~~ could be kept closed by means of corks.

(6) <u>Air straighteners:-</u> The air straighteners ~~have been~~ were manufactured out of a sheet 0.026 inch thick, (which is quite close to the 22 U.S. gage sheet as recommended by A.S.M.E.).

(7) <u>Pitot tube:-</u> Figure 24 shows the pitot tube used in this experimentation work. The design of this tube ~~is~~ was based on the paper forwarded by Lionel S. Marks and published in Transactions of A.S.M.E. V57, n 7, Oct. 1935 page 430. (For more details see the standardization of pitot tube given in Appendix D).

The pitot tube used in the present experimentation work Figure 24 was manufactured in Roorkee University. The outside dia. of the tube was $\frac{5}{16}$ inch. The length of straight tubing between the static holes and the impact end was $2\frac{1}{2}$ inches (8 times the tube dia.), and the distance between the static holes and axis of stem was 5 inches (16 times the tube diameter). The size of impact hole was $\frac{1}{8}$ inch (0.4 of the tube dia.). The static holes were 8 in number and equally spaced around the circumsference. The diameter of the static holes was $\frac{1}{32}$ inch (0.03125 inch).

When the pitot tube axis is pointing correctly in the upstream direction then the impact reading will be having no error.

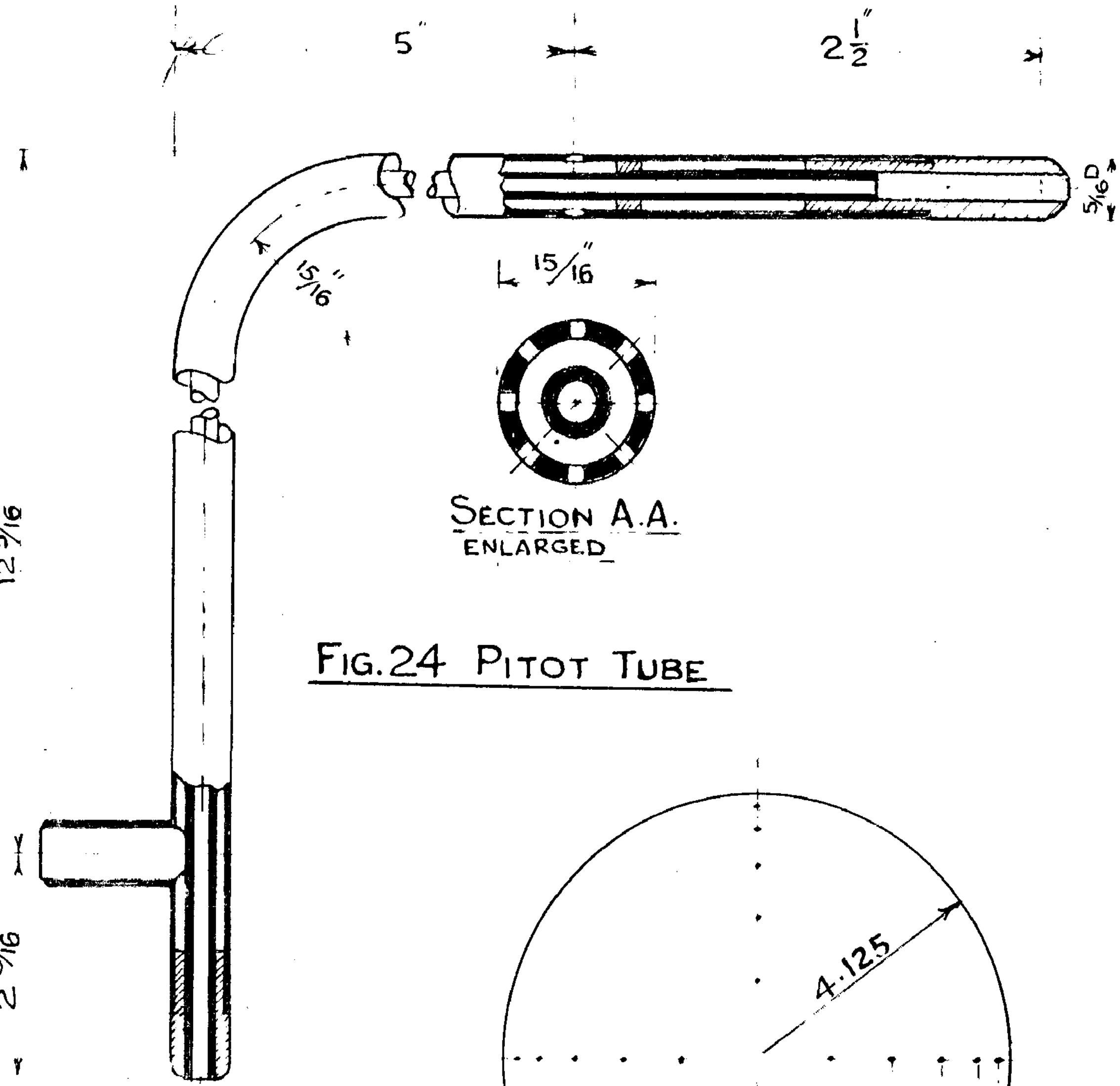

FIG. 24 PITOT TUBE

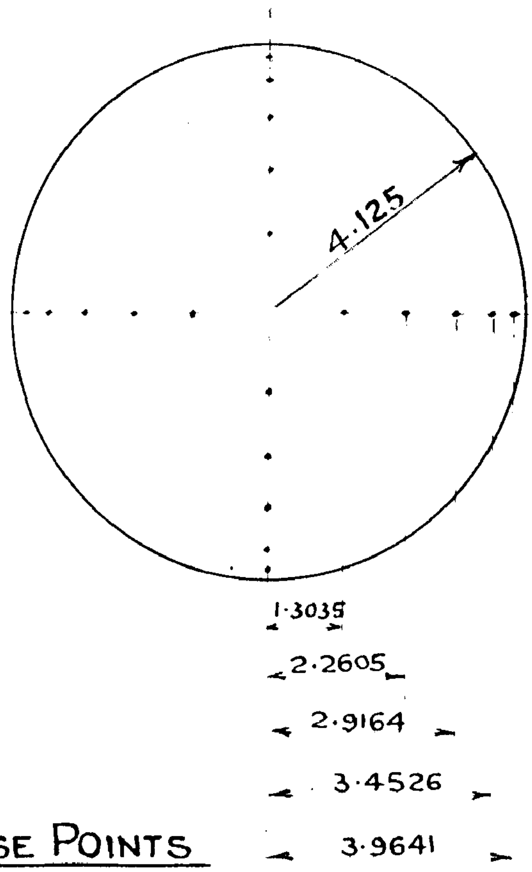

FIG. 25 TRAVERSE POINTS

If the distance of static holes is 8 tube diameters from the tip then the presence of tip reduces the static pressure reading by 0.2 percent of the velocity head according to experiments conducted by Ower. He further observed that if the distance of static holes from stem axis is 16 tube diameters, then the presence of stem increases the static pressure by 0.5 percent of velocity head. As a result the static pressure will be greater than its actual value by 0.3 percent (0.5 - 0.2 = 0.3) of the velocity head. Since velocity pressure is equal to impact reading minus static pressure reading, hence the velocity pressure given by the pitot tube will be less than its actual value by 0.3 percent. Consequently the pitot tube coefficient should be $\frac{1}{0.997}$ which is nearly equal to one. The American Society of Mechanical Engineers (A.S.M.E.) and the National Association of Fan Manufacturers (N.A.F.M.), also take the coefficient for their pitot tubes to be unity, although the pitot tubes used by the two are of different designs.

Proceeding fundamentally we know that

$$pv = \frac{\rho V^2}{2g}$$

where p_v = velocity pressure in lbs./sq.ft.

ρ = air density in lbs./cu.ft.

V = air velocity in feet/sec.

g = acceleration due to gravity in ft./sec^2.

If p'_v = velocity pressure in inches of water

then $$\frac{p'_v \times 62.3}{12} = \frac{\rho V^2}{2g}$$

where 62.3 = density of water at 70°F.

$$\therefore \quad V = \sqrt{\frac{62.3 \times 2g}{12}} \sqrt{\frac{p_v'}{\rho}} \text{ ft./sec.}$$

$$V = 60 \sqrt{\frac{62.3 \times 2g}{12}} \sqrt{\frac{p_v'}{\rho}} \text{ ft./mt.}$$

$$= 1096.2 \sqrt{\frac{p_v'}{\rho}} \text{ ft./mt.}$$

When the tube is not in alignment with fluid flow then both the static and impact pressure readings as indicated by pitot tube will be lower than their actual values. But no consideration can be given to it.

(8) Wooden box:- It was made from Deodar wood planks, Rubber packings were inserted in the joints to make it air tight. The inside dimensions ~~are~~ were 24" x 24" x 36½". The thickness of the side on which the fan ~~is~~ was attached ~~is~~ was one inch while the thickness of the remaining three sides ~~is~~ was ⅜ inch. It ~~is~~ was provided with a static tap at the centre of one face. This was used for measuring the static pressure at the plenum chamber. A 1 inch diameter hole ~~has been~~ was made in the top face for inserting the thermometer for taking the temperature at plenum chamber. The thermometer was attached with a cork. The same hole was also used for inserting the pitot tube into the plenum chamber for taking the total pressure reading at plenum chamber.

(9) Screen:- The screen was made from a mosquito-net cloth. The mosquito net cloth ~~has been~~ was attached to a wooden frame on both sides by means of shoe nails. An extra layer of cloth ~~has been~~ was

provided in the central portion of the screen to act as a baffle plate.

(10) <u>Auxilliary fan:-</u> It was a forward curved bladed centrifugal fan.

<u>MANOMETER</u>

The manometer used was an inclined U-tube manometer, with water as manometer liquid. The U-tube was fixed to a wooden stand. The wooden stand was provided with ordinary wooden screws acting as levelling screws. A fixed inclination of the U-tube could be maintained by levelling the wooden stand, with the help of a spirit level placed at its base.

The inclination of U-tube with horizontal was 5.75 degrees.

But $\sin 5.75^{\circ} = 0.1$

Therefore the range of the pressure readings was ten times of ~~its~~ the actual value.

<u>NAME PLATE DATA OF FAN MOTOR</u>

R.P.M. = 900

Amps = 0.7

Volts = 220/240

Size = 18"

Circuit = 50 $\sim$

H.P. = 0.1

<u>MEASUREMENTS</u>

<u>Barometric pressure:-</u> was measured with a Fortin's barometer. The least count of the instrument was 0.005 cms. Hg.

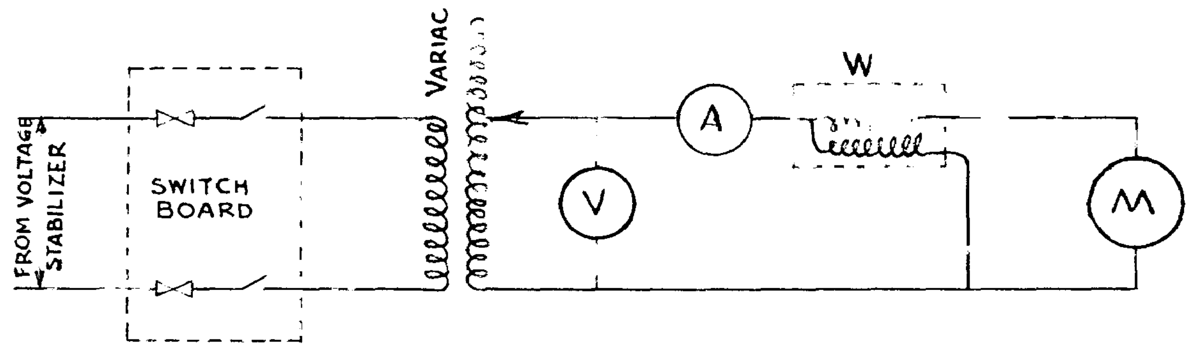

Fig. 26 Electrical Circuit for Test Fan Motor

Temperatures:- The temperatures were measured with mercury thermometers. All thermometers were graduated in degrees centigrade.

Speed of the test fan:- In the beginning the speed measurements were started with strobotac. After taking some readings it was found, that the strobotac was not giving reliable results. Therefore all the readings taken by that time had to be cancelled. After this the speed of the test fan was measured with a tachometer. The range of the tachometer was 0-2000 R.P.M. The tachometer was in a good state and was quite light in operation.

Electrical input to test fan motor:- The electrical circuit used for the test fan motor is shown in Fig. 26. The range of the ammeter was 0-1 Amperes. The range of the voltmeter was 0-300 volts. The range of wattmeter was 0-260 watts.

Traverse at the duct:- According to A.S.M.E. the distances of the fixed points at plane of traverse from the duct centre are 0.316R, 0.548R, 0.707R, 0.837R and 0.961R. For this experiment set-up, the value of duct radius R ~~is~~ was equal to 4.125 inches. Substituting this value of R the above distances will become to be 1.3035, 2.2605, 2.9164, 3.4526 and 3.9641 as shown in Fig. 25.

Measurement of pressures:-

The static pressure at plane of traverse was measured with the static connection of the pitot tube connected to one leg of the manometer while the other end of the manometer was communicating with atmosphere. The static pressure readings were taken at twenty fixed points as prescribed by A.S.M.E.(Fig. 25), and then the arithmetic average of these twenty readings was taken to be the static pressure at plane of traverse.

The static pressure at plenum chamber was measured by means of a static tap provided at the centre of one of the sides of plenum chamber.

The total pressure at plenum chamber was measured with the pitot tube. Before starting the test runs, the total pressure readings at different distances from the top face of the box were taken. The table below shows the values of total pressures for different distances of the pitot tube axis from the top face of the plenum chamber.

Distance of pitot tube axis from top face of plenum chamber	2"	3"	4"	5"	6"
Total pressure reading inches of water	0.275	0.25	0.224	0.212	0.206

Distance of pitot tube axis from top face of plenum chamber	7"	8"	9"	10"
Total pressure reading inches of water.	0.205	0.205	0.203	0.202

as [illegible]

From the table it can be seen that the total pressure at about 7 inches distance from top face, ~~gains a constant value.~~ Therefore for taking all the total pressure readings at plenum chamber the pitot tube axis was held at 7 inches distance from the top face.

Measurement of volume:- The volume delivered by the fan for the two lowest volume test runs was measured with the orifice. For remaining of the test runs, the volume delivered by the fan was measured with pitot tube traverses. The velocity pressures were taken at the twenty fixed points as in figure 25, and then the square of the average of square roots of velocity pressures at the twenty fixed points was taken to be the velocity pressure at plane of traverse.

Power input to fan:- The power input to fan was measured indirectly. During the regular test runs the electrical energy input to fan motor was measured. At the end the motor was tested for efficiency by applying the belt brake test. For brake test the springs used were quite sensitive. The range of springs was 0-25 lbs.

For all the test runs in which the fan motor was supplied with nearly 230 volts, the motor was tested for efficiency at nearly the same voltage.

For the test runs in which the test fan was run at reduced speed by supplying it with low voltage, the fan motor could not be tested for efficiency at the same voltage. Whenever the brake test was tried by applying the same voltage as used during the test run, the motor used to stop. Due to this difficulty the brake tests ~~have~~ were been carried out by supplying the fan motor with a voltage nearly 15 volts higher than that during the test run.

OBSERVATIONS

The following observations were taken during each test run:

1. Barometric pressure
2. Temperature at plenum chamber
3. Temperature at plane of traverse
4. Temperature of atmospheric air
5. Wet bulb temperature
6. Speed of fan
7. Current input to fan motor
8. Voltage input to fan motor
9. Watts input to fan motor
10. Static pressure at plane of traverse
11. Velocity pressure at plane of traverse
12. Static pressure at plenum chamber
13. Total pressure at plenum chamber.

The static and velocity pressure readings at plane of traverse were taken at 20 fixed points. These twenty readings were divided into 4 sets of 5 readings each. The different sets were taken by inserting the pitot tube from different sides of the duct. The other readings were repeated along with each set and then the average was taken.

All the readings taken during the different test runs are given in Appendix A.

CALCULATIONS

The calculations are based on the procedure discussed in Chapter III. Calculations for test runs Nos. 4, 9 and 23 are given in Appendix B as a specimen.

PERFORMANCE CURVES

From the brake test of fan motor the efficiency curve Fig. 27 was drawn for the test fan motor with power input in watts plotted along X-axis and the motor efficiency plotted along y-axis.

With the help of above curve, the fan power characteristic (fig. 28) was plotted with volume delivered by the fan along x-axis and the horse power input to fan along y-axis.

The fan static pressures and fan total pressures were also plotted versus volume delivered by fan (fig. 29).

The curves of figures 28 and 29 were retraced in figure 30 and by taking different points on these curves the fan static and total efficiencies were calculated and then the curves were drawn for these efficiencies.

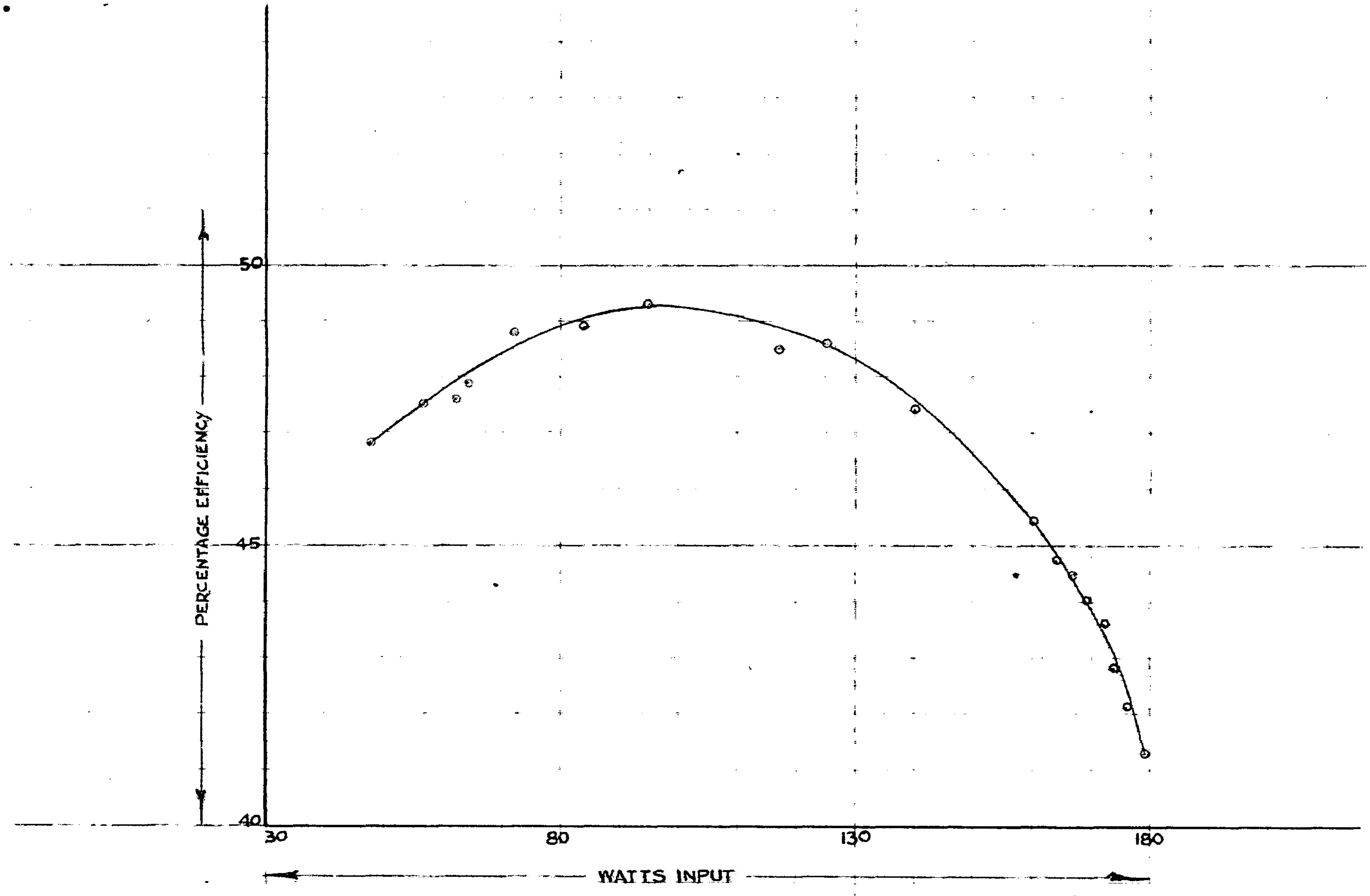

Fig. 27 Fan Motor Efficiency Curve

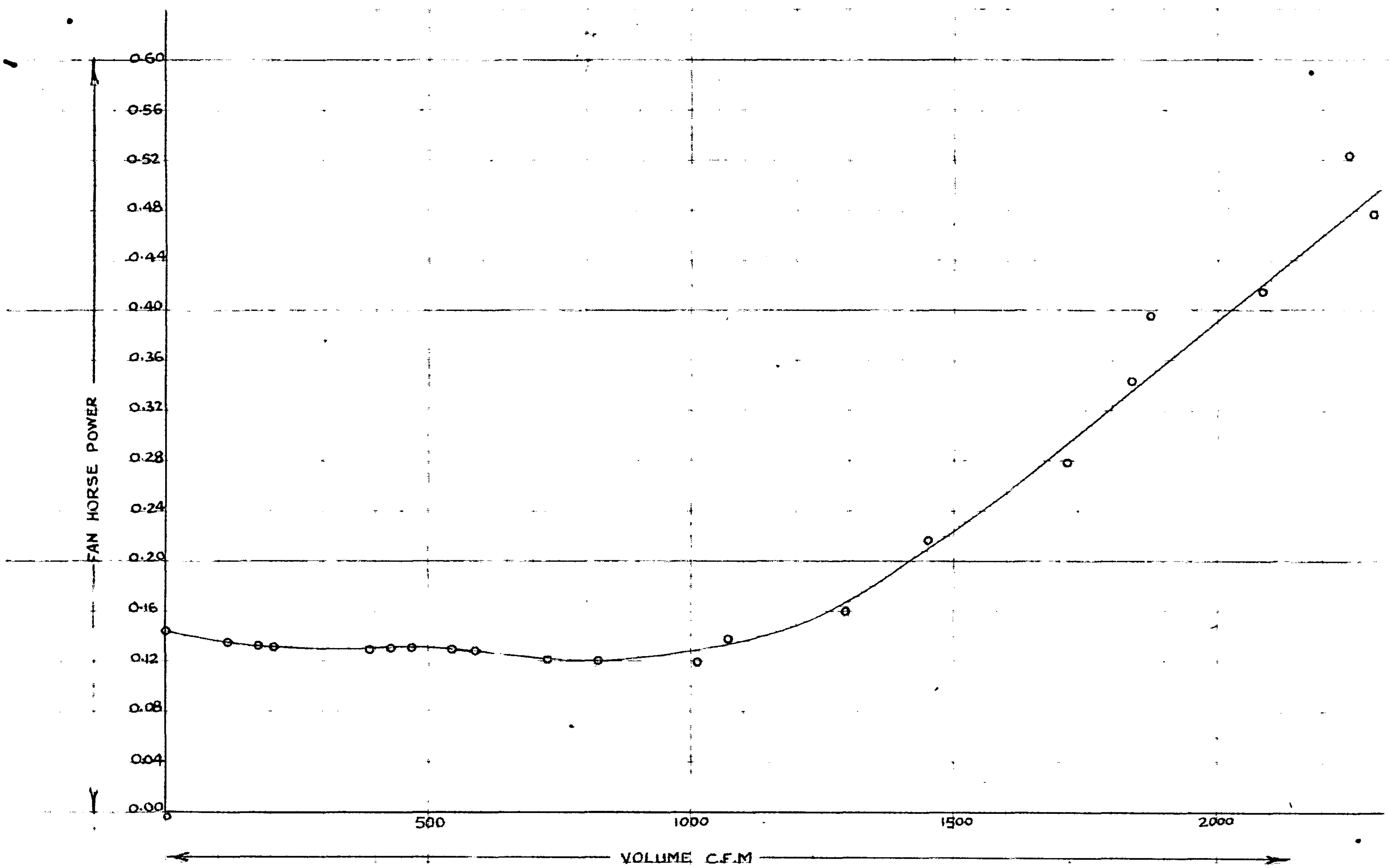

FIG. 28 POWER CHARACTERISTICS

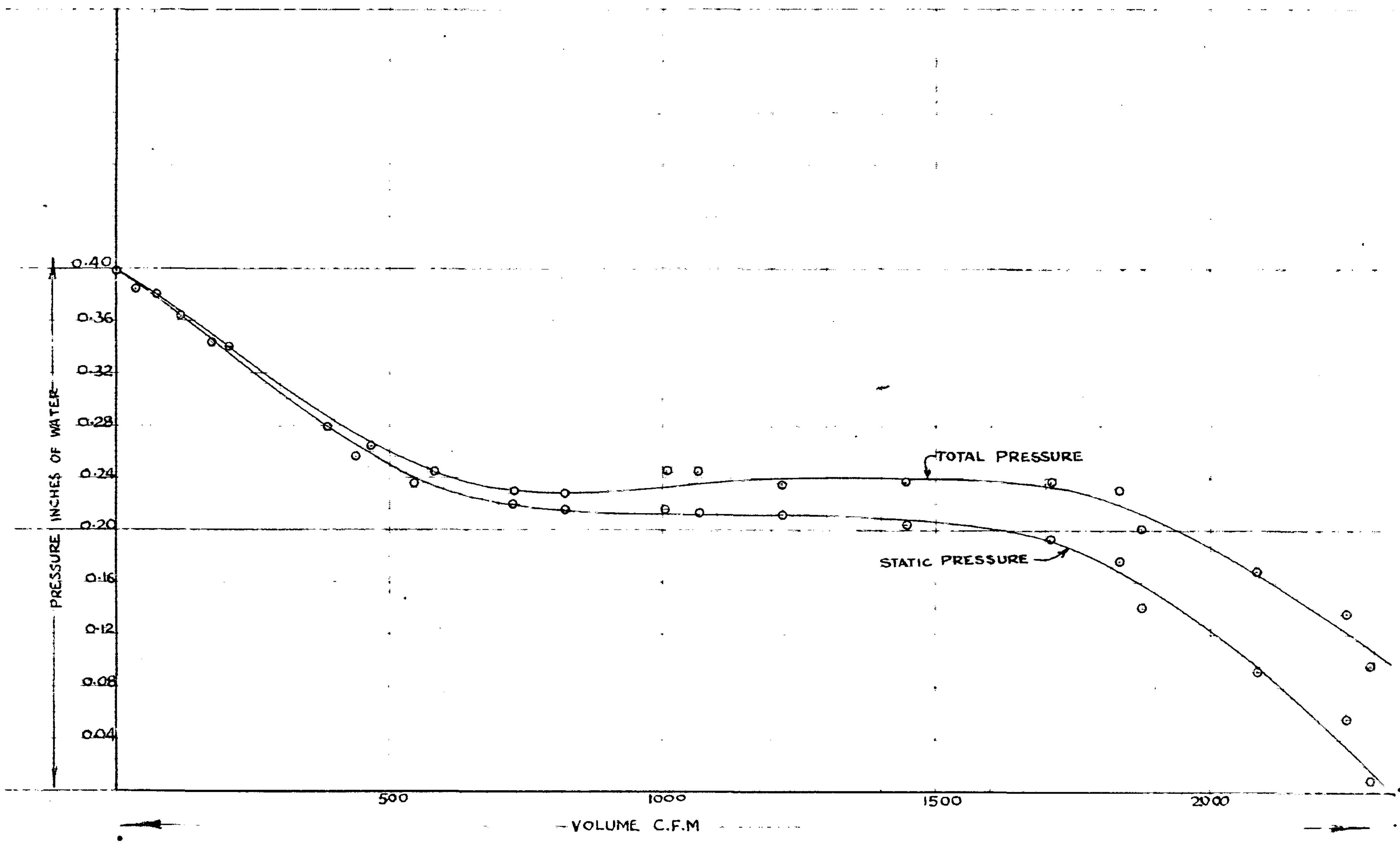

FIG. 29 PRESSURE CHARACTERISTICS

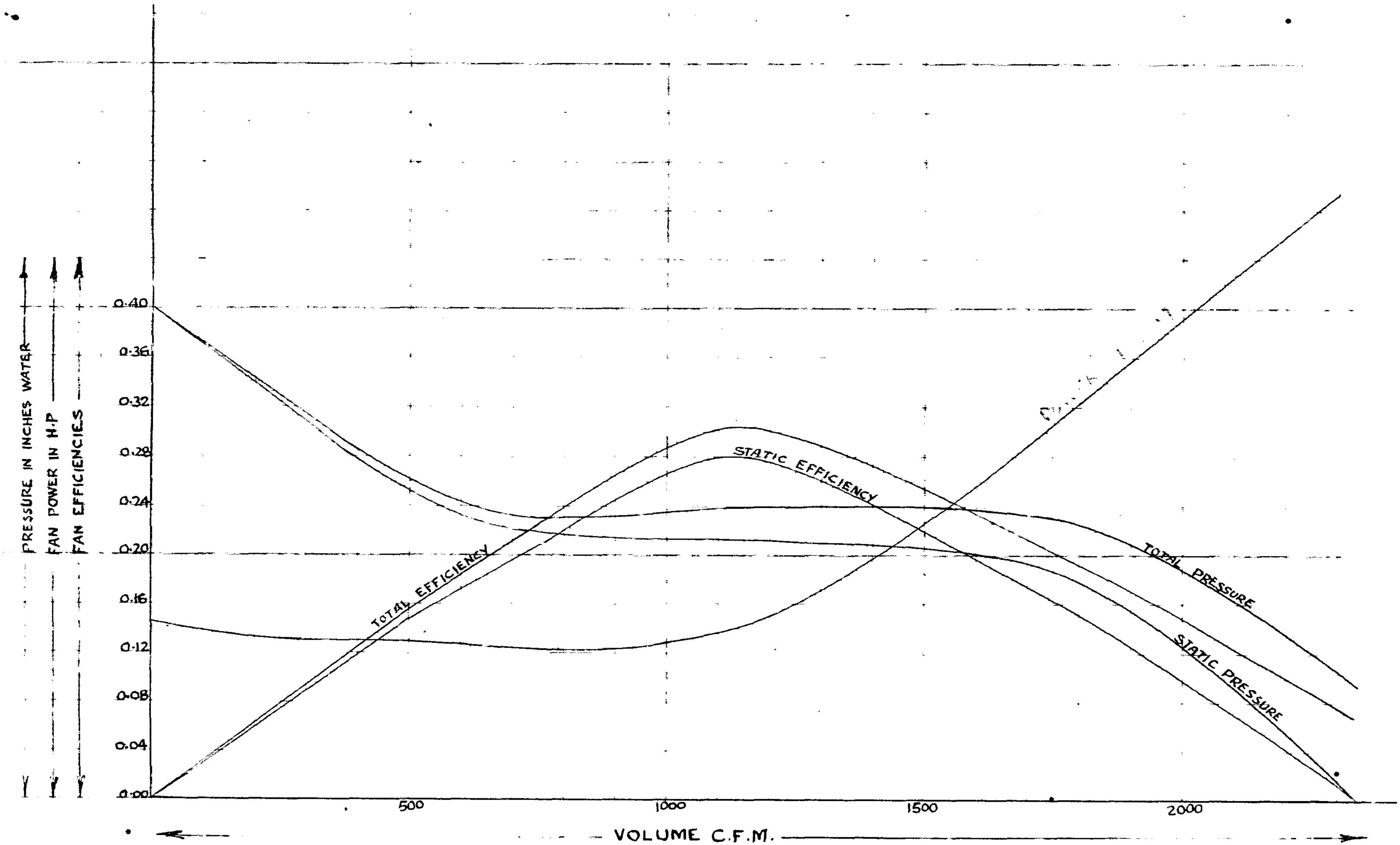

FIG. 30 FAN PERFORMANCE CURVES

Test run No. 1 Dated 27/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.06	74.06	74.06	74.05	74.06
Temp. at Plenum Chamber deg. C.			24.8	24.8	24.8	24.8	24.8
Temp. at Traverse deg. C.			24.2	24.2	24.3	24.3	24.35
Temp. of atmosphere deg. C.			23.8	23.9	23.9	24.0	23.9
Wet bulb Temp. Deg. C.			18.7	18.7	18.6	18.6	18.65
R.P.M. of fan.			813	811	813	811	812
Fan motor	Current in Amps.		0.792	0.790	0.790	0.789	0.790
	Voltage.		228	228	228	228	228
	Watts input.		178	178.4	178.4	178.4	178.3
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	00	00	00	00	00
		2.2605	00	00	00	00	
		2.9164	00	00	00	00	
		3.4526	00	00	00	00	
		3.9641	00	00	00	00	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	1.585	1.585	1.57	1.585	-1.5755
		2.2605	1.585	1.575	1.56	1.58	
		2.9164	1.575	1.585	1.56	1.58	
		3.4526	1.575	1.580	1.56	1.58	
		3.9641	1.575	1.575	1.56	1.58	
Static preesure reading at plenum chamber			3.02	3.055	3.045	3.07	-3.0475
Total pressure reading at plenum chamber			3.005	3.05	3.01	3.02	-3.021

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 00 inches water

Average Static Pressure at Plane of Traverse = -0.15755 inches of water

Average Static Pressure at Plenum Chamber = -0.30475 inches water

Average Total Pressure at Plenum Chamber = -0.3021 inches water.

Test run No. 2 Dated 28/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.19	74.195	74.195	74.20	74.195
Temp. at Plenum Chamber deg. C.			23.1	23.2	23.4	23.9	23.4
Temp. at Traverse deg. C.			22.0	22.1	22.2	22.9	22.3
Temp. of atmosphere deg. C.			22.2	22.4	22.7	23.0	22.58
Wet bulb Temp. Deg. C.			18.9	19.1	19.1	19.4	19.13
R.P.M. of fan.			827	827	827	825	~~826.5~~
Fan motor	Current in Amps.		0.777	0.776	0.774	0.774	0.7753
	Voltage.		228.4	229.0	228.0	228.0	228.35
	Watts input.		176	176.4	175.2	175.2	175.7
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	0.0575	0.050	0.055	0.050	0.04688
		2.2605	0.055	0.0475	0.050	0.050	
		2.9164	0.050	0.0425	0.050	0.050	
		3.4526	0.050	0.0375	0.0475	0.050	
		3.9641	0.040	0.0325	0.035	0.0425	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	1.415	1.455	1.445	1.425	-1.433
		2.2605	1.415	1.455	1.435	1.425	
		2.9164	1.420	1.455	1.440	1.405	
		3.4526	1.415	1.450	1.440	1.405	
		3.9641	1.430	1.465	1.450	1.415	
Static preesure reading at plenum chamber			2.94	2.94	2.97	2.98	-2.9575
Total pressure reading at plenum chamber			2.95	2.92	2.915	2.915	-2.925

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.004688 inches water

Average Static Pressure at Plane of Traverse = -0.1433 inches water

Average Static Pressure at Plenum Chamber = -0.29575 inches water

Average Total Pressure at Plenum Chamber = -0.2925 inches water

Test run No. 3 Dated 28/11/63

Set No.		1	2	3	4	Average
Barometric Pressure Cms. Hg.		74.225	74.225	74.225	74.225	74.225
Temp. at Plenum Chamber deg. C.		22.2	22.5	23.1	23.0	22.7
Temp. at Traverse deg. C.		21.2	21.5	22.3	22.0	21.75
Temp. of atmosphere deg. C.		22.1	22.2	22.5	22.3	22.28
Wet bulb Temp. Deg. C.		18.9	18.9	18.9	18.9	18.9
R.P.M. of fan.		833	833	832	830	832
Fan motor	Current in Amps.	0.774	0.770	0.768	0.768	0.770
	Voltage.	230.0	229.8	229.2	228.8	229.45
	Watts input.	175.2	174.8	174.4	173.6	174.5
Valocity Pressure reading at plane of Traverse — Pitot tube distance from centre in inches.	1.3035	0.1275	0.1225	0.1225	0.135	0.1118
	2.2605	0.120	0.1175	0.1125	0.135	
	2.9164	0.115	0.120	0.1125	0.1225	
	3.4526	0.105	0.1125	0.1025	0.115	
	3.9641	0.085	0.095	0.075	0.095	
Static pressure reading at plane of Traverse ( ) — Pitot tube distance from centre in inches.	1.3035	1.265	1.415	1.395	1.440	-1.371
	2.2605	1.265	1.425	1.405	1.435	
	2.9164	1.26	1.425	1.395	1.38	
	3.4526	1.26	1.425	1.395	1.385	
	3.9641	1.255	1.425	1.400	1.375	
Static preesure reading at plenum chamber		2.85	2.86	2.845	2.81	-2.841
Total pressure reading at plenum chamber		2.835	2.78	2.76	2.84	-2.804

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.01118 inches water

Average Static Pressure at Plane of Traverse = -0.1371 inches water

Average Static Pressure at Plenum Chamber = -0.2841 inches water

Average Total Pressure at Plenum Chamber = -0.2804 inches water

Test run No. 4 Dated 28/11/63

Set No.		1	2	3	4	Average
Barometric Pressure Cms. Hg.		74.02	74.02	74.02	74.02	74.02
Temp. at Plenum Chamber deg. C.		21.8	22.4	22.0	22.3	22.12
Temp. at Traverse deg. C.		21.2	22.0	21.4	21.8	21.6
Temp. of atmosphere deg. C.		21.4	21.9	22.2	22.2	21.93
Wet bulb Temp. Deg. C.		19.0	19.7	19.9	19.9	19.63
R.P.M. of fan.		836	838	877	837	837
Fan motor	Current in Amps.	0.770	0.767	0.766	0.771	0.7685
	Voltage.	229	229	229.9	230.4	229.58
	Watts input.	174.8	173.6	174.8	176.4	174.9
Valocity Pressure reading at plane of Traverse — Pitot tube distance from centre in inches.	1.3035	0.17	0.1675	0.17	0.17	0.1568
	2.2605	0.17	0.165	0.165	0.1625	
	2.9164	0.165	0.165	0.155	0.17	
	3.4526	0.16	0.15	0.145	0.165	
	3.9641	0.14	0.1325	0.12	0.135	
Static pressure reading at plane of Traverse ( ) — Pitot tube distance from centre in inches.	1.3035	1.435	1.395	1.415	1.425	-1.418
	2.2605	1.44	1.41	1.415	1.415	
	2.9164	1.44	1.42	1.42	1.405	
	3.4526	1.435	1.41	1.415	1.405	
	3.9641	1.44	1.395	1.41	1.42	
Static preesure reading at plenum chamber		2.93	2.935	2.89	2.88	-2.909
Total pressure reading at plenum chamber		2.81	2.84	2.80	2.79	-2.81
Differential pressure reading at orifice tap		2.425	2.44	2.415	2.42	2.425

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.01568 inches water

Average Static Pressure at Plane of Traverse = − 0.1418 inches water

Average Static Pressure at Plenum Chamber = − 0.2909 inches water

Average Total Pressure at Plenum Chamber = −0.281 inches water

Differential pressure at orifice tap = 0.2425 inches water.

Test run No. 5 Dated 29/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.18	74.20	74.19	74.20	74.193
Temp. at Plenum Chamber deg. C.			21.5	21.7	21.8	21.9	21.73
Temp. at Traverse deg. C.			21.3	21.6	21.7	21.9	21.63
Temp. of atmosphere deg. C.			22.0	21.8	22.0	22.3	22.03
Wet bulb Temp. Deg. C.			19.1	19.0	19.1	19.1	19.08
R.P.M. of fan.			841	837	836	836	837.5
Fan motor	Current in Amps.		0.764	0.763	0.766	0.762	0.7638
	Voltage.		229.2	228.6	228.5	228.0	228.6
	Watts input.		170.8	172.4	172	172	171.8
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	0.61	0.66	0.605	0.65	0.5614
		2.2605	0.575	0.595	0.60	0.68	
		2.9164	0.58	0.585	0.595	0.62	
		3.4526	0.495	0.575	0.45	0.60	
		3.9641	0.43	0.49	0.375	0.515	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	1.45	1.455	1.425	1.44	-1.435
		2.2605	1.445	1.45	1.42	1.445	
		2.9164	1.435	1.445	1.42	1.425	
		3.4526	1.435	1.44	1.42	1.43	
		3.9641	1.435	1.44	1.415	1.43	
Static preesure reading at plenum chamber			2.965	2.995	2.95	2.95	-2.965
Total pressure reading at plenum chamber			2.36	2.295	2.34	2.32	-2.326

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.05614 inches water

Average Static Pressure at Plane of Traverse = -0.1435 inches water

Average Static Pressure at Plenum Chamber = -0.2965 inches water

Average Total Pressure at Plenum Chamber = -0.2326 inches water

Test run No. 6 Dated 27/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.40	74.40	74.395	74.385	74.395
Temp. at Plenum Chamber deg. C.			23.1	23.5	23.8	23.9	23.58
Temp. at Traverse deg. C.			23.0	23.5	23.7	23.7	23.48
Temp. of atmosphere deg. C.			22.8	23.1	23.2	23.3	23.1
Wet bulb Temp. Deg. C.			18.8	19.1	19.0	19.0	18.98
R.P.M. of fan.			840	836	836	836	837
Fan motor	Current in Amps.		0.765	0.765	0.761	0.763	0.7635
	Voltage.		229.8	229.6	228.5	229.6	229.4
	Watts input.		172.8	173.0	172	173.6	172.85
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	0.835	0.89	0.91	0.858	0.8128
		2.2605	0.895	0.815	0.92	0.895	
		2.9164	0.978	0.905	1.00	0.795	
		3.4526	0.86	0.785	0.875	0.72	
		3.9641	0.65	0.59	0.58	0.60	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	1.485	1.495	1.485	1.495	-1.482
		2.2605	1.49	1.485	1.485	1.51	
		2.9164	1.485	1.495	1.485	1.45	
		3.4526	1.46	1.485	1.485	1.46	
		3.9641	1.47	1.48	1.48	1.48	
Static preesure reading at plenum chamber			2.89	2.81	2.80	2.84	-2.835
Total pressure reading at plenum chamber			2.195	2.195	2.15	2.155	-2.174

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.08128 inches water

Average Static Pressure at Plane of Traverse = -0.1482 inches water

Average Static Pressure at Plenum Chamber = -0.2835 inches water

Average Total Pressure at Plenum Chamber = -0.2174 inches water

Test run No. 7

Dated 27/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.025	74.02	74.02	74.015	74.02
Temp. at Plenum Chamber deg. C.			24.0	24.2	23.9	24.0	24.03
Temp. at Traverse deg. C.			24.1	24.1	23.8	24.0	24
Temp. of atmosphere deg. C.			24.1	24.1	23.8	23.8	23.95
Wet bulb Temp. Deg. C.			18.8	18.9	18.7	18.7	18.78
R.P.M. of fan.			836	839	836	836	836.75
Fan motor	Current in Amps.		0.761	0.758	0.760	0.760	0.7598
	Voltage.		228.0	228.2	228.0	228.0	228.05
	Watts input.		170	170.8	171.2	171.4	170.85
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	0.985	0.955	1.03	1.105	0.8567
		2.2605	0.965	0.93	0.915	1.19	
		2.9164	0.93	0.91	0.87	0.885	
		3.4526	0.83	0.785	0.755	0.805	
		3.9641	0.595	0.66	0.53	0.665	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	1.485	1.455	1.445	1.42	-1.449
		2.2605	1.485	1.445	1.44	1.425	
		2.9164	1.485	1.45	1.44	1.415	
		3.4526	1.485	1.445	1.44	1.42	
		3.9641	1.485	1.44	1.445	1.42	
Static preesure reading at plenum chamber			2.83	2.82	2.79	2.78	-2.805
Total pressure reading at plenum chamber			2.145	2.11	2.08	2.09	-2.106

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.08567 inches water

Average Static Pressure at Plane of Traverse = -0.1449 inches water

Average Static Pressure at Plenum Chamber = -0.2805 inches water

Average Total Pressure at Plenum Chamber = -0.2106 inches water

Test run No. 8

Dated 23/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			73.85	73.81	73.83	73.80	73.823
Temp. at Plenum Chamber deg. C.			25.4	25.5	25.5	25.6	25.5
Temp. at Traverse deg. C.			25.5	25.5	25.5	25.7	25.55
Temp. of atmosphere deg. C.			24.9	24.8	24.8	24.8	24.83
Wet bulb Temp. Deg. C.			18.3	18.3	18.4	18.6	18.4
R.P.M. of fan.			836	844	838	838	839
Fan motor	Current in Amps.		0.754	0.754	0.747	0.749	0.751
	Voltage.		230.0	230.1	228.0	229	229.3
	Watts input.		172	169.6	168	170.8	170.1
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	1.33	1.122	1.45	1.18	1.1145
		2.2605	1.37	1.035	1.365	1.17	
		2.9164	1.355	1.12	1.36	1.055	
		3.4526	1.065	0.985	1.13	0.865	
		3.9641	1.06	0.72	0.91	0.825	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	1.18	1.215	1.235	1.34	-1.2570
		2.2605	1.195	1.21	1.24	1.385	
		2.9164	1.19	1.205	1.24	1.345	
		3.4526	1.195	1.26	1.25	1.35	
		3.9641	1.20	1.245	1.305	1.355	
Static preesure reading at plenum chamber			2.58	2.725	2.845	2.74	-2.7225
Total pressure reading at plenum chamber			1.83	1.875	2.02	2.0075	-1.933

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.11145 inches water

Average Static Pressure at Plane of Traverse = -0.1257 inches water

Average Static Pressure at Plenum Chamber = -0.27225 inches water

Averagg Total Pressure at Plenum Chamber = -0.1933 inches water

Test run No. 9 Dated 25/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.25	74.25	74.25	74.25	74.25
Temp. at Plenum Chamber deg. C.			22.8	22.8	22.8	23.0	22.85
Temp. at Traverse deg. C.			22.7	22.6	22.9	23.0	22.8
Temp. of atmosphere deg. C.			22.6	22.6	22.6	22.8	22.65
Wet bulb Temp. Deg. C.			18.0	17.0	17.2	17.3	17.38
R.P.M. of fan.			840	836	836	835	836.75
Fan motor	Current in Amps.		0.759	0.753	0.754	0.754	0.755
	Voltage.		227.2	226.6	226.0	226.0	226.45
	Watts input.		169.6	168.8	168	168	168.6
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	1.46	1.365	1.525	1.215	1.2534
		2.2605	1.52	1.28	1.605	1.195	
		2.9164	1.595	1.28	1.15	1.095	
		3.4526	1.26	1.105	1.29	1.045	
		3.9641	1.04	1.34	0.99	0.87	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	1.375	1.425	1.405	1.41	-1.392
		2.2605	1.375	1.42	1.40	1.405	
		2.9164	1.375	1.415	1.405	1.365	
		3.4526	1.365	1.415	1.40	1.345	
		3.9641	1.37	1.40	1.41	1.355	
Static preesure reading at plenum chamber			2.77	2.83	2.76	2.865	-2.804
Total pressure reading at plenum chamber			1.95	1.975	2.00	2.020	-1.989

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.12534 inches water

Average Static Pressure at Plane of Traverse = -0.1392 inches water

Average Static Pressure at Plenum Chamber = -0.2804 inches water

Average Total Pressure at Plenum Chamber = -0.1989 inches water

Test run No. 10 Dated 25/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.2	74.215	74.210	74.12	74.186
Temp. at Plenum Chamber deg. C.			23.9	24.1	24.4	24.9	24.33
Temp. at Traverse deg. C.			23.9	24.1	24.4	24.9	24.33
Temp. of atmosphere deg. C.			23.6	23.8	24.0	24.3	23.93
Wet bulb Temp. Deg. C.			17.8	17.4	17.9	18.1	17.8
R.P.M. of fan.			854	854	854	854	854
Fan motor	Current in Amps.		0.739	0.740	0.739	0.740	0.7395
	Voltage.		228.4	228.3	228	228	228.2
	Watts input.		166.8	166.6	166.2	167.0	166.65
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	2.68	2.465	2.475	2.035	2.0258
		2.2605	2.745	2.39	2.30	1.97	
		2.9164	2.725	2.135	2.285	1.72	
		3.4526	2.30	1.62	1.91	1.60	
		3.9641	1.755	1.175	1.545	1.235	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	1.175	1.10	1.105	1.115	-1.106
		2.2605	1.165	1.085	1.085	1.135	
		2.9164	1.155	1.095	1.08	1.07	
		3.4526	1.155	1.09	1.08	1.07	
		3.9641	1.155	1.06	1.075	1.06	
Static preesure reading at plenum chamber			2.435	2.405	2.445	2.44	-2.431
Total pressure reading at plenum chamber			1.892	1.870	1.895	1.885	-1.885

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.20258 inches water

Average Static Pressure at Plane of Traverse = -0.1106 inches water

Average Static Pressure at Plenum Chamber = -0.2431 inches water

Average Total Pressure at Plenum Chamber = -0.1885 inches water

Test run No. 11

Dated 25/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			73.94	73.95	73.92	73.89	73.925
Temp. at Plenum Chamber deg. C.			25.0	25.1	25.2	25.1	25.1
Temp. at Traverse deg. C.			25.0	25.1	25.2	25.1	25.1
Temp. of atmosphere deg. C.			24.4	24.5	24.4	24.4	24.42
Wet bulb Temp. Deg. C.			18.3	18.5	18.3	18.3	18.35
R.P.M. of fan.			858	859	858	858	858.25
Fan motor	Current in Amps.		0.730	0.728	0.729	0.730	0.7293
	Voltage.		228.2	227.9	228.0	228.0	228.02
	Watts input.		163.4	162	163.4	163.8	163.15
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	3.335	3.095	3.05	2.845	2.651
		2.2605	3.48	2.825	3.005	2.685	
		2.9164	3.00	2.85	2.955	2.30	
		3.4526	2.99	2.275	2.53	2.24	
		3.9641	2.35	1.77	2.09	1.98	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	0.92	0.88	0.87	0.90	-0.866
		2.2605	0.91	0.875	0.86	0.885	
		2.9164	0.91	0.85	0.845	0.835	
		3.4526	0.895	0.82	0.845	0.835	
		3.9641	0.885	0.825	0.84	0.835	
Static preesure reading at plenum chamber			1.795	1.76	1.77	1.72	-1.761
Total pressure reading at plenum chamber			1.83	1.86	1.86	1.82	-1.842

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.2651 inches water

Average Static Pressure at Plane of Traverse = -0.0866 inches water

Average Static Pressure at Plenum Chamber = -0.1761 inches water

Average Total Pressure at Plenum Chamber = -0.1842 inches water

Test run No. 12 Dated 22/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			73.945	73.950	73.90	73.90	73.924
Temp. at Plenum Chamber deg. C.			26.1	26.0	26.1	26.0	26.15
Temp. at Traverse deg. C.			26.2	26.1	26.2	26.0	26.13
Temp. of atmosphere deg. C.			25.2	25.2	25.3	25.1	25.2
Wet bulb Temp. Deg. C.			19.2	19.1	18.9	18.9	19.03
R.P.M. of fan.			869	863	862	862	864
Fan motor	Current in Amps.		0.729	0.731	0.732	0.733	0.7313
	Voltage.		229	228.8	228	228	228.45
	Watts input.		162.4	164.2	163.6	163.6	163.45
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	5.20	4.165	4.675	3.715	
		2.2605	5.105	3.795	4.63	3.915	
		2.9164	5.165	3.725	4.88	3.335	3.9655
		3.4526	4.28	3.87	4.105	3.185	
		3.9641	3.23	3.095	3.325	2.52	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	0.63	0.67	0.635	0.73	
		2.2605	0.63	0.615	0.625	0.705	
		2.9164	0.605	0.59	0.60	0.585	-0.612
		3.4526	0.59	0.56	0.59	0.575	
		3.9641	0.59	0.535	0.595	0.585	
Static preesure reading at plenum chamber			2.125	2.075	2.05	2.045	-2.074
Total pressure reading at plenum chamber			1.94	1.898	1.83	1.84	-1.877

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.39655 inches water

Average Static Pressure at Plane of Traverse = -0.0612 inches water

Average Static Pressure at Plenum Chamber = -0.2074 inches water

Average Total Pressure at Plenum Chamber = -0.1877 inches water

Test run No. 13 Dated 21/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.03	74.01	74.02	74.02	74.02
Temp. at Plenum Chamber deg. C.			25.3	25.4	25.5	25.4	25.4
Temp. at Traverse deg. C.			25.5	25.7	25.6	25.5	25.58
Temp. of atmosphere deg. C.			24.6	24.8	24.8	24.8	24.75
Wet bulb Temp. Deg. C.			18.8	18.9	18.9	18.9	18.88
R.P.M. of fan.			798	797	795	796	796.5
Fan motor	Current in Amps.		0.721	0.720	0.720	0.725	0.7215
	Voltage.		204	203.4	203.2	204	203.65
	Watts input.		140.8	140	141.2	142	141
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre *in inches.*	1.3035	4.78	4.085	4.425	3.885	3.7912
		2.2605	4.81	3.975	4.385	3.645	
		2.9164	4.88	3.91	4.325	3.355	
		3.4526	3.95	3.465	3.755	3.065	
		3.9641	3.95	2.76	2.955	2.595	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre *in inches.*	1.3035	0.465	0.46	0.435	0.455	-0.4164
		2.2605	0.435	0.44	0.45	0.47	
		2.9164	0.425	0.40	0.425	0.37	
		3.4526	0.425	0.395	0.405	0.305	
		3.9641	0.425	0.36	0.39	0.38	
Static preesure reading at plenum chamber			1.82	1.755	1.86	1.78	-1.804
Total pressure reading at plenum chamber			1.55	1.605	1.615	1.565	-1.584

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.37912 inches water

Average Static Pressure at Plane of Traverse = -0.04164 inches water

Average Static Pressure at Plenum Chamber = -0.1804 inches water

Average Total Pressure at Plenum Chamber = -0.1584 inches water

Test run No. 14

Dated 21/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.02	74.00	73.92	73.94	73.97
Temp. at Plenum Chamber deg. C.			25.7	25.7	25.6	25.6	25.65
Temp. at Traverse deg. C.			25.9	25.9	25.9	25.9	25.9
Temp. of atmosphere deg. C.			24.8	24.8	24.8	24.8	24.8
Wet bulb Temp. Deg. C.			19.1	19.3	19.3	19.4	19.28
R.P.M. of fan.			715	720	715	722	718
Fan motor	Current in Amps.		0.695	0.696	0.695	0.692	0.6945
	Voltage.		181.2	181.9	181.8	180.6	181.38
	Watts input.		117.2	119	117.6	116.4	117.55
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	4.79	4.15	4.86	4.035	3.9446
		2.2605	4.90	3.60	4.61	3.90	
		2.9164	4.84	3.84	4.355	3.71	
		3.4526	4.015	3.405	4.395	3.255	
		3.9641	3.61	2.77	3.72	2.68	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	-0.07	-0.03	-0.03	-0.035	-0.0145
		2.2605	-0.06	-0.015	-0.025	-0.025	
		2.9164	-0.05	-0.005	-0.005	-0.01	
		3.4526	-0.035	+0.015	+0.015	000	
		3.9641	-0.035	+0.07	+0.02	+0.02	
Static preesure reading at plenum chamber			1.445	1.52	1.505	1.492	-1.491
Total pressure reading at plenum chamber			1.242	1.285	1.288	1.225	-1.260

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.39446 inches water

Average Static Pressure at Plane of Traverse = -0.00145 inches water

Average Static Pressure at Plenum Chamber = -0.1491 inches water

Average Total Pressure at Plenum Chamber = -0.1260 inches water

Test run No. 15 Dated 21/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.31	74.31	74.32	74.28	74.305
Temp. at Plenum Chamber deg. C.			24.3	23.4	23.0	23.6	23.58
Temp. at Traverse deg. C.			23.9	23.1	23.8	23.7	23.63
Temp. of atmosphere deg. C.			23.8	23.2	23.2	23.3	23.38
Wet bulb Temp. Deg. C.			18.2	18.8	18.7	18.7	18.6
R.P.M. of fan.			595	606	605	605	602.75
Fan motor	Current in Amps.		0.660	0.654	0.654	0.655	0.6558
	Voltage.		159.8	159.4	159.2	159.6	159.5
	Watts input.		95.6	95.2	94.4	94.8	95.00
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	4.59	4.73	4.615	4.16	3.7750
		2.2605	4.715	4.14	4.785	3.93	
		2.9164	4.445	3.785	4.44	3.755	
		3.4526	3.36	2.845	4.085	3.425	
		3.9641	2.59	2.185	3.055	2.73	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	0.235	0.215	0.235	0.22	+0.2595
		2.2605	0.235	0.24	0.275	0.215	
		2.9164	0.26	0.31	0.27	0.195	
		3.4526	0.28	0.325	0.295	0.21	
		3.9641	0.29	0.36	0.30	0.225	
Static pressure reading at plenum chamber			1.06	1.14	1.18	1.12	-1.125
Total pressure reading at plenum chamber			0.855	0.865	0.875	0.885	-0.87

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.3775 inches water

Average Static Pressure at Plane of Traverse = +0.02595 inches water

Average Static Pressure at Plenum Chamber = -0.1125 inches water

Average Total Pressure at Plenum Chamber = -0.087 inches water

Test run No. 16 Dated 26/11/63

Set No.		1	2	3	4	Average
Barometric Pressure Cms. Hg.		74.19	74.19	74.13	74.13	74.16
Temp. at Plenum Chamber deg. C.		24.7	24.7	24.7	24.8	24.73
Temp. at Traverse deg. C.		24.8	24.8	24.8	24.9	24.83
Temp. of atmosphere deg. C.		23.6	23.7	23.5	23.6	23.6
Wet bulb Temp. Deg. C.		19.7	19.7	19.4	19.7	19.63
R.P.M. of fan.		536	535	542	535	537
Fan motor	Current in Amps.	0.635	0.632	0.630	0.630	0.6318
	Voltage.	150.8	150.0	150.0	150.0	150.2
	Watts input.	85.6	84.4	84.6	84.6	84.8
Valocity Pressure reading at plane of Traverse — Pitot tube distance from centre in inches.	1.3035	4.90	4.575	4.78	4.205	3.9414
	2.2605	4.79	3.775	4.805	4.055	
	2.9164	4.75	4.155	4.645	3.51	
	3.4526	4.015	3.585	4.190	3.14	
	3.9641	3.45	2.56	3.085	2.575	
Static pressure reading at plane of Traverse ( ) — Pitot tube distance from centre in inches.	1.3035	0.68	0.685	0.64	0.59	+0.687
	2.2605	0.685	0.715	0.645	0.635	
	2.9164	0.70	0.735	0.66	0.685	
	3.4526	0.71	0.74	0.675	0.69	
	3.9641	0.71	0.775	0.685	0.705	
Static preesure reading at plenum chamber		0.917	0.915	0.907	0.905	-0.911
Total pressure reading at plenum chamber		0.65	0.657	0.64	0.657	-0.651

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.39414 inches water

Average Static Pressure at Plane of Traverse = +0.0687 inches water

Average Static Pressure at Plenum Chamber = -0.0911 inches water

Average Total Pressure at Plenum Chamber = -0.0651 inches water

Test run No. 17 Dated 21/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.295	74.02	74.015	74.13	74.115
Temp. at Plenum Chamber deg. C.			23.9	25.3	24.9	24.2	24.58
Temp. at Traverse deg. C.			24.2	25.5	24.9	24.5	25.78
Temp. of atmosphere deg. C.			23.6	24.6	23.9	23.9	24.00
Wet bulb Temp. Deg. C.			18.9	18.9	19.0	19.1	18.98
R.P.M. of fan.			476	473	474	468	472.75
Fan motor	Current in Amps.		0.593	0.595	0.593	0.592	0.5933
	Voltage.		140	140.6	140	139.8	140.1
	Watts input.		73.4	72.8	73.0	73.2	73.1
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	4.615	4.325	4.41	4.08	3.7142
		2.2605	4.60	3.70	4.485	3.945	
		2.9164	4.52	3.59	4.11	3.37	
		3.4526	3.365	3.525	3.96	3.385	
		3.9641	2.94	2.76	2.72	2.47	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	0.67	0.61	0.615	0.62	0.6544
		2.2605	0.685	0.62	0.615	0.605	
		2.9164	0.67	0.645	0.62	0.685	
		3.4526	0.725	0.655	0.63	0.695	
		3.9641	0.685	0.675	0.635	0.73	
Static preesure reading at plenum chamber			0.715	0.685	0.705	0.705	-0.7025
Total pressure reading at plenum chamber			0.452	0.455	0.465	0.460	-0.0458

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.37142 inches water

Average Static Pressure at Plane of Traverse = +0.06544 inches water

Average Static Pressure at Plenum Chamber = -0.07025 inches water

Average Total Pressure at Plenum Chamber = -0.0458 inches water

Test run No. 18 Dated 22/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.245	74.245	74.14	74.13	74.19
Temp. at Plenum Chamber deg. C.			24.0	24.3	25.0	25.1	24.6
Temp. at Traverse deg. C.			24.2	24.3	25.1	25.0	24.65
Temp. of atmosphere deg. C.			23.6	23.8	24.3	24.5	24.05
Wet bulb Temp. Deg. C.			18.7	18.9	18.9	18.7	18.8
R.P.M. of fan.			426	429	432	433	430
Fan motor	Current in Amps.		0.566	0.564	0.564	0.563	0.5643
	Voltage.		132	132	132	132	132
	Watts input.		65.8	65.2	65.2	65.6	65.45
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	4.71	3.57	4.01	3.70	3.4764
		2.2605	4.89	3.30	4.01	3.435	
		2.9164	4.61	3.62	3.875	3.205	
		3.4526	4.105	2.885	3.53	2.365	
		3.9641	3.425	2.25	2.645	2.235	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	0.74	0.79	0.69	0.675	+0.7505
		2.2605	0.745	0.805	0.71	0.66	
		2.9164	0.75	0.825	0.73	0.72	
		3.4526	0.765	0.83	0.74	0.73	
		3.9641	0.77	0.85	0.755	0.73	
Static preesure reading at plenum chamber			0.52	0.52	0.507	0.535	-0.520
Total pressure reading at plenum chamber			0.25	0.312	0.327	0.33	-0.305

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.34764 inches water

Average Static Pressure at Plane of Traverse = +0.07505 inches water

Average Static Pressure at Plenum Chamber = -0.052 inches water

Average Total Pressure at Plenum Chamber = -0.0305 inches water

Test run No. 19 Dated 26/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.23	74.23	74.225	74.22	74.226
Temp. at Plenum Chamber deg. C.			23.8	24.0	24.0	24.1	23.98
Temp. at Traverse deg. C.			24.1	24.2	24.1	24.3	24.18
Temp. of atmosphere deg. C.			22.2	22.8	22.8	22.9	22.68
Wet bulb Temp. Deg. C.			19.3	19.6	19.6	19.7	19.55
R.P.M. of fan.			418	419	424	418	419.75
Fan motor	Current in Amps.		0.555	0.551	0.550	0.547	0.5508
	Voltage.		128.3	128.1	128	128	128.1
	Watts input.		62.4	62.2	62.0	61.4	62
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	4.89	4.19	5.14	4.27	4.0761
		2.2605	4.865	3.80	5.125	3.935	
		2.9164	4.895	3.76	4.845	3.63	
		3.4526	4.23	3.435	4.735	3.62	
		3.9641	3.49	2.595	3.775	3.025	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	1.09	1.14	1.085	1.075	+ 1.123
		2.2605	1.095	1.125	1.105	1.045	
		2.9164	1.115	1,135	1.11	1.165	
		3.4526	1.13	1.155	1.115	1.18	
		3.9641	1.13	1.16	1.12	1.18	
Static preesure reading at plenum chamber			0.412	0.438	0.440	0.425	- 0.429
Total pressure reading at plenum chamber			0.18	0.182	0.203	0.200	- 0.191

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.40761 inches water

Average Static Pressure at Plane of Traverse = +0.1123 inches water

Average Static Pressure at Plenum Chamber = -0.0429 inches water

Average Total Pressure at Plenum Chamber = -0.0191 inches water

Test run No. 20 Dated 23/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.135	74.10	74.115	74.110	74.115
Temp. at Plenum Chamber deg. C.			23.6	23.8	24.0	24.6	24.0
Temp. at Traverse deg. C.			23.8	23.8	24.1	24.6	24.08
Temp. of atmosphere deg. C.			23.1	23.1	23.4	23.7	23.33
Wet bulb Temp. Deg. C.			17.1	17.2	17.7	18.0	17.5
R.P.M. of fan.			388	393	390	394	391.25
Fan motor	Current in Amps.		0.528	0.525	0.522	0.522	0.5243
	Voltage.		122	122	122	122	122
	Watts input.		56	56.2	56.2	56.0	56.1
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	4.745	4.36	4.49	4.215	
		2.2605	4.80	3.98	4.33	3.995	
		2.9164	4.745	3.78	4.14	3.41	3.7929
		3.4526	3.88	3.43	3.31	2.795	
		3.9641	3.62	2.745	2.79	2.79	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	0.975	1.00	0.975	0.96	
		2.2605	0.955	1.005	0.975	0.955	
		2.9164	1.015	1.02	0.995	1.055	1.01
		3.4526	1.03	1.025	1.005	1.07	
		3.9641	1.03	1.06	1.01	1.08	
Static preesure reading at plenum chamber			0.34	0.332	0.322	0.305	-0.325
Total pressure reading at plenum chamber			0.095	0.107	0.092	0.105	-0.100

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.37929 inches water

Average Static Pressure at Plane of Traverse = +0.101 inches water

Average Static Pressure at Plenum Chamber = -0.0325 inches water

Average Total Pressure at Plenum Chamber = -0.010 inches water

Test run No. 21 Dated 26/11/63

Set No.			1	2	3	4	Average
Barometric Pressure Cms. Hg.			74.10	74.11	74.11	74.09	74.103
Temp. at Plenum Chamber deg. C.			25.0	25.0	25.0	25.0	25.0
Temp. at Traverse deg. C.			25.1	25.0	25.2	25.2	25.13
Temp. of atmosphere deg. C.			23.8	23.8	23.8	23.9	23.83
Wet bulb Temp. Deg. C.			19.4	19.4	19.5	19.5	19.45
R.P.M. of fan.			365	365	365	365	365
Fan motor	Current in Amps.		0.485	0.485	0.484	0.487	0.4853
	Voltage.		113.6	114	114	113.9	113.88
	Watts input.		48.0	48.2	48.0	48.2	48.1
Valocity Pressure reading at plane of Traverse	Pitot tube distance from centre in inches.	1.3035	4.685	4.055	4.50	4.13	
		2.2605	4.515	3.99	4.34	3.945	
		2.9164	4.41	3.775	4.29	3.48	3.7219
		3.4526	3.70	3.155	3.69	3.315	
		3.9641	3.125	2.425	2.865	2.625	
Static pressure reading at plane of Traverse ( )	Pitot tube distance from centre in inches.	1.3035	1.25	1.23	1.155	1.185	
		2.2605	1.26	1.245	1.15	1.18	
		2.9164	1.275	1.27	1.14	1.265	1.239
		3.4526	1.275	1.28	1.205	1.265	
		3.9641	1.305	1.325	1.25	1.27	
Static preesure reading at plenum chamber			-0.245	-0.25	-0.2475	-0.245	-0.2469
Total pressure reading at plenum chamber			-0.0125	-0.005	-0.0125	-0.0125	-0.0106

Inclination of manometer = 1 : 10

Average velocity pressure at plane of Traverse = 0.37219 inches water

Average Static Pressure at Plane of Traverse = +0.1239 inches water

Average Static Pressure at Plenum Chamber = -0.02469 inches water

Average Total Pressure at Plenum Chamber = -0.00106 inches water

TEST RUN NO. 22 Dated 28/11/63

Barometric pressure	= 74.00 cms. Hg.
Temperature at plenum chamber	= 22.4°C
Temperature of atmosphere	= 22.3°C
Wet bulb temperature	= 19.9°C
Speed of fan	= 815 R.P.M.
Fan motor current	= 0.780 Amperes
Fan motor voltage	= 227.0 Volts
Fan motor watts input	= 175.4 Watts
Differential pressure reading at orifice	= 0.08
Static pressure reading at plenum	= -3.27
Total pressure reading at plenum	= -3.00
Inclination of manometer	= 1 : 10
Average differential pressure at orifice	= 0.008 inches water
Average static pressure at plenum	= -0.327 inches water
Average total pressure at plenum	= -0.300 inches water

TEST RUN NO. 23 Dated 28/11/63

Barometric pressure	= 74.02 cms. Hg.
Temperature at plenum chamber	= 22.2°C.
Temperature of atmospheric air	= 22.2°C.
Wet bulb temperature of air	= 19.9°C.
Speed of fan	= 821 R.P.M.
Fan motor current	= 0.774 Amperes
Fan motor voltage	= 227.5 Volts
Fan motor watts input	= 175.2 Watts
Differential pressure reading at orifice	= 0.245
Static pressure reading at plenum	= -2.975
Total pressure reading at plenum	= -3.02
Inclination of manometer	= 1 : 10
Average diff. pressure at orifice	= 0.0245 inches water
Average static pressure at plenum	= -0.2975 inches water
Average total pressure at plenum	= -0.302 inches water

FAN MOTOR BRAKE TEST

Dated 29/11/63

S.No.	Voltage in volts V	Current in Amperes I	Power in Watts W	R.P.M.	Belt tension in lbs Tight side T_1	Belt tension in lbs Slack side T_2	Motor Efficiency η
1	130	0.417	48	842	2.8	1.0	0.467
2	147.5	0.428	56.4	889	3.1	1.1	0.475
3	153	0.442	62	888	3.4	1.2	0.476
4	154	0.457	64	882	3.6	1.3	0.479
5	155.5	0.510	72	844	4.2	1.45	0.488
6	164.5	0.561	84	825	4.8	1.5	0.489
7	185	0.562	95	873	5.65	2.1	0.493
8	202	0.626	117	865	6.8	2.45	0.485
9	206	0.655	125	865	7.2	2.55	0.486
10	220	0.673	140	860	8.2	3.1	0.474
11	228.5	0.730	160	862	9.6	4.0	0.454
12	228.5	0.736	164	840	10.0	4.2	0.4475
13	228.5	0.746	167	839	10.4	4.5	0.445
14	228.5	0.762	169	838	10.7	4.8	0.440
15	228.5	0.764	172	833	10.9	4.9	0.436
16	228.5	0.782	174	825	10.9	4.9	0.428
17	228.5	0.786	176	820	10.9	4.9	0.421
18	230	0.784	179	828	10.8	4.9	0.413

Dia. of pulley D = 2.45"

Thickness of belt= 0.10"

$(D + t) = \frac{2.45 + 0.10}{12} = 0.2125$ ft.

$$\eta = \frac{(D + t)\, N\, (T_1 - T_2)}{W} \times \frac{746}{33000}$$

APPENDIX - B

APPENDIX - B.

CALCULATIONS OF TEST RUN NO. 4

Temperature of atmospheric air t_a = 21.93°C = 71.5°F

Wet bulb temperature t_w = 19.63°C = 67.35°F

Temperature at plane of traverse = 21.6°C = 70.85°F

Temperature at plenum chamber = 22.12°C = 71.8°F

Density of Hg at 71.5°F = 845.28 lbs/cu.ft.

Density of Hg at 32°F = 848.72 lbs/cu.ft.

Barometric pressure p_t = 74.02 cms Hg at 71.5°F of Hg.

= 29.18" Hg at 71.5°F of Hg

$$= 29.18 \times \frac{845.28}{848.72} \text{ inches of Hg at } 32^{o}\text{F of Hg.}$$

= 29.03 inches Hg at 32°F of Hg.

Saturation pressure of water vapour p_s, corresponding to W.B.T. (i.e. 67.35°F) = 0.675" Hg.

$\therefore$ Partial pressure of water vapour p_w is given by

$$p_w = p_s - \frac{p_t(t_a - t_w)}{2700}$$

$$= 0.675 - \frac{29.03(71.5 - 67.35)}{2700}$$

$$= 0.675 - \frac{29.03(4.15)}{2700}$$

$$= 0.675 - 0.0446 = 0.6304$$

DENSITY OF ATMOSPHERIC AIR

$$\rho_a = \frac{p_t - 0.38\ p_w}{0.754\ T_a}$$

where ρ_a = density of atmospheric air in lbs/cu.ft.

T_a = absolute temperature of atmospheric air deg.F. + 460.

$$\rho_a = \frac{29.03 - 0.38 \times 0.6304}{0.754\ (71.5 + 460)}$$

$$= \frac{29.03 - 0.239}{0.754 \times 531.5}$$

$$= \frac{28.791}{0.754 \times 531.5}$$

$$= 0.0718 \text{ lbs/cu. ft.}$$

DENSITY OF AIR AT PLANE OF TRAVERSE

$$\rho_d = \rho_a \frac{p_t + (p_s'/13.6)}{p_t} \times \frac{T_a}{T_d}$$

where ρ_d = density of air at plane of traverse lbs./cu.ft.

p_s' = measured static pressure at plane of traverse, inches of water.

T_d = absolute temperature of air at plane of traverse

$$\therefore \quad \rho_d = 0.0718 \times \frac{29.03 - (\frac{0.1418}{13.6})}{29.03} \times \frac{460 + 71.5}{460 + 70.85}$$

$$= 0.0718 \times \frac{29.03 - 0.0104}{29.03} \times \frac{531.5}{530.85}$$

$$= 0.0718 \times \frac{29.02}{29.03} \times \frac{531.5}{530.85}$$

$$= 0.0718$$

DENSITY OF AIR AT PLENUM CHAMBER

$$\rho_i = 0.0718 \times \frac{29.03 - \frac{0.2909}{13.6}}{29.03} \times \frac{71.5 + 460}{71.8 + 460}$$

$$= 0.0718 \times \frac{29.03 - 0.0214}{29.03} \times \frac{531.5}{531.8}$$

$$= 0.0718 \times \frac{29.0086}{29.03} \times \frac{531.5}{531.8}$$

$$= 0.0716$$

VELOCITY AT PLANE OF TRAVERSE

$$V = 1096.2 \sqrt{\frac{p'_v}{\rho_d}}$$

where p'_v = velocity pressure in inches of water

ρ_d = density of air at plane of traverse lbs./cu.ft.

$$\therefore \quad V = 1096.2 \sqrt{\frac{0.015677}{0.0718}}$$

$$= 1096.2 \sqrt{0.218}$$

$$= 511.5 \text{ F.P.M.}$$

VOLUME RATE OF FLOW AT PLANE OF TRAVERSE

Q_d = volume flow rate at plane of traverse.

= V A

where A = cross-sectional area of duct

$$= \frac{\pi}{4} \left(\frac{8.25}{12}\right)^2 = 0.3712 \text{ sq.ft.}$$

$$\therefore Q_d = 511.5 \times 0.3712 = 190 \text{ C.F.M.}$$

VOLUME RATE OF FLOW CORRECTED TO FAN INLET

$$Q_i = Q_d \times \frac{\rho_d}{\rho_i}$$

$$= 190 \times \frac{0.0718}{0.0716} = \underline{190.5 \text{ C.F.M.}}$$

DISCHARGE VELOCITY PRESSURE

$$p_v = p_v' \times \left(\frac{A}{A_o}\right)^2 \times \frac{\rho_o}{\rho_a}$$

where ρ_o = density of air at fan outlet

ρ_o = density of atmospheric air in this case.

$$\therefore p_v = 0.015677 \times \frac{0.0718}{0.0718} \times \left(\frac{8.25}{18.75}\right)^4$$

(where 18.75 = diameter of fan outlet)

$$= 0.000587 \text{ inches of water.}$$

FAN STATIC PRESSURE

Here static pressure at fan outlet is atmospheric (zero gauge). Therefore fan static pressure will be equal to the total pressure in plenum chamber only with sign reversed.

$\therefore$ Fan static pressure = 0.2810 inches water.

FAN TOTAL PRESSURE

Fan total pressure = fan static pressure + discharge velocity pressure.

= 0.2810 + 0.000587

= 0.281587 inches water.

FAN POWER INPUT

Power input to fan motor = 174.9 watts

From Fig. 27, the motor efficiency at 174.9 watts input = 0.428

$$\therefore \text{Power input to fan} = \frac{174.9 \times 0.428}{746} = 0.1005 \text{ H.P.}$$

CORRECTIONS FOR SPECIFIED INLET AIR DENSITY
(0.075 lbs/cu.ft.) AND FAN SPEED (900 R.P.M.).

Volume rate of flow corrected to specified inlet air density and fan speed

$$Q = Q_i \times \left(\frac{900}{837}\right)$$

$$= 190.5 \times \left(\frac{900}{837}\right) = \underline{204.5 \text{ C.F.M.}}$$

Fan static pressure corrected to specified inlet air density and fan speed

$$H_s = 0.281 \times \frac{0.075}{0.0716} \left(\frac{900}{837}\right)^2$$

$$= 0.3400 \text{ inches water.}$$

Fan total pressure corrected to specified inlet air density and fan speed.

$$H_t = 0.281587 \times \frac{0.075}{0.0716} \left(\frac{900}{837}\right)^2$$

$$= 0.341 \text{ inches water.}$$

Fan power input corrected to specified inlet air density and fan speed.

$$P = 0.1005 \times \frac{0.075}{0.0716} \left(\frac{900}{837}\right)^3 = \underline{0.1312 \text{ H.P.}}$$

DISCHARGE COEFFICIENT FOR ORIFICE

Volume rate of flow at test speed (837 R.P.M.)

$$= 190 \text{ C.F.M.}$$

Let C_d = coefficient of discharge of the orifice

then $190 = C_d \times \frac{\pi}{4} \left(\frac{5.25}{12}\right)^2 \times 1096.2 \sqrt{\frac{0.2425}{0.0718}}$

where 5.25 = orifice dia. in inches

0.2425 = Diff. pressure at orifice inches water

0.0718 = density of atmospheric air.

$$\therefore \quad 190 = C_d \times \frac{\pi}{4}\left(\frac{5.25}{12}\right)^2 \times 1096.2 \sqrt{\frac{0.2425}{0.0718}}$$

$$= C_d \times 0.1504 \times 1096.2 \times \sqrt{3.375}$$

$$= 303\ C_d$$

$$C_d = \frac{190}{303} = 0.627$$

(Note) The calculations for readings Nos. 1 to 3 and 5 to 21, will be similar. Only discharge coefficient for orifice will be not required.

CALCULATIONS FOR TEST RUN NO. 23

Temperature at plenum chamber = 22.2°C = 71.96°F

Temperature of atmospheric air t_a = 22.2°C = 71.96°F

Wet bulb temperature t_w = 19.9°C = 67.82°F.

Density of Hg at 71.96°F = 845.244 lbs/cu.ft.

Density of Hg at 32°F = 848.72 lbs/cu.ft.

Barometric pressure p_t = 74.02 cms. of Hg at 71.96°F of Hg

= 29.14" Hg at 71.96°F of Hg.

$= 29.14 \times \frac{845.244}{848.72}$ inches of Hg at 32°F of Hg.

= 29.01 inches of Hg at 32°F.

Saturation pressure of water vapour p_s, corresponding to wet bulb temperature (i.e. 67.82°F) = 0.6860 Hg.

Partial pressure of water vapour p_w is given by :

$$p_w = p_s - \frac{p_t(t_a - t_w)}{2700}$$

$$= 0.6860 - \frac{29.01(71.96 - 67.82)}{2700}$$

$$= 0.6860 - \frac{29.01 \times 4.14}{2700}$$

$$= 0.6860 - 0.0445$$

$$= 0.6415'' \text{ Hg.}$$

DENSITY OF ATMOSPHERIC AIR

$$\rho_a = \frac{p_t - 0.38\ p_w}{0.754\ T_a}$$

where ρ_a = density of atmospheric air in lbs/cu.ft.

T_a = absolute temperature of atmospheric air deg.F + 460

$$\therefore \rho_a = \frac{29.01 - 0.38 \times 0.6415}{0.754 \times 531.96}$$

$$= \frac{29.01 - 0.244}{398.1}$$

$$= \frac{28.766}{398.1} = 0.07205 \text{ lbs/cu.ft.}$$

DENSITY OF AIR AT PLENUM CHAMBER

$$\rho_i = \rho_a \frac{p_t + (p_s'/13.6)}{p_t} \quad \frac{T_a}{T_d}$$

$$= 0.07205 \quad \frac{29.01 + \frac{-0.2975}{13.6}}{29.01} \times \frac{531.96}{531.96}$$

$$= 0.07205 \quad \frac{29.01 - 0.02187}{29.01}$$

$$= 0.07205 \quad \frac{28.9881}{29.01} = 0.07185$$

Volume rate of flow at test speed

$$= C_d \ \frac{\pi}{4}\left(\frac{5.25}{12}\right)^2 \times 1096.2 \sqrt{\frac{0.0245}{0.07205}}$$

$$= 0.627 \times \frac{\pi}{4} \times 0.196 \times 1096.2 \sqrt{0.340}$$

$$= 0.627 \times \frac{\pi}{4} \times 0.196 \times 1096.2 \times 0.583$$

$$= 61.65 \text{ C.F.M.}$$

Velocity at fan outlet $= \dfrac{61.65}{\frac{\pi}{4}\left(\frac{18.75}{12}\right)^2}$ ft./mt.

$$V_1 = 32.20 \text{ ft./mt.}$$

Discharge velocity pressure p_v will be given by the equation.

$$V_1 = 1096.2 \sqrt{\frac{p_v}{\rho_a}}$$

$$\text{or } 32.20 = 1096.2 \sqrt{\frac{p_v}{0.07205}}$$

$$\text{or } p_v = \left(\frac{32.20}{1096.2}\right)^2 \times 0.07205$$

$$= 0.00006215 \text{ inches of water.}$$

FAN STATIC PRESSURE

Fan static pressure = 0.302

FAN TOTAL PRESSURE

Fan total pressure = 0.302 + 0.00006215

= 0.30206215

FAN POWER INPUT

Power input to fan $= \dfrac{175.2 \times 0.426}{746}$ H.P.

= 0.100 H.P.

Corrections for specified inlet air density (0.075 lbs/cu.ft.) and fan speed (900 R.P.M.)

Volume rate corrected to specified speed

$$Q = 61.65 \times \left(\frac{900}{821}\right) = 67.6 \text{ C.F.M.}$$

Fan static pressure corrected to specified inlet air density and fan speed

$$H_s = 0.302 \times \frac{0.075}{0.07185} \left(\frac{900}{821}\right)^2$$

$$= 0.3795$$

Fan total pressure corrected to specified inlet air density and fan speed

$$H_t = 0.30206215 \times \frac{0.075}{0.07185} \left(\frac{900}{821}\right)^2$$

$$= 0.3795$$

Fan power input corrected to specified inlet air density and fan speed

$$P = 0.10 \times \frac{0.075}{0.07185} \left(\frac{900}{821}\right)^3$$

$$= 0.1377 \text{ H.P.}$$

After doing these calculations for all the test runs, the curves are plotted for fan static pressure, fan total pressure and fan power input Vs. volume of air delivered by the fan. Then the efficiencies are calculated by taking points on the curves mentioned above and using the formula

$$\eta = \frac{H \times Q}{6356\ P}$$

For example, if the efficiencies are to be calculated when the volume of air delivered by the fan is 1000 C.F.M. than the procedure will be as follows.

From static pressure curve (fig. 29) the static pressure corresponding to 1000 C.F.M. is 0.212 inches water. From total pressure curve (fig. 29) the corresponding total pressure is 0.232 inches water.

From Horse Power curve (fig. 28) the H.P. input to fan at 1000 C.F.M. is 0.128.

$$\therefore \quad \text{Fan static efficiency} = \frac{1000 \times 0.212}{6356 \times 0.128} = 0.266$$

$$\text{and} \quad \text{Fan total efficiency} = \frac{1000 \times 0.232}{6356 \times 0.128} = 0.286$$

APPENDIX - C.

INDIAN STANDARDS FOR TESTING OF PROPELLER TYPE A.C. VENTILATING FANS:- (I.S.2312,1963)

The necessary information from the code concerned with this dissertation work is as given below.

AIR DELIVERY TEST

The following method of determining the air delivery of the fan shall be carried out with the guard, if any removed.

To the inlet side of the fan shall be attached an air way figure 31 of square cross-section having sides x and length $1\frac{1}{3}$ x, x being not less than 1.5 times the diameter of orifice in which the impeller rotates. To the inlet end of this airway shall be attached a mouth piece of length $\frac{2}{3}$x, having an included angle of 20 degrees converging towards the fan, and a radial flange at its inlet end of width 1/9x.

The average air velocity shall be determined by readings of a vane anemometer taken across a plane, measuring $\frac{1}{3}$x from the junction of the mouth piece with the inlet airway. The anemometer shall be placed at each of the locations defined in figure 32, being supported by wires or other means giving negligible obstruction to the air-flow across the section.

The measurement of air velocity shall be taken at each location, each reading occupying an equal interval of time not less than 30 seconds. Each reading shall then be corrected in

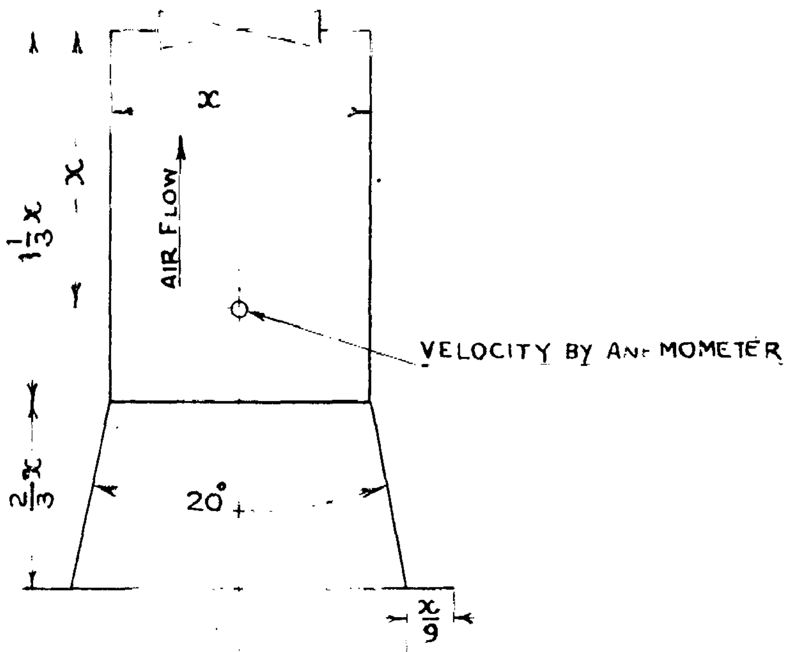

FIG. 31 AIR DELIVERY TEST

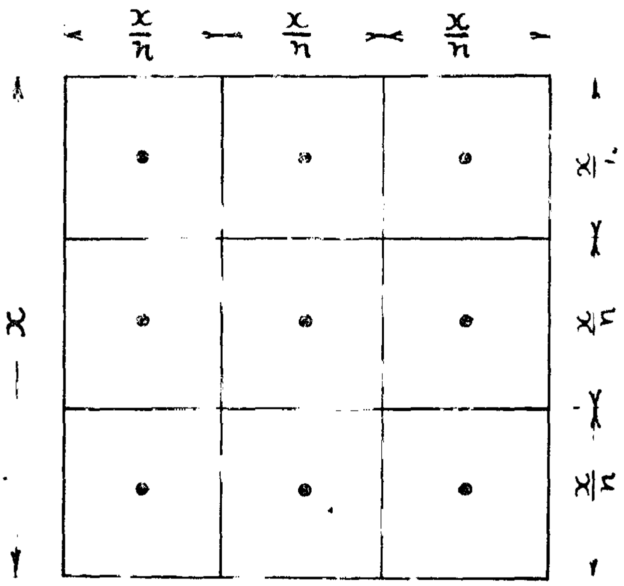

FIG. 32. DUCT TRAVERSE

accordance with any calibration correction of the instrument. The arithmatical mean of the readings so obtained shall be taken to be the average velocity of air flowing in the airway. Care shall be taken to see that the air flow through the anemometer is in the direction for which the instrument is calibrated. The average velocity so obtained multiplied by the cross-sectional area of the air-way at the plane of measurement shall be taken to be equal to the intake volume.

The cross-sectional area of the airway at the plane of measurement shall be further sub-divided into a number of equal areas (figure 32) depending upon the ratio of the length of side of test section and the diameter d of the ring shrouding the anemometer. The table below gives the number of readings to be taken and the sub-division of each side for different ratios of the length of side to the diameter d of the ring shrouding the anemometer.

Length of side of Airway	No. of readings	Sub-division of each side
Not less than 4d, less than 5d	9	3
Not less than 5d, less than 8d	16	4
Not less than 8d, less than 15d	25	5
15d or greater.	36	6

(26)

ELECTRICAL INPUT TEST

The electrical input to the fan in watts shall be determined by running the fan at the test voltage and at the highest speed. (26)

MEASUREMENT OF FAN SPEED

The speed of rotation of the fan shall be determined by running the fan at the test voltage and at its rated frequency. The method of measurement of the speed of fan shall be such that the speed of fan is not appreciably affected. (26)

TOLERANCES

The permissible tolerances will be as follows:

Air delivery	- 10 percent
Watts input	+ 10 percent
Speed	+ 10 - 5 percent

(26)

APPENDIX - D

APPENDIX - D.

STANDARDIZATION OF THE PITOT TUBE

The desirable characteristics which a standardized pitot tube should possess are (1) Known coefficient, eliminating the necessity of calibration; (2) Ease in duplication with ordinary machine shop methods; (3) Insensitivity to minor changes in form and dimensions; (4) Static holes to be as near the tip as possible; (5) Rigidity and (6) Small size. (23)

The features to be taken into consideration while designing such a pitot tube are (1) Diameters, thicknesses, and materials of the outer and inner tubes; (2) The shape of the impact tube; (3) The distance of the static holes from the tip; (4) The distance of the static holes from the stem; (5) The size, number, and arrangement of the static holes; and (6) The size of the impact opening.

The researches of Ower and Johansen, and the unpublished researches of Merriam and Spaulding at Worcester Polytechnic Institute are very helpful in choosing the pitot tube design. Although these two investigations were carried out independantly, but the results obtained are in complete agreement and justify their reliability. (23)

1. SELECTION OF DIAMETERS AND THICKNESSES OF TUBES

The selection of diameters and thicknesses of the outer and inner tubes is arbitrary. According to Lionel S. Marks a sufficient stiffness will be obtained to traverse the ducts 3 or 4 feet across by taking the outer tube of 0.04 inch thick brass (No. 18 U.S. gage), 5/16 inch outside diameter, and the inner tube preferably of 0.0285 inch thick copper (No. 21 U.S. gage), $\frac{1}{8}$ inch outside diameter. Other sizes can also be used without affecting accuracy as long as geometrical similarity is maintained. (23)

2. SHAPE OF IMPACT TUBE

The shape of the impact tube should preferably be hemispherical. A conical tip properly designed can also give equally satisfactory results, but it is more liable to be damaged, increases the total length of the tube and the static holes have to be located at a greater distance from the impact end.

3. LOCATION OF STATIC HOLES WITH RESPECT TO TIP

The presence of tip affects the flow pattern and has a tendency to decrease the static pressure at the static holes. The magnitude of this decrease in static pressure depends upon the length of straight tubing between the static holes and the tip, and upon the velocity head. (Note:- The distance between the tip and the static holes is made up of the length of tip plus the length of straight tubing. Only the length of straight tubing has effect on static pressure reading). This affect goes on decreasing as the length of the straight tubing is increased and becomes constant when this length is eight tube diameters. At this location the static-pressure reading is low by about 0.2 percent of the velocity head according to Ower; and according to Merriam and Spaulding it is still smaller. This distance of eight tube diameters for the location of static holes can be considered to be desirable. (23)

4. LOCATION OF STATIC HOLES WITH RESPECT TO STEM

The presence of the stem tends to increase the static pressure at the static holes by sending back a pressure wave in the upstream direction. According to Ower, the presence of stem increases the upstream static pressure as in the following tabulation:

Distance of stem aft of static holes, in tube diameters	5	10	15
Excess pressure due to stem in percent of velocity head	1.75	0.8	0.5

The investigations of Merriam and spaulding show a slightly greater pressure excess.

The error changes very slowly after 16 diameters length. Therefore it seems to be desirable to take the distance between the static holes and the stem axis to be 16 tube diameters.

With the stem extending across the duct (which is generally required for large ducts) the static error has twice the value of that given in above table. For a 16 diameters length it may be as much as 1 percent of the velocity head. (23)

5. NUMBER AND SIZE OF STATIC HOLES

The number of static holes as suggested by Lionel S. Marks should be eight and uniformly distributed.

Smaller size of holes (0.02 inch diameter) have the advantage of being less sensitive to Yaw, but are likely to clog.

The tube with clogged static holes becomes more sensitive to yaw. In addition to this the yaw conditions are very sensitive to unequal diameters of static holes. (Drilling of smooth and accurate holes is more difficult with smaller holes).

With holes 0.04 inches diameter, the following results were obtained by Merriam and Spaulding.

Yaw angle degrees	8.0	12.0	16.0	20.0
Static pressure error percent of velocity head	1.5	3.5	5.5	8.0
Static error including stem effect percent.	1.0	3.0	5.0	7.5

It is proposed for the purpose of achieving comparative insensitiveness to slight imperfections, that the static holes be made 0.04 inch in diameter. The inner tube should be maintained coanial,with the outer tube in the vicinity of the static holes. A spacing ring soldered to the copper tube will achieve this object. (23)

6. DIAMETER OF IMPACT OPENING

The diameter of the impact opening is immaterial when the tube is directed correctly into the air stream. Under conditions of yaw it has considerable effect on the impact reading The table below shows the effect of impact opening under yaw conditions.

Ratio of impact opening to tube diameter	0.5	0.4	0.3	0.2
Error in impact reading percent of velocity head,with 16 deg.yaw	3.3	4.5	6.0	8.5

The velocity head reading is difference of impact reading and static pressure reading. If under yaw both decrease by same amount then the velocity pressure may remain unaffected. But this condition is difficult to be achieved in practice by any design of pitot tube.

With impact opening to be 0.2 of the tube diameter and static holes 0.04 inch dia the velocity head reading varies less with yaw than when the impact openings are larger.

If it is required to find only the velocity of air or gas then impact opening of 0.2 times the tube diameter should be selected. If both velocity pressure and impact pressure readings are to be taken which is generally required in fan testing, then impact opening to be 0.4 times the tube diameter is recommended. But with effective straightener to eliminate yaw, all sizes of impact openings will give the same indications. The depth of this opening has no influence on impact reading.

Table below gives the error in velocity pressure reading for impact opening 0.4 times the tube diameter under yaw conditions (with stem effect error included).

Yaw angle deg.	8.0	12.0	16.0	20.0
Error in velocity head percent	-0.8	-1.7	-0.5	1.7

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