M.E.
Thesis case

58.

The Development and Characteristics of Fans

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I. Uses of Fans

Fans and blowers are used for many purposes the most important of which are as follows: (1) Power Plant Work - The fans here are used for furnishing the forced draught thru the fuel bed and to furnish the induced draft in case where economizers are used or where conditions are such that to furnish the required draft too tall a stack would be necessary. If a stack has been designed for a definite boiler rating any increased rating of the boilers will require additional draft. There is no practicable means of increasing natural draft as such after the maximum has been reached. Again chimney draft is susceptible to a large extent to natural atmospheric conditions, winds and air currents. So that for many power plant installations artificial draft is a necessity. It is very flexible and can be readily adjusted to meet various rates of combustion, it does not depend on climatic and atmospheric conditions and permits any degree of overload without decreasing the effeciency unduly. (2) Heat, Ventilating, and Air Conditioning Processes- In many large buildings, factories, auditoriums etc. it is necessary to keep contents of the rooms withim certain limits. This is accomplished by sending air thru the buildings by means of fans. This air may be sent over heating stacks and in this way heat the buildings. The value of a proper ventilating system can be seen from the following report of Prof. S.H. Woodbridge of Boston which says: "death rates have been reduced by the introduction of effecient ventilating systems in children's hospitals from 50% to 5%." To accomplish ventilation by forced circulation it is necessary to either exhaust air from the building or force air into it or by a combination of the two. In any case the fan is used as being the source of the most efficient aeromotive force. (3) General Engineering Processes-- Included under this there are fans and blowers used for sending air to blast furnaces, fans for properly ventilating mines and many other minor uses.

II. Testing of Fans

In the test of a fan the quanities to be determined and the general characteristics sought should depend upon the uses to which the fan is to be put. If the operating conditions of the fan are such that its speed is to remain sensibly constant this should be the independent variable. Or the speed of the fan may be varied and the pressure and velocity of the air changed. The air velocity and pressure may be varied for a given speed only by changing the size of the air discharge pipe. A useful set of tests may be had by keeping the speed of the fan practically constant and varying the pressure. Or the fan may be operated at a variable speed and the pressure kept constant by adjusting the external resistence to the air. This may be done by using different nozzles or orifices as outlets from the discharge pipe. A convenient arrangement for doing this is to have an arrangement at the end of a pipe whereby the opening may be changed. This can be done by making a holder at the end of the pipe and making arrangements for inserting plates with different sized openings into this holder. The duration of a rum should be about ten minutes and several sets of readings made during the time and their average used, as the readings for the run. The following measurements are usually made.

(1) Measurement of Air Volume

By this is meant the amount of air in cubic feet per minute flowing thru the discharge duct or pipe, and it may be measured with a good degree of accuracy. There are two general methods in use for obtaining this quantity, the pitot tube and the anemometer methods. The anemometer consists of a light, delicately constructed fan wheel whose motion is transmitted to a system of practically frictionless gearing within an attached case. The movement of the fanwheel is registered on a dial by hands revolving upon graduated circles and the velocity of the air in feet per minute is thus indicated. Such an instrument if well calibrated will give good results. In using the anemometer the fan ~~which~~ wheel is held perpendicular to the direction of the flow of the air and the instrument is moved slowly. The time is noted and the readings of the dials divided by the time in minutes will give the velocity of the air. The velocity thus obtained corrected for any known error of the instrument multiplied by the area of the passage will give the volume of air passing. It is necessary to correct all readings by means of a factor determined for the particular instrument used. As no two anemometers are alike this correction will be different for each instrument and it therefore is a necessity. The anemometer should not be used where great accuracy is required.

A second method of measuring the quantity of air is by use of the Taylor Pitot Tube. This consists of two tubes concentrically placed into the pipe or duct in which the air is moving and are turned parellel to the direction of flow so that the moving air impinges on the end of the tube. The two tubes are connected to U tubes containing a column of water. The inner tube having the open end receives the full force of the air and registers both pressure and velocity head or the dynamic head so called. The end of the outer tube is closed and slits are provided on the side of the tube so that only the static head is registered. If now the inner tube is connected to one end of a differential gauge while the other is connected to the other arm the head registered by the gauge will be the difference between the total or dynamic head and the static head; and is therefore the velocity head. Having obtained the velocity head the following relation holds:

$$V = \sqrt{2gH}$$

V is the velocity in feet per second.

G is the acceleration due to gravity or approximately 32.2.

H is the velocity head in feet of air.

But $PV = WRT$

$$\therefore w = \frac{PV}{RT} = \frac{144P}{53.4T} = \frac{2.7P}{T} \ \#/\text{cu. foot}$$

If the value of H as read by the gauge is in inches of water the following transformation is made:

$$H = \frac{62.3}{2.7p} \times \frac{h}{12} \times T$$

$$\therefore V = \sqrt{\left(\frac{2 \times 32.2 \times 62.3}{2.7 \times 12}\right) \frac{hT}{p}}$$

$$= 11.2 \sqrt{\frac{hT}{p}}$$

V is the velocity of air in feet per second.

H is the velocity head as read by the differential gauge in inches of water.

T is the absolute room temperature and is equal to the room temperature in degrees Fahrenheit plus 459.5.

P is the absolute pressure of the air in the pipe in pounds per square inch.

Knowing the velocity of the air the pipe we multiply it by the cross sectional area of the pipe in the proper units and obtain the quantity of air flowing thru the pipe per minute. Various differential gauges are used for measuring the velocity head. The "Ounce Gauge" made by the Sturtevant Company will measure large differences in pressure up to twenty ounces of water pressure The Ellison differential draft gauge is also used. For the determination of very small differences in pressure a device known as a hook gauge is used. This is a form of U tube but the level of the water is accurately determined by the aid of a hook gauge. With this instrument the readings of differences of one one-hundreth of an inch of water is a simple matter.

In using the Taylor pitot tube for accurate work the cross section of the pipe or duct should be divided up into equal concentric areas and readings for velocity obtained for each area and then the mean of these readings obtained should be used for determining the air velocity. If the work does not have to be very accurate the tube may be held in the center of the air duct and a single reading made. Experiments have shown that a good value for the pipe factor is .828 i.e. if a reading is made at the center of a circular pipe, this reading multiplied by .828 is a good value to use for the mean velocity in the pipe. If this coefficient is used and the air is assumed at standard conditions that is 14.7 lbs. per square inch and 70° F the expression for the velocity is:

$$V = 4000\sqrt{h}$$

V is the velocity in feet per minute.

H is the velocity head in inches of water.

The above is a good expression to use for all practical work.

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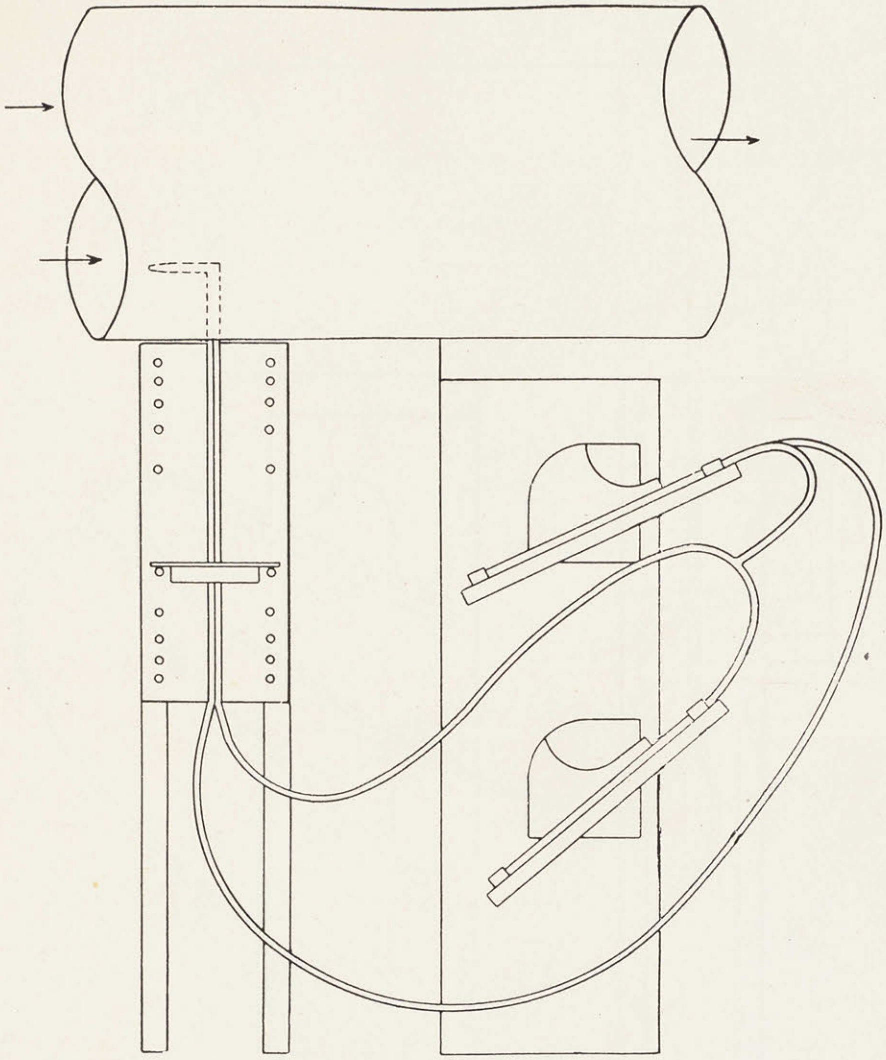

Fig. 11—Arrangement of Connections from Pitot Tubes to Manometers for "Blowing" Test.

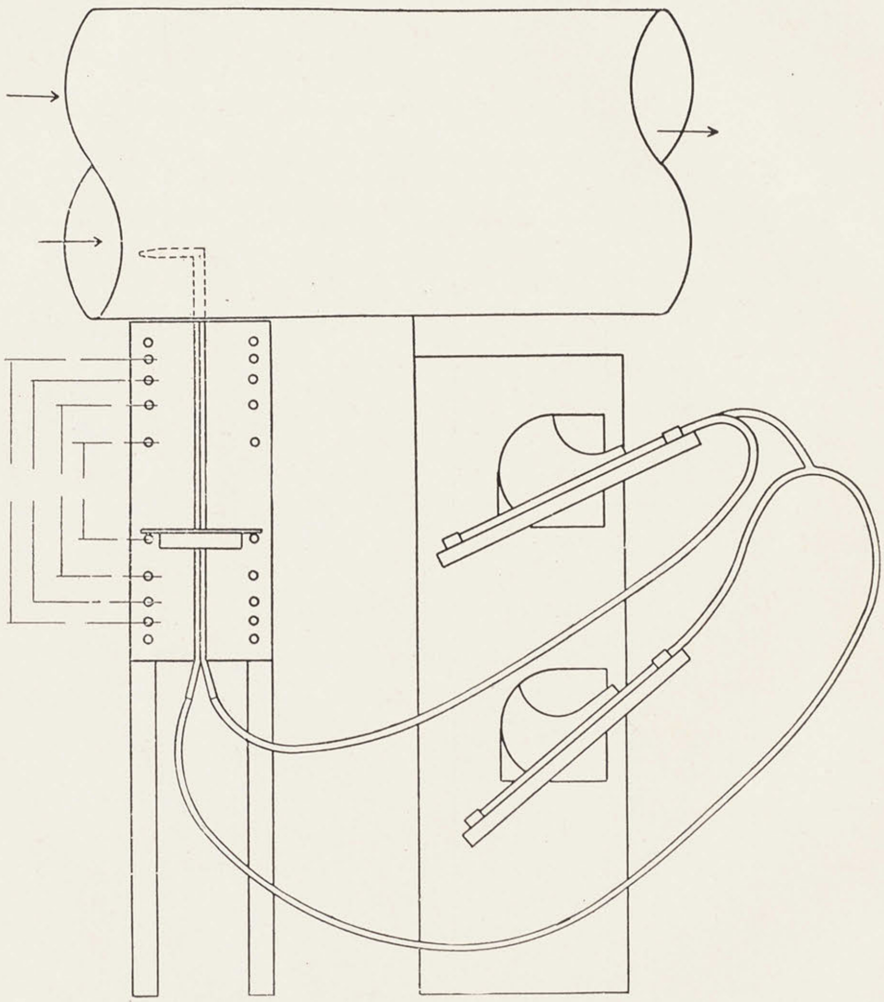

Fig. 12—Arrangement of Connections from Pitot Tube to Manometers for "Exhausting" Test.

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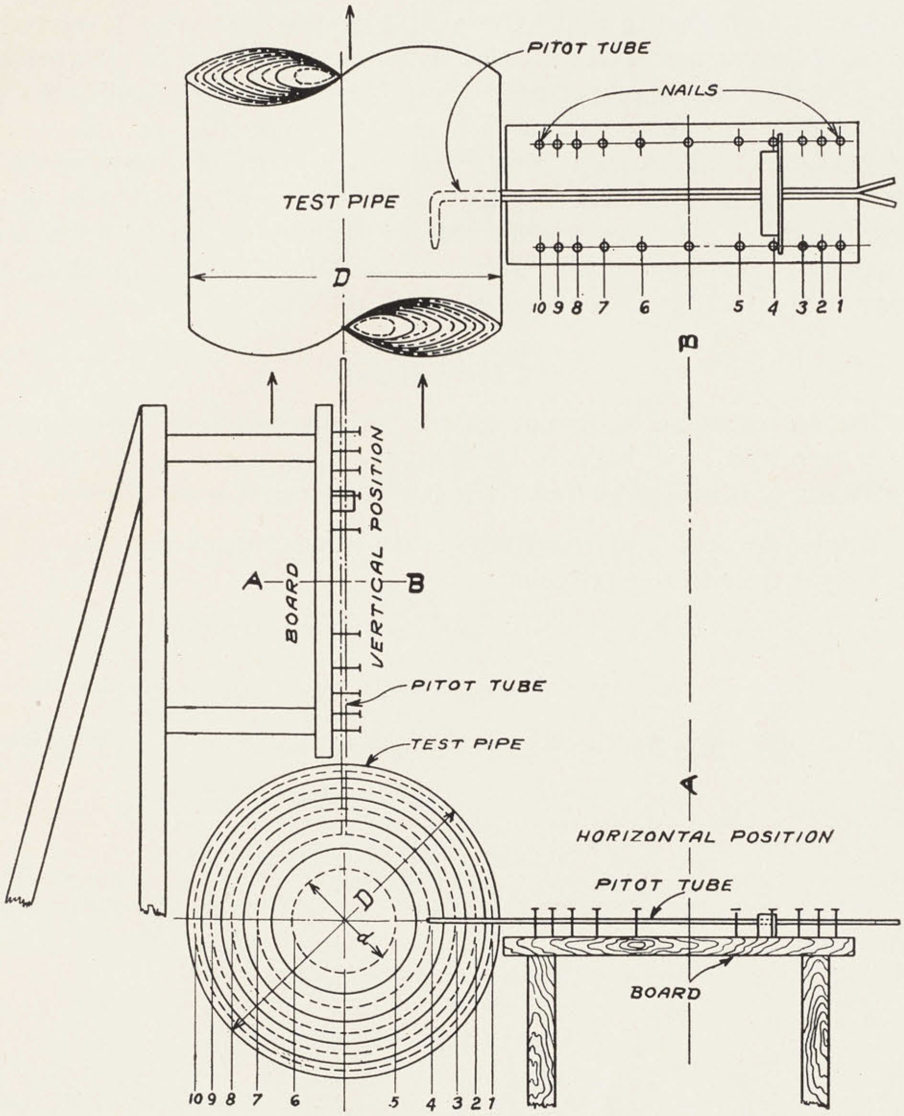


Fig. 13—Method of Adjusting Pitot Tube.

velocity and high static pressure results. If the end of the test pipe is partially closed, the effect, as regards the resistance against which the fan must operate, is the same as attaching the fan to a system of ducts. In blowing tests the outlet of the test pipe is restricted and for exhausting tests the inlet is restricted. Thin flat plates with round orifices are used for restricting the area of the test pipes and it is usually ample to test a fan with the pipe full $\frac{3}{4}$, $\frac{5}{8}$, $\frac{1}{2}$, $\frac{3}{8}$, and $\frac{1}{4}$ open and entirely closed off. The area of the opening divided by the area of the test pipe is the ratio of opening or equivalent orifice. The power required for driving a fan in determined by means of one of the usual forms of dynamometers or by the power input of an electric motor which drives the fan.

Calculations

The relations between fan speed, pressure, volume delivered and power required have been fully verified by tests and under the same conditions of air density and with the same fan are as follows:

a. The volume of air discharged by a fan varies directly as the number of revolutions, that is:

$$\frac{\textit{velocity}}{\text{Velocity}} = \frac{\textit{volume}}{\text{Volume}} = \frac{\textit{revolutions}}{\text{Revolutions}}$$

b. The pressures produced vary as the square of the revolutions,

$$\frac{\textit{static}}{\text{Static}} = \frac{\textit{total}}{\text{Total}} = \frac{\textit{revolutions}^2}{\text{Revolutions}^2}$$

c. The power required to drive the fan varies as the cube of the revolutions,

$$\frac{\textit{power}}{\text{Power}} = \frac{\textit{revolutions}^3}{\text{Revolutions}^3}$$

In making fan tests the speed may not be constant at all times for a particular test and the above relations are used for making corrections for a common speed.

It is important to have a standard weight of air at which fan performances may be compared. The density, while ordinarily a small factor, affects the results considerably if extremes are great, as with heated or cold air and air at high altitudes.

(2) Measurement of Horse Power Input

The determination of the horse power supplied to the fan will depend of course on the method by which the fan is driven. If the fan is motor driven the input to the motor must be determined and then subtracting from this input the motor losses the input to fan can be determined. Usually the unit as a whole is tested and then the input to the unit is the input to the motor. If the fan is steam driven the energy removed from the steam multiplied by the mechanical effeciency of the engine at that load is the input to the fan, and as before if the unit as a whole is tested the input to the unit is the input to the steam engine or turbine. The same method is used when the fan is driven by a gas engine. If the fan is belt driven a transmission dynamometer can be used to advantage. Allowance should be made for belt losses as the power supplied to the fan shaft is desired. This may be done by using the revolutions per minute of the fan shaft with the torque shown by the dynamometer to calculate horse power and multiplying the result by the ratio of diameters of fan pully to dynamometer pully. This will allow for the belt slip. A useful quantity used in the testing of a fan is the gross air horse power.

$$\text{Gross Air Horse Power} = \frac{W\ (H)}{33000}$$

W is the pounds of air delivered per minute.

H is the total head as determined by the gauge in feet of air.

The speed of the fan is noted by a counter or by means of a Hassler Tachometer. The mechanical effeciency of a fan is the air horse divided by the power supplied. Another useful quantity by which fans can be judged is the horse power supplied per thousand cubic feet of free air per minute. This expression is analogous to the duty of a pump. An expression frequently mentioned in the performance of fans is the manometric effeciency, this effeciency being the ratio of the dynamic head as actually observed to the maximum theoratical dynamic head. The theoratical maximum dynamic head being the dynamic head of the air at the tips of the blades.

The volumetric effeciency of a fan is the ratio of the actual volume of air passing in a given time divided by the impeller displacement for the same period. The expression for this is as follows:

$$E_{\text{volumetric}} = \frac{\text{Volume discharged in cubic feet per minute}}{\text{Impeller displacement in cubic feet per min.}}$$

$$= \frac{Q}{\pi D^2 N B}$$

Q is the volume discharged in cubic feet per minute.

D is the diameter of the impeller in feet.

N is the revolutions per minute of the wheel.

B is width of the impeller in feet.

General Relations between the quantities is rather interesting. The horse power required to operate a fan will vary directly as the work the fan does and inversly as the effeciency of the unit that is:

$$\text{Horse Power to drive fan} = K \frac{Q \times P_a}{E}$$

Where K is a constant

Q is the volume of air handled.

P_a is the dynamic pressure.

E is the total effeciency of the unit.

But the velocity varies as the square root of the head that is:

$$V = K\sqrt{h}$$

and $Q = K'AV$

therefore $Q = K''\sqrt{h} = K'''\sqrt{p}$

where P is the dynamic pressure of the air.

Therefore substituting in the expression for horse power for the value of Q we obtain:

$$\text{Horse Power to drive fan} = K\,p^{\frac{3}{2}}$$

This shows that the horse power required to drive a fan varies as the three halves power of the dynamic pressure of the air. Since the capacity of a fan is directly proportional to the peripheral velocity or fan speed and the pressure developed varies directly as the square of the speed it follows that the horse power required to drive a fan varies as the cube of the speed that is:

$$Q \doteq v$$

$$v \doteq \sqrt{p}$$

$$\therefore p \doteq v^2 \doteq N^2$$

$$\text{but} \quad H.P. \doteq p^{3/2} \doteq (N^2)^{3/2} \doteq N^3$$

$$\text{or} \quad H.P. \doteq MN^3$$

H.P. is the horse power required to drive the fan.

M is a constant for a given fan.

N is the speed of the fan in revolutions per minute.

Hence it is important to note that even for a moderate increase in speed the horsepower to drive goes up very fast. Hence in selecting a fan either the right size should be chosen or a larger size. It is as a rule more economical to err in selecting too large a fan than one which must be forced above its rated speed. Also as the capacity varies directly with the speed the horse power will vary directly as the capacity and consequently the remarks in regards to speed regulation apply here also.

Several approximate rules which are useful in power plant work in connection with fans are as follows:

Rule (1) the cubic feet of air to be supplied per minute by a forced draft apparatus is equal to four times the number of pounds of coal burned per hour.

Rule (2) the cubic feet of gases handled per minute by the induced draft apparatus when no economizer is used is equal to eight times the number of pounds of coal burned per hour.

The above are practical rules determined as a result of tests by A.A. Potter, dean of Engineering at the Kansas State Agricultural College and S.L. Simmering, an instructor at the same institution.

The Influence of Air Temperature on the Performance of Fans

In quoting the output or capacity of a fan or blower the temperature of the air handled should be noted as the volume per unit weight of the air changes with the temperature. A fan chosen to handle air at say 60° F will not give the same performance when handling flue gas of a temperature of 500 or more degrees Fahrenheit.

The expression for velocity of air previously derived shows that the velocity is proportional is to the square root of the dynamic pressure and inversly proportional to the square root of the density that is:

$$v = k\sqrt{\frac{p}{d}}$$

Hence it is evident that the greater the density the less the velocity and the less the density the greater the velocity. The density of air varies directly with the temperature, the higher the temperature the less the density. Hence for the same value of dynamic pressure air at a high temperature will have a greater velocity of flow than air at a lower temperature.

Relation Between Actual Pressure to Maintain a Given Velocity of Air and the Theoratical

Value

If there were no friction or pipe constructions the quantity of air flowing thru a given section of pipe would be determined by multiplying the velocity by the area of the section. Usually however there is a friction loss depending on the shape of the section thru which the air is flowing, and the expression for the quantity has to be multiplied by a coefficient depending on the shape of the section. What value to use for C depends upon the judgement and experience of the engineer. Approximate values for the coefficient are as follows:

square tube	$C = .82$
round tube	$C = .85$
rectangular tube	$C = .80$

J.H. Kinealy on "Centrifugal Fans"

Blast Areas of Fans

Up till now in the discussion of fans nothing has been mentioned as to the size or area of the outlet orifice although it has been assumed that it was large enough to allow the air to pass out. If the outlet orifice is small the air will pass out of it with the velocity of the tips of the blades. If the opening is increased this same condition will hold until the opening is of such size beyond which the velocity of the air is less than that of the tips of the blades. Hence the blast area of a fan may be defined as that theoratical area of outlet which will allow the maximum quantity of air to pass out while the pressure in the fan housing remains equal to that corresponding to the velocity of the tips of the floats. By theoratical is meant one whose coefficient as presiously discussed is one. Hence if the R.P.M. of a fan is known and its capacity the blast area can be calculated from the following:

$$Q = AV = \pi DNA$$

$$\therefore \quad A = \frac{Q}{\pi DN}$$

In ordinary heating and ventilating work the blast area is a matter of small importance but in exhauster work for mills and factories where it is necessary to choose a fan to carry away shavings, lint, etc. and where the velocity of the air in the ducts must be quite large, the blast area is of importance and determines the size of fan to use. In many fan catalogues the blast areas of fans are given. If in practice the discharge opening of the fan is to be less than the blast area the quantity of air which will be handled can be determined as follows:

$$Q_1 = \pi D N c a$$

Where Q_1 is the discharge

D is the diameter of the fan

N is the revolutions per minute of the fan

A is the outlet area

C is a constant depending upon the size, shape and length of the outlet.

If the discharge area is greater than the blast area the quantity of air which can be handled cannot be definitely known and experiments are the only means for determining the value.

Typical Test on a Fan as performed in M.E. laboratory of M.I.T.

Object: To determine the working characteristics of a Fan.

Apparatus:

A D.C. Shunt Motor was used to drive a 24" Fan, Sirocco Type, supplying air to a 19 inch pipe. The back pressure in the pipe was varied by changing the size of the outlet at the end of the pipe. The dynamic pressure and velocity pressure were measured by means of a Pitot inserted as shown in figure on the following page. Due to friction near the surface of the pipe the velocity there will be less than at the center. To obtain the average velocity pressure it is necessary to read the Pitot tube for several sections as shown in figure on next page. The cross section of the pipe is divided into five concentric rings, and readings are taken at the intersection of the mean circumference of the rings and the diameter of pipe, and also at the center of the pipe, making eleven readings in all, the average of which is assumed to be the true value for the velocity. The Pitot Tube does not read velocity directly but reads the velocity head. To get the velocity the formula :

$$V = \sqrt{2gH}$$

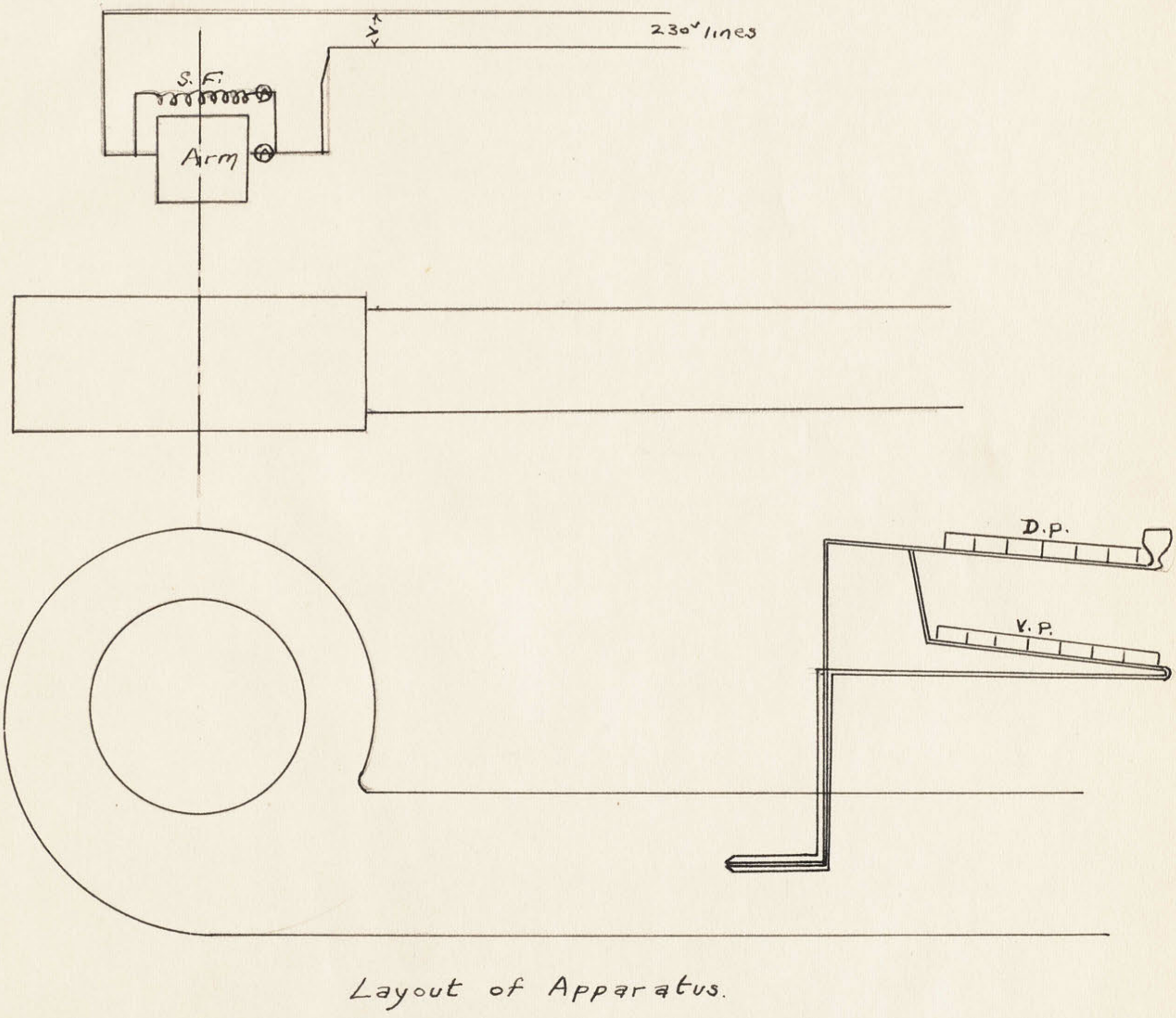
H is the velocity head in feet of the fluid.
This formula is not convenient, and one is desired in which H can be expressed in inches of water.

By dividing the average velocity by the velocity at the center a value known as the pipe factor may be obtained. If conditions are then allowed to remain unchanged it will not be necessay to take eleven readings of the Pitot Tube for each run, but merely the velocity at the center to obtain the average value in the pipe.

The output is calculated by multiplying the average velocity in feet per second by the area of the pipe. The input to the fan must be measured indirectly from the output of the motor, which will be given by:

Input — Losses

$$EI - R_a I_a^2 - V I_f - \text{Stray Power}$$



Layout of Apparatus.

Data:

Barometer	(30.17")	14.8# Sq. in.
Room Temperature		73° F
Diameter of Pipe		A = 1.97 Sq. ~~in~~. FT.
Field Current of Motor		0.80 amp.
Armature Resistance		0.48 ohm.

Stray Power

700	r.p.m.	413 watts
710	-------------	422
720	-------------	431
730	-------------	441
740	-------------	451
750	-------------	461

Location of Pitot Tube from pipe wall to give mean zone velocity.

Zone 1	0.0256 D
2	0.0817 D
3	0.1460 D
4	0.2260 D
5	0.3420 D
Center	0.5000 D

Size of opening

	Zone	DP	VP	SP	VP	Volts	Amps	RPM
	1	1.23	.98		.99			
	2	1.32	1.09		1.045			
100%	3	1.35	1.09		1.045			
A= 1.97	4	1.32	1.06		1.03			
sq. ft.	5	1.31	1.08		1.04			
	Center	1.23	1.07		1.035			
	6	1.24	1.08		1.04			
	7	1.24	1.08		1.04			
	8	1.26	1.05		1.025			
	9	1.21	.98		.99			
	10	1.01	.72		.848			
100% Average		1.24	.99	.25	1.012	230	22.55	720
75% A= 1.40								
sq. ft. avg.		1.64	.45	1.19	.69	228	16.00	730
50% A=1.08		1.77	.26	1.51	.54	230	13.50	740
25% A= .54		1.92	.093	1.83	.31	230	10.75	746
0 A= 0		2.29	0	2.29	0	230	10.25	754

Results:

	Percent Opening				
	100%	75%	50%	25%	0
Density of Air in Room	075				
" " " " Pipe	075				
Ave. Velocity Air Pipe	67.6	46.1	36.1	20.7	0
Velocity at each zone 1	66.2				
2	69.9				
3	69.9				
4	68.8				
5	69.5				
6	69.5				
7	69.5				
8	68.5				
9	66.2				
10	56.6				
c	69.2				
Velocity at Center (ft.)(sec.)	69.2				
Pipe Factor	.978				
Velocity at Tips of Fan Blades ft./sec.	75.5				
Capacity (cu. ft.)(min)	7992	5448	4266	2448	0
Input to Fan(Horsepower)	5.80	3.90	3.20	2.40	1.83
Air Horsepower	1.56	1.41	1.19	.74	0
Mechanical Efficiency%	26.9	36.0	37.1	30.9	0

Calculations:

(1) Density of Air in Room

$$PV = R.T.$$

$$V = \frac{53.35\ (533)}{14.8 \times 144} = 13.32 \text{ cu ft. per lb.}$$

$$\text{density} = \frac{1}{13.32} = .075$$

(2) Density of Air in Pipe

$$V = \frac{53.35\ (533}{144 \times S.P.} \qquad S.P. = \frac{h_{sp}}{12}\ \frac{62.5}{144} = h_{sp}\ (0362)\ \#/sq."$$

100%

$$= 25\ (0362) = .0091$$

$$= \frac{53.35\ (533)}{144 \times 14.83} = 13.32 \qquad \text{density} = .075$$

71%

$$V = \frac{53.35\ (533)}{144\ (14.86)} \qquad S.P. = 1.19\ (0362) = \frac{.043}{14.82}$$

$$V = 13.28 \qquad \text{density} = .0753$$

54.8%

$$V = \frac{53.35\ (533)}{144\ (14.87)} \qquad S.P. = 1.51.(0362) = .0546$$

$$V = 13.28 \qquad \text{density} = .o753$$

27.4%

$$V = \frac{53.35\ (533)}{144\ (14.89)} \qquad S.P. = 1.86\ (0362) = .6672$$

$$V = 13.25 \qquad \text{density} = .0754$$

0%

$$V = \frac{53.35\ (533)}{144\ (14.9)} \qquad S.P. = 2.29\ (0362) = .083$$

$$V = 13.24 \qquad \text{density} = .0755$$

Consider density equals .075 for all runs, error very slight.

Calculations:

(3) Average velocity of air in pipe

$$100\% = 66.8 \times 1.012 = 67.6 \text{ ft./sec.}$$

$$71\% = 66.8 \times .69 = 46.1$$

$$55\% = 66.8 \times .54 = 36.1$$

$$7\% = 66.8 \times .31 = 20.7$$

$$. = 0$$

(4) Velocity in each zone (for 100% opening only.)

$$1 = 66.8 \times .99 = 66.2 \text{ 1/sec.}$$

$$2 = 66.8 \times 1.045 = 69.9$$

$$3 = 66.8 \times 1.045 = 69.9$$

$$4 = 66.8 \times 1.03 = 68.8$$

$$5 = 66.8 \times 1.04 = 69.5$$

$$6 = 66.8 \times 1.04 = 69.5$$

$$7 = 66.8 \times 1.04 = 69.5$$

$$8 = 66.8 \times 1.025 = 68.5$$

$$9 = 66.8 \times .99 = 66.2$$

$$10 = 66.8 \times .848 = 56.6$$

(5) Velocity at Center (for 100% opening only.)

$$c = 66.8 \times 1.035 = 69.2$$

(6) Pipe factor (for 100% opening only.)

$$\text{P.F.} = \frac{\text{vel. ave.}}{\text{vel. center}} = \frac{67.6}{69.2} = .978$$

Calculations:

(7) Velocity of tips of fan blades (100% run)

24" diameter

$$\frac{\pi DN}{60} \text{ ft./sec.} = \frac{\pi \; 2 \; 720}{60} = 75.5 \text{ ft/sec.}$$

(8) Capacity of Fan

$Q = A$

100% run	1.97 x 67.6 =	133.2 cu. ft/sec.	=	7992 cu ft. min
75% "	1.97 x 46.1 =	90.8	=	5448
50% "	1.97 x 36.1 =	71.1	=	4266
25% "	1.97 x 20.7 =	40.8	=	2488
0% "	1.97 x 0 =	0	=	---

(9) Horsepower Input

$$HP = \frac{E\,I - Ra\,Ia^2 - V\,I_f - \text{Stray Power}}{746}$$

100% run $$HP. = \frac{(230 \times 22.5) - .48(21.75)^2 - 230 \times 8 - 431}{746}$$

$$= 5.80$$

75% run $$HP. = \frac{(228 \times 16) - .48\,(15.2)^2 - 228(.8) - 441}{746}$$

$$= 3.90$$

50% run $$HP. = \frac{230\,(13.5) - .48\,(12.7)^2 - 230\,(.8) - 451}{746}$$

$$= 3.20$$

25% run $$HP. = \frac{(230 \times 10.75) - .48\,(10)^2 - (230 \times .8) - 457}{746} = 2.40$$

0 $\quad HP. = \dfrac{(230 \times 10.25) - .48\,(9.5)^2 - (230 \times 8) - 465}{746} = 1.83$

(11) Air Horsepower

$$HP. = \frac{Q \times 5.2 \times \text{Dyn. Press.}}{550}$$

100% $\quad HP. = \dfrac{133.2 \times 5.2 \times 1.24}{550} = 1.56$

75% $\quad = \dfrac{90.8 \times 5.2 \times 1.64}{550} = 1.41$

50% $\quad = \dfrac{71.1 \times 5.2 \times 1.77}{550} = 1.19$

25% $\quad = \dfrac{40.8 \times 5.2 \times 1.92}{550} = .741$

0 $\quad$ 0

(12) Mechanical Efficiency $= \dfrac{\text{output}}{\text{Input}}$

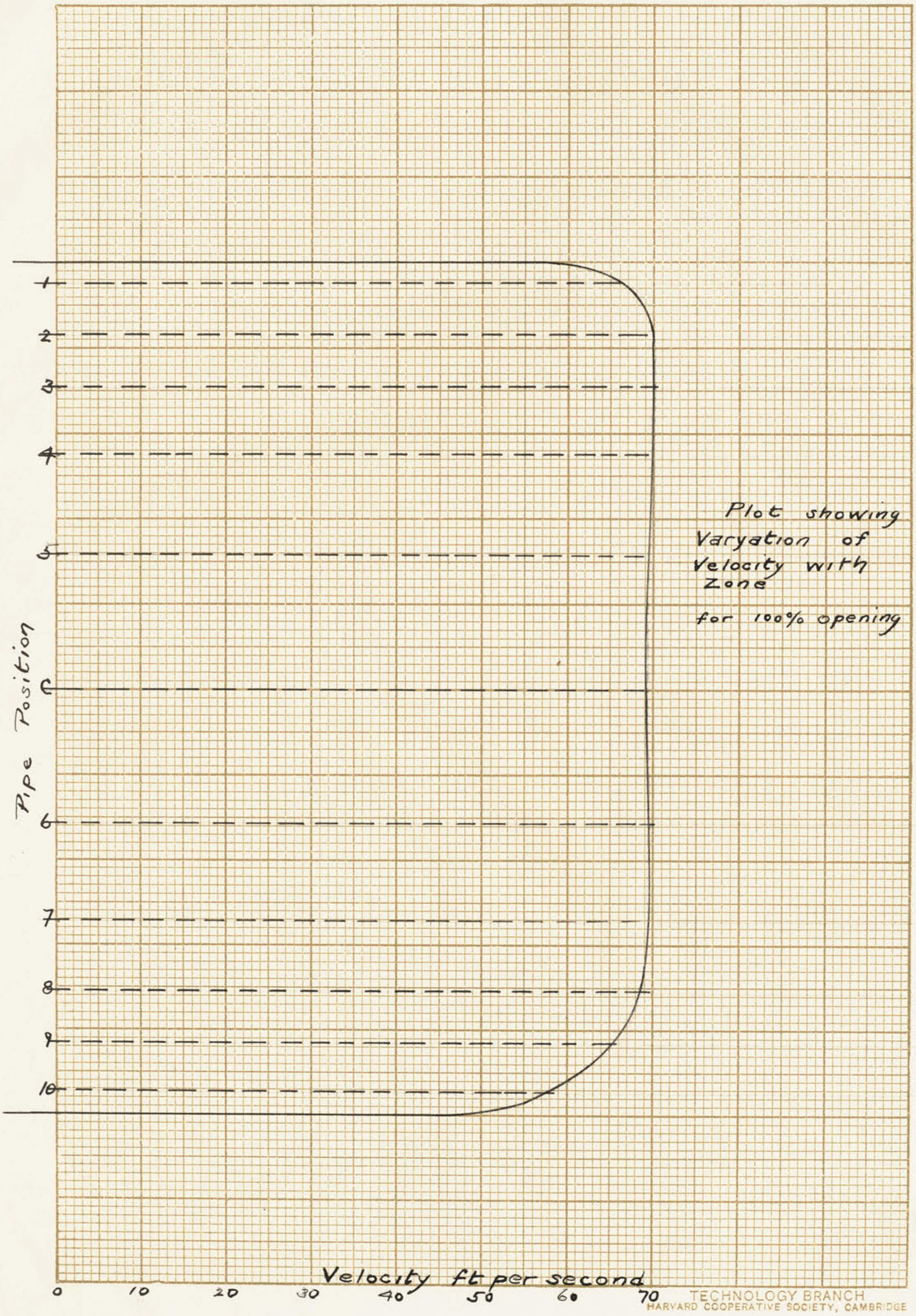
100% $\quad \dfrac{1.56}{5.80} = 26.9\%$

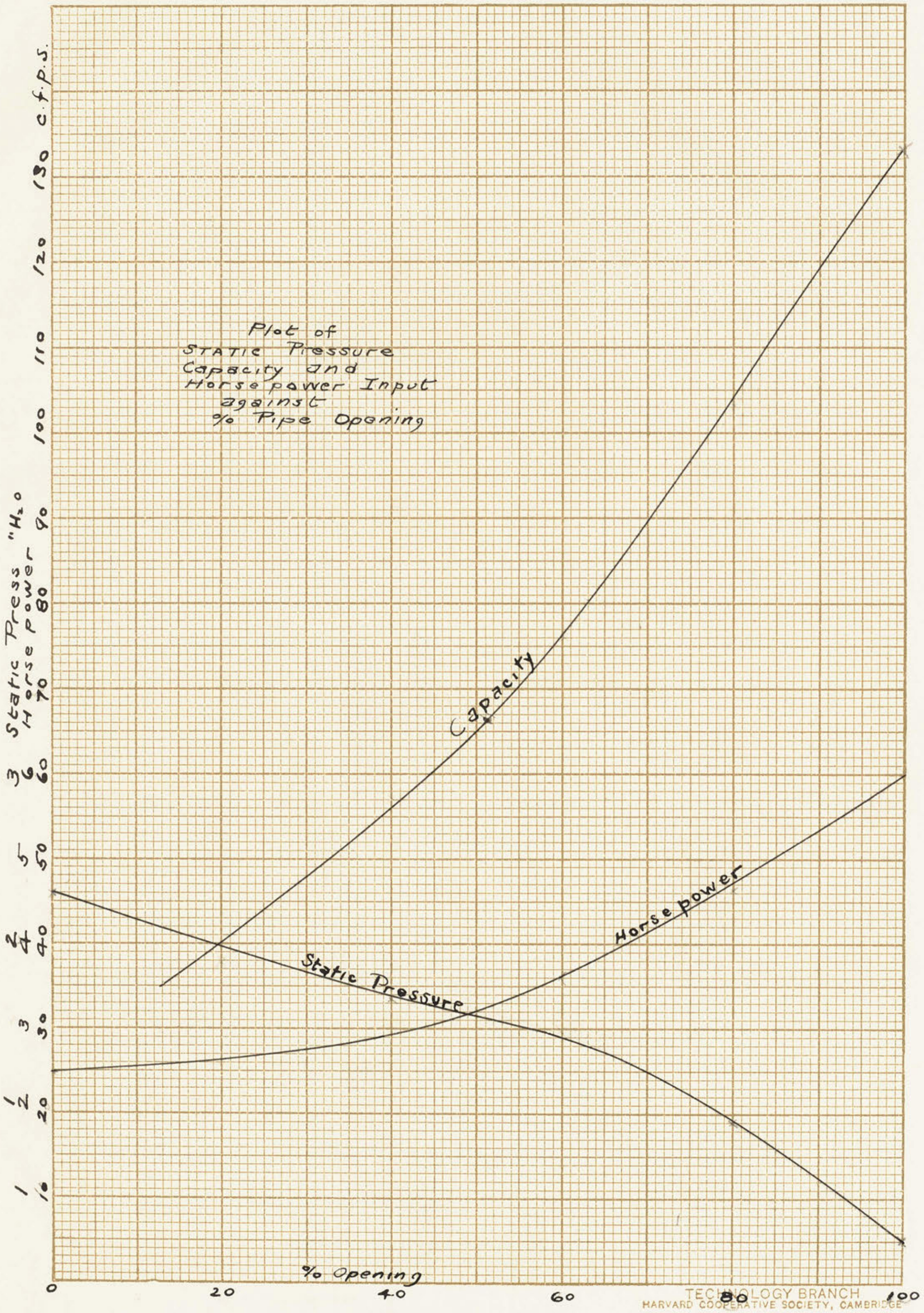
75% $\quad \dfrac{1.41}{3.90} = 36.0\%$

50% $\quad \dfrac{11.9}{320} = 37.1\%$

25% $\quad \dfrac{.741}{2.400} = 30.9\%$

0 $\quad$ 0


Plot showing
Varyation of
Velocity with
Zone
for 100% opening
Pipe Position
1
2
3
4
5
C
6
7
8
9
10
Velocity ft per second
0
10
20
30
40
50
60
70
TECHNOLOGY BRANCH
HARVARD COOPERATIVE SOCIETY, CAMBRIDGE


Plot of
STATIC Pressure
Capacity and
Horsepower Input
against
% Pipe Opening
Capacity
Horsepower
Static Pressure
% Opening
Static Press "H_2O
Horse power
c.f.p.s.
0
20
40
60
80
100
TECHNOLOGY BRANCH
HARVARD COOPERATIVE SOCIETY, CAMBRIDGE

Use of Curves to show Fan Characteristics

After a test has been made on a Fan the performance and characteristics of a Fan can best be judged by plotting the values obtained. Various methods are used for this purpose. One is to plot the volume in cubic feet per minute as abscissae and the velocity pressure, static pressure, horsepower input, and mechanical effeciency as ordinates. The ratio of the static to the dynamic pressure is also frequently plotted. For a given type of Fan, if a Fan is tested handling air at a standard density, and the diameter of the Fan wheel is one foot comparisons can easily be made. The Fan should be run at such a speed as to produce a pressure of one inch of water due to the velocity of the blades, this being at 1274 revolutions per minute. Then from the curves the performance at any other speed can be computed by noting that the volume varies as the first power of the speed, the pressures as the second power, and horsepower to drive as the third power. The effeciency remaining constant for any given load points of the curve. It should also be noted that the capacity of a given design of Fan operating against a given static pressure at a given speed varies as the square of the diameter. From the above simple rules Fans of symmetrical design can be judged from a curve such as mentioned. Frequently the ratio of effect or full opening is plotted as an ordinate. This gives a means of judging the characteristics of a fan against various sized openings. If this data is plotted and the use to which a fan is to be put is known the effective opening can be closely arrived at and the operation of the Fan noted. Just what the effective opening of a Fan will be in operation (actual) cannot be definitely determined but the judgement and experience of the engineer will lead him to choose a proper value.

(III) Drives for Fans

Three general methods for driving Fans are used, by motor, steam engine and steam turbine. Which method should be used is governed by the existing conditions. Motors are quite dependable when a reliable source of power can be secured either from the generators of the power plant or from some public service company nearby.

Whenever it is desired to have the fan independent of external sources of supply steam should be used, either a steam engine or steam turbine. When starting the plant steam may be generated in one of the boilers by natural draft, and sent to the fan. As the fan starts the draft through the fire bed increases generating steam more rapidly in the boiler and increasing the pressure. As the pressure increases the fan will operate more efficiently and the steam boilers will soon be generating the steam required for the plant.

Whether a steam turbine or a steam engine should be used depends upon the speed and output desired. Steam turbines operate most efficiently from at 2000-3000 r.p.m. and at such a speed volumes as high as 194,000 CFM may be handled according to data from a catalogue of B.F. Sturtevant Co. Hyde Park, Boston, Mass. When still larger outputs are desired the size of the fan must be increased, and as the maximum speed is controlled by the strength of the fan blades the speed must decrease that the fiber stress at the tips of the blades shall not exceed a proper working value. To reduce the speed to lower values, say 100 R.P.M. reduction gears must be used with the turbine or the turbine be replaced by a reciprocating engine. In general a steam turbine will be used to blow relatively small quantities of air against a high static pressure, and a reciprocating engine will be used when it is necessary to handle large quantities of air but against a small static head. Turbine blowers will be used for forced draft and engine driven fans for induced draft.

Fans are often belt driven, allowing slow speed engines to be used to drive high speed fans. In places where there was a large cheap supply of gas, such as about steel works, the fans might be driven by gas engines. As the head against which the fan must work would be larger a high speed fan would be required. Gas engines run at a rather low speed, and a belt drive would be needed. One disadvantage of such a drive is that the fan must be started before any gas would be available for driving it, requiring either the use of a secondary gas supply, such as illuminating gas or a separate auxiliary method of driving.

Steam driven fans are preferable to motor driven fans for high pressure steam from the boilers may be used to supply the energy, and the exhaust steam may be used for heating purposes.

(IV) The Design of Fans

Under this heading we will discuss the general features of the design, while later under the performances of fans we will mention the relation of design to performance.

The principal dimensions of a fan should be made some function of the fan diameter, that is, the diameter across the tips of the blades, in this way comparisons of performance are facilitated.

a. Design of the Casings

If the air is given off equally for each unit of length around the periphery of a centrifugal fan wheel the casing should have a uniform increase in area outside of the wheel for the passage of air. The curve of the outer or scrollsheet of the casing is therefore that of the spiral of Archimides and its equation is $R = M + K a$.

Where R is the radial distance from the center of the wheel to any point of the curve, M, is the radius of the wheel, a the angle of advance in radius, and K a constant. The air will not be uniformly given off all around the circumference of the wheel except when the fan is properly loaded. If the load is too light the air will blow back at certain points while being delivered at others. The curve of the spiral scroll is stopped some distance from the point of intersection with the wheel and formed into the cut off piece. The value of K for a given diameter wheel is best determined by tests on the wheel. The point where the spiral, if extended, would intersect the wheel circumference varies and extends from about 30 degrees ahead of the point from where the angle of advance is measured to 30 degrees in back. The following table from tests on a standard form of multiblade fan shows the effect of varying the value of K. These tests were conducted by Mr. E.B. Williams.

Scroll Equation	Maximum	Efficiency	Percent
	Mech.	Vol.	Man.
$R = M (1\ 0.148a)$	48.5	367	130
$R = M (1\ 0.198a)$	52.0	452	128
$R = M (1\ 0.396a)$	36.0	493	117

Sometimes casings are made with outlets 180 degrees apart. In these the spiral is developed as for a single discharge fan stopping off at 180 degrees and starting at this point a second spiral. Although this gives the same total outlet area and the same velocity outside of the wheel with a given volume flowing as with a single discharge fan having the same equation the various efficiencies fall off.

With forward curved blade fans where the velocity of the air leaving the fan is high a scroll whose radius increases in a geometrical ratio is used. In this way a more complete transformation of kinetic to static pressure is obtained within the fan casing, in this way eliminating the need for an expansion piece at the outlet and thereby increasing the efficiency.

A fan wheel placed in a rectangular or circular casing or housing without a scroll will not give as efficient results as when a scroll properly made is used. An outlet projecting radially outward is out of the question as the air has to make a right angled bend to leave the casing, although at low speeds it may be used. As has been previously stated in order for a fan to work at its capacity the outlet area multiplied by its coefficient of discharge must not be less than the blast area. If the coefficient of discharge of the outlet is unity then:

$$W x = A$$

Where A is the area of the outlet

W is the width

X is the height of the casing above the wheel at the outlet.

Or $$X = \frac{A}{W}$$

By increasing W the width of the fan the height of the housing can be cut down. It is good practice to make the outlet opening wquare and to make the actual area greater than the blast area to allow for friction and for the fact that the coefficient of discharge is not unity. Many manufacturers make the width of the casing one half the diameter of the wheel, others make what is called a narrow fan whose width is say about three eights of the diameter of the wheel. The acroll is uaually approximated by the arcs of three circles. If X is the distance from the wheel to the top of the scroll at discharge, the radii of the three arcs may be as follows:

$$R_1 = \frac{D}{2}$$

$$R_2 = \frac{D}{2} + \frac{X}{2}$$

$$R_3 = \frac{D}{2} + X$$

For a fan whose width is about one half the diameter of the wheel and whose inlet is 5/8 of the diameter X is about 0.30 D and the following values for the radii are good practice in fan work.

$$R_1 = \frac{D}{2}$$

$$R_2 = .65\ D$$

$$R_3 = 0.80\ D$$

The sides pieces and the scroll piece of the housing of a fan when made of sheet steel must be braced with angle irons and made sufficiently thick to resist the pressure of the air and the straining action due to the movement of the wheel. There seems to be no particular rule or formula followed in determining the thickness of the plates used or the size of the angle irons used for bracing them.

Relation between the width of the casing and the width of the Wheel

In general low pressure volume fans have a width of casing 20 to 40 percent greater than the width of the wheel at its periphery. As the relative width is increased beyond this the air must distribute itself laterally to such an extent that serious eddy currents are produced in the casing, resulting in loss of efficiency. Fans with cast iron casings to be used for high pressures are often made with a casing of gradually increasing width. Casings are frequently made double width, double inlet which for a given volume, pressure and efficiency, gives a relative speed 41 percent higher. This has decided advantages where direct connected high speed drivers such as electric motors and steam turbines are used.

Size and Shapes of Inlet of Casing

In the earlier types of fans the diameter of the inlet was almost always made equal to one half the diameter of the wheel but in modern fans the diameter of the inlet is proportioned to the use to which the fans is to be put. If the fan is to work against comparatively low pressures and is intended primarily to move a large amount of air the diameter of the inlet is larger than it would be if the fan were used to work against high pressures and handle a comparatively small amount of air. The ratio of the diameter of the inlet to the diameter of the wheel is usually designated by the letter M. In most of the fans used for heating and ventilating work M is either equal to 0.625 or 0.707, but for fans which work against a considerable pressure M will be equal to 0.5 or even less. If the value of M is made much more than 0.7 it will be found that the fan cannot be used to operate against any pressure because the distance which the air travels in passing radially thru the wheel is so short that it does not have time to acquire the velocity of rotation of the floats.

The shape of the inlet connection should be conformed to the natural path of the outer particles of air by means of a conical or bill shaped inlet. From the manufacturers viewpoint the conical shaped inlet is to be preferred. Variations in the form of the inlet connections produces changes both in the mechanical and volumetric efficiency. Increases of efficiency (mechanical) of 10 percent have been obtained by changed orifices, the straight cylindrical inlet being replaced by a conical inlet.

The angle of convergance of the cone for best results is about 15 degrees and should not exceed thirty. Its larger diameter should be at least 25 percent greater than its least. Where a fan draws its air direct from the atmosphere or from the rooms in which it is placed and does not have a pipe attached to its inlet, it is of a material advantage to use a fan with two inlets. For the same volume of air handled fans having two inlets have shown an increased efficiency of 4 percent over those having but one.

Outlet or Discharge of Casing

The area of the outlet of a spiral casing is always made larger than the blast area previously discussed. A large velocity pressure exists at this point and in fans of small and medium size no attempt is made to conserve the kinetic energy of the blast at this point. In most installations the velocity pressure here is from 25 to 50 percent of the total head developed. Fans in general have outlet areas of from 25 to 75 percent greater than the blast area. The expansion is abrupt, so that although some of the velocity pressure is transferred into static pressure there is a considerable loss. If the fan discharges directly into the atmosphere the entire velocity pressure in the outlet is lost. Most of this can be saved however if the outlet is fitted with a proper evasé discharge piece. By this is meant a conical flaring connection between the cut off point say, and the outlet. If the fan is connected to a duct larger than the outlet it should have a long tapered connection.

Location of Cut off Point of Scroll

The point where the scroll or spiral discontinues its approach to the circumference of the wheel is called the cut off point. In some types of fans nearly half the diameter of the wheel is exposed when looking along the axis of discharge, while the other extreme is found where no part of the wheel is exposed, in which case a taugent to the wheel passing thru the cut off point is parellel to the axis of discharge. A fan with a large exposure of the wheel will deliver more air than a fan with a smaller or no exposure in cases where there are few restrictions to the flow of air. If the fan is to operate against average restrictions and considerable resistence, a smaller exposure of the wheel is best. Experiments have shown that cut off points with very small clearances have a merit which exists only in fancy. The effect upon efficiency of a reasonably large clearance at this point is negligable, the amount of clearance may be 5 percent of the wheel diameter in medium sized fans.

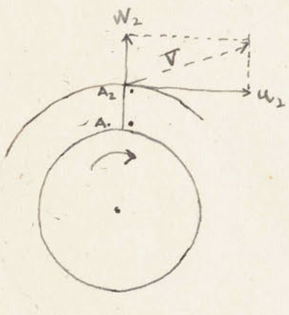
Types of Blades Used on Fan Wheels

a. Number of Vanes or Floats

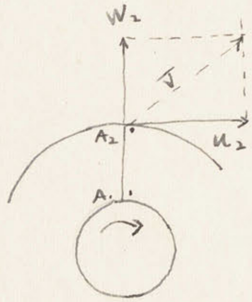
There does not seem to be any rule or formula for determining the number of blades a wheel should have. There should be enough to insure that the air passing from the wheel will give the same velocity that the wheel has before bearing the periphery, on the other hand it should not have so many so that the space between two consecutive blades becomes unduly narrow. In this case there is undue friction imposed to the flow of air from the center to the periphery of the wheel. Small sizes of wheels usually have six or more blades while the larger fans may have twelve. An increase in the number of blades involves constructional difficulties also. It becomes a matter of experiment to find the most desirable number of blades both from a viewpoint of volumetric capacity and mechanical efficiency. As the mumber of blades is increased the volumetric efficiency will still be on the increase after the mechanical efficiency has passed its maximum. A fan with many short blades has a greater volumetric and manomatric efficiency than one with a few long blades. Special types of fans put out such as a multi-vane fan may have as many as 50 to 100 vanes, these will be described later.

Blade Curvature

The blades of centrifugal fans may be either straight or curved. If the blade is straight and extends radially out from the center and the fan is usually spoken of as a steel plate fan. The blades may be either straight curved forward or curved backward in relation to the direction of rotation. The characteristics of blades curved forward and backward are quite different. The diagrams of what happens to the particles of air is instructive: Diagram number 1. shows the effect of cemtrifugal force on the resultant velocity.

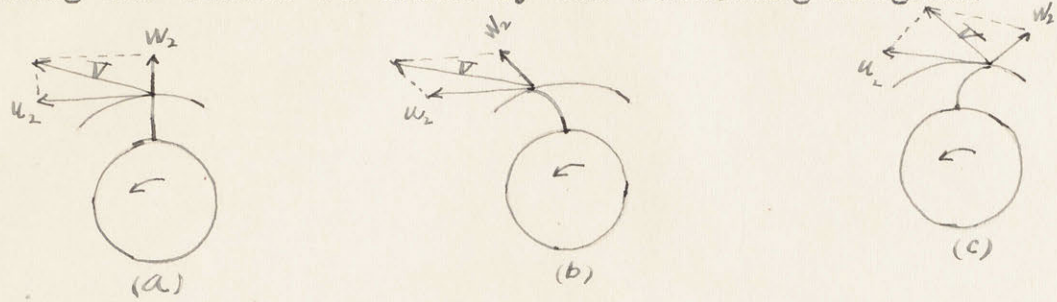


(1)



(2)

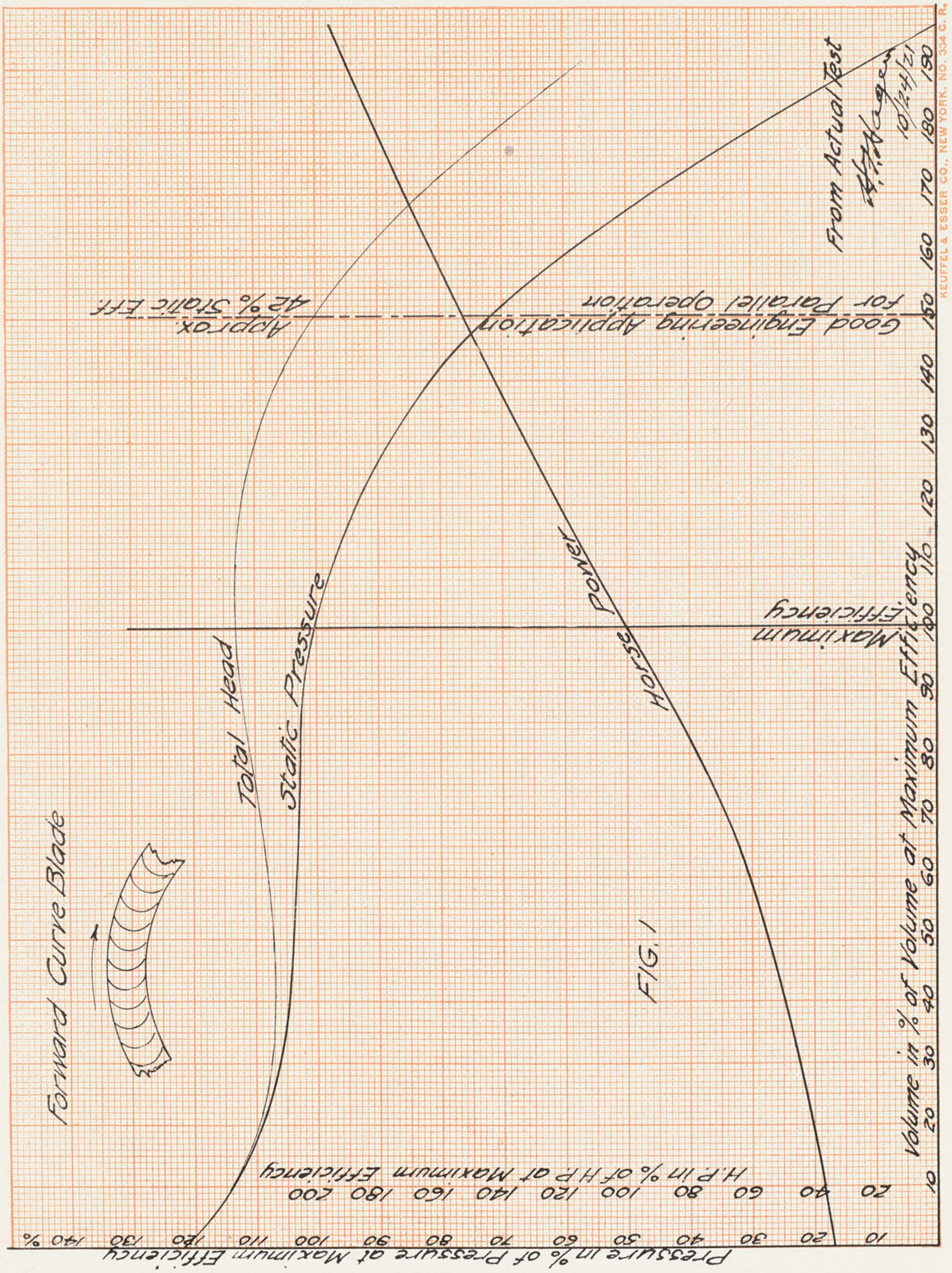
A particle of air located at A, the heel of the blade will have imparted to it a centrifugal force by the time it gets to H_2 . If we represent the velocity due this centrifugal force by W_2 it is evident from the sketches that for the same periphial velocity of the wheel the resultant velocity of the particle of air will be greater with a long blade than it would be with a short one, assuming both blades to be straight. That is the pressure for a given peripheral speed is increased by increasing the relative length of the blades. The method used for overcoming this fault of the short straight blade is to make it curved. The effect of curving the blades is shown by the following diagram:



It will be noticed that in a blade bent forward a pressure greater than that corresponding to the peripheral velocity is obtained. This results in the same pressure with a lower speed than would be necessary with a straight blade when using the same wheel diameter. On the other hand it is sometimes desired to direct connect a fan to a high speed unit without developing the corresponding high pressure. This is accomplished by bending the blade backwards so obtaining a pressure less than that corresponding to the peripheral velocity. The pressure characteristics of the straight and curved blade fans are also quite ~~fidd~~different. With a straight blade fan the pressure tends to build up as the load on the fan is reduced while operating at constant speed. Thus if such a fan be used to supply forced draft to a boiler, and due to the thickening of the fuel bed the discharge from the fan should be throttled, the pressure will be increased. This just the exact condition which is desired. On the other hand with a forward curved fan blade, as the air delivery is decreased the pressure would also fall off. Another case where this peculiarity is of importance is when a fan is required to operate part of the time at a considerable underload, yet a definite pressure must be maintained. For such a case the straight blade fan should be used.

A combination of the straight and curved blades have been used in which there are the many curved blades of the multi-vane fans and on the inside there are several straight blades set at an angle. The scooping action of the blades starts the air in motion gradually giving it a slowly whirling motion and then a rapid radial motion towards the small peripheral blades.

A type of fan called a propeller fan has been put out, the blades of this fan being in the nature of a screw. The probable action of a propeller fan may be compared to the ordinary vise, where the blade is the screw of the vise and the jaw is represented by the air.

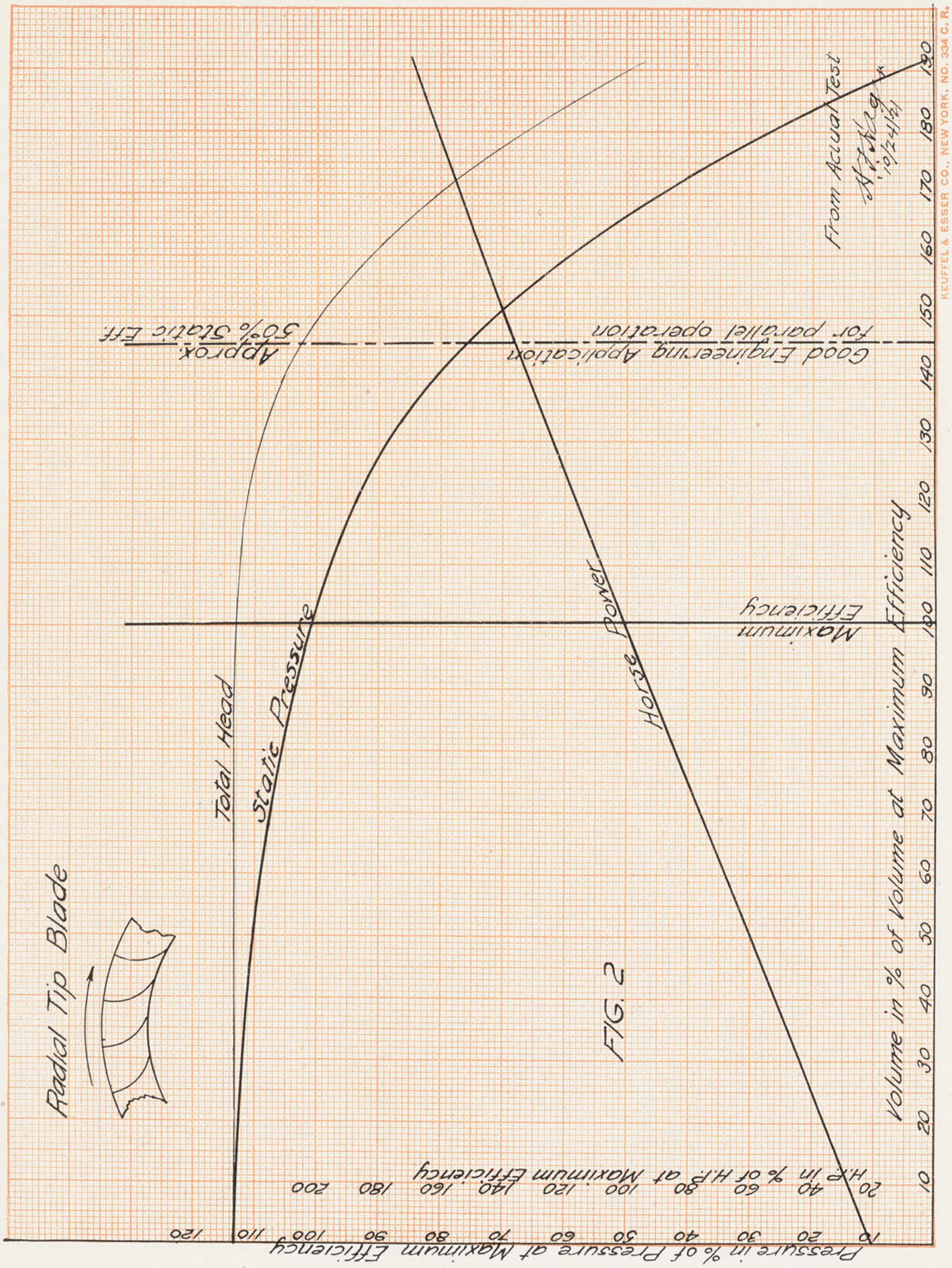

Forward Curve Blade
Total Head
Static Pressure
Horse Power
FIG. 1
Maximum Efficiency
Good Engineering Application for Parallel Operation
Approx. 42% Static Eff.
From Actual Test
10/24/21
Volume in % of Volume at Maximum Efficiency
10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190
H.P. in % of H.P. at Maximum Efficiency
20 40 60 80 100 120 140 160 180 200
Pressure in % of Pressure at Maximum Efficiency
10 20 30 40 50 60 70 80 90 100 110 120 130 140 %
KEUFFEL & ESSER CO., NEW YORK, NO. 334 C.R.

of a smooth flow of air, usually ten or fifteen diameters, measurements are taken with the double pitot tube of static and velocity pressures, from the latter of which is calculated the volume flowing. Next, a restriction, which may be of any convenient form, is placed across the end of the duct, thus causing a reduction in the volume delivered by the fan. With the fan still at the same speed measurements are made of the new volume and new pressure. This operation is repeated with further restrictions until sufficient points have been obtained to show the nature of the fan performance throughout its whole range. These test points are then plotted with volumes as abscissae and pressures as ordinates. A smooth curve drawn through the test points is the desired pressure volume characteristic of the fan. Simultaneously with the above determinations readings are made of the horse power consumed giving data sufficient for the plotting of the horsepower volume curve. These characteristic curves are most important. From them the proper size fan and the speed of that fan is calculated for any particular inquiry. It is too often not clearly understood that a fan can operate only on its characteristic. It is obvious that the curve we draw from our tests gives every volume and pressure that the fan can deliver at that speed. When we furnish a fan for a given installation we take care that its characteristic will run through, or usually a little above, the volume pressure point asked for by the customer. Fans driven by variable-speed motors or turbines afford the opportunity of taking care of errors in estimating stoker and duct resistance. It has seemed to me that seldom is there enough information asked for by engineers buying fans. For example, it is usual to request data for, say, 300% of rating, 250, 200, and perhaps 150% with the corresponding volumes and pressures. Skilled stoker operators can easily secure these ratings with the pressures specified. In actual plant operation the stoker may be indifferently operated by cheap, ignorant labor requiring higher pressures for the necessary volumes, thus operating the fans at points on their characteristics quite different from the ones given in the specifications, and for which no data was secured. If, however, curves of the fans were requested for the speeds corresponding to the desired ratings, it would be a simple matter quickly to observe the performance of the fan under any and all probabilities of required performance. You would also eliminate those manufacturers of fans who do not know how to test their product. Fan efficiencies are maintained high over a sufficient range of operation to make possible the selection of a fan to insure high efficiency with all usual variations in load. The only way to make certain of securing this desirable condition is to see from the curves that it exists.

A brief discussion of the nature of the characteristics of the various types of fans on the market today will serve to bring out further the importance of these curves in determining the suitability or unsuitability of any type for a given service.

Fans can be classified with respect to the direction of the blade at the periphery into four general classes: 1, the forward curve; 2, the radial tip; 3, the partial backward curve; and 4, the full backward curve.

The forward curve in a fan blade has a distinctive action in that the


Pressure in % of Pressure at Maximum Efficiency
10 20 30 40 50 60 70 80 90 100 110 120
H.P. in % of H.P. at Maximum Efficiency
20 40 60 80 100 120 140 160 180 200
Volume in % of Volume at Maximum Efficiency
10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190
Radial Tip Blade
FIG. 2
Total Head
Static Pressure
Horse Power
Maximum Efficiency
Good Engineering Application for parallel operation
Approx. 50% Static Eff.
From Actual Test
10/24/21
KEUFFEL & ESSER CO., NEW YORK. NO. 334 C. R.

rotational component of the air velocity leaving the wheel is actually higher than the rotational speed of the wheel itself. The result is a peculiar characteristic (see Fig. 1) which tends to rise with increasing volume until a point is reached when the velocity through the wheel is so high that the air can no longer follow the blade. The total head rises on all fans of this type, the static on some makes is flat and on others rises slightly. The maximum efficiency occurs where the static pressure just begins to fall. The horsepower curve of these fans is an upward curve, increasing rapidly with an increase in volume.

The radial-tip fan, a multiblade, has its blades curved tangent at the periphery to a radius of the wheel. In this fan the tendency is to maintain the pressure constant and, if well designed, it will give a total head practically flat to the point of extreme velocities, the static pressure steadily drooping but only very slightly (Fig. 2).

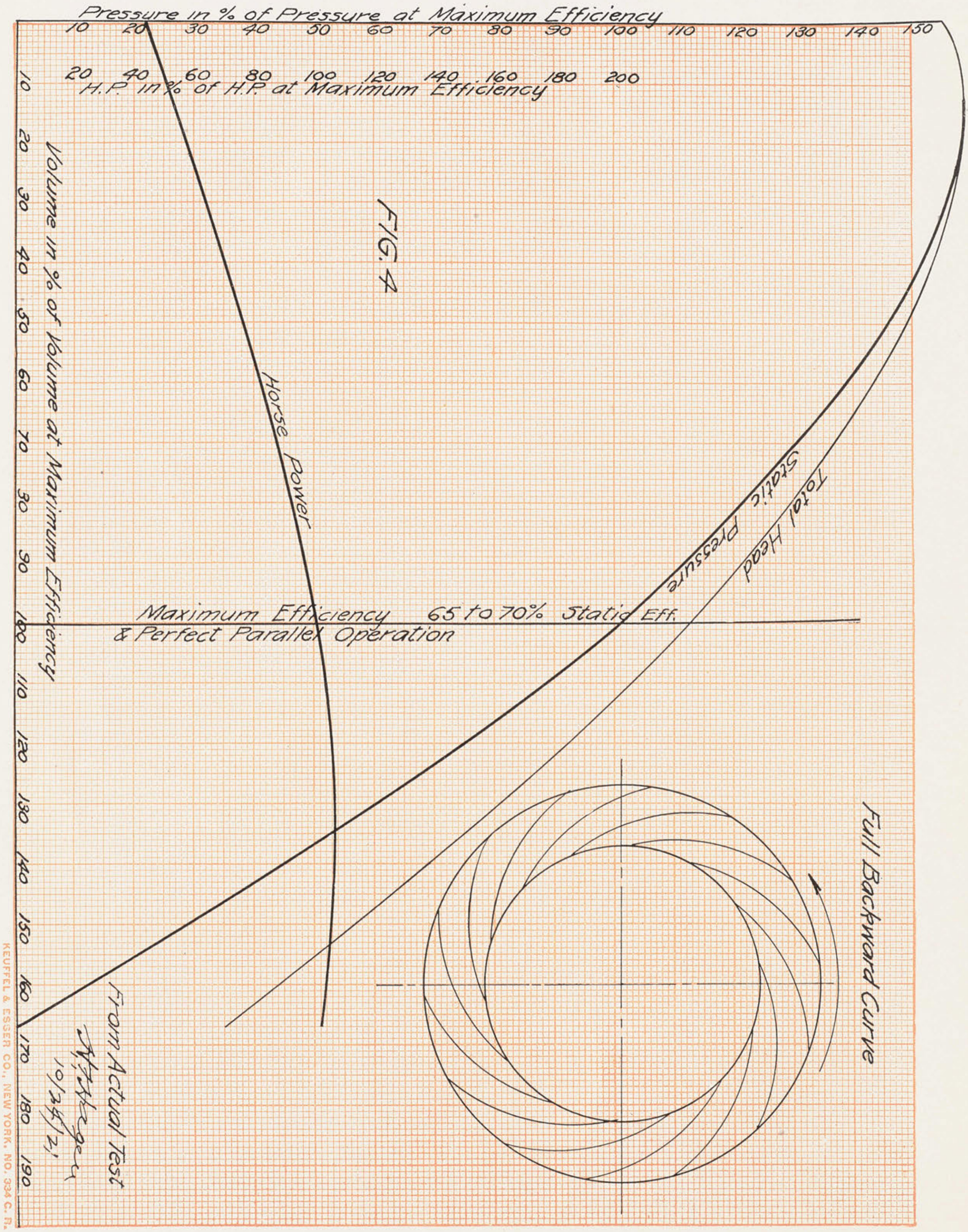
The partial backward-curve fan is one in which the backward curvature is not sufficient to affect the horse-power curve to secure the feature of self limitation. These fans have practically the same characteristics as the steel plate or few-bladed paddle wheels. The pressure curve tends to fall continually with the volume, although in the more efficient types there is a slight rise at low volumes. (See Fig. 3.) This is unimportant, as this portion of the curve practically never is in a working range of the fan. The maximum efficiency occurs where the curve is drooping markedly. This fan is an intermediate type, and not all the advantages of backward curvature are secured. It has the drooping pressure, but the horse power is still a straight line.

In the full backward-curve fan the blade curvature is carried back to a much greater degree. As might be expected, the pressure curve is sloped more than with any of the other types. A great change occurs in the horsepower curve. It is no longer a straight line, but a convex curve rising to a maximum and then, as the volume increases still further, falling away. (See Fig. 4.) This horse-power curve is typical of a backward-curve blade and is also found in the better-grade centrifugal pumps.

With all these characteristics before us we can discuss the selection from these types of a fan suitable to our particular problem, that is, stoker draft. Suitable forced-draft fans are an absolute essential for the underfeed stoker. We may set down the requirements of a fan for this duty as follows: first, reliability; secondly, successful parallel operation; thirdly, a reserve of pressure; fourthly, a high static efficiency and, in view of turbine and motor drive, a high speed.

The first requirement, that of strength and reliability, can be successfully met with any type fan and necessitates only purchasing from a reputable manufacturer.

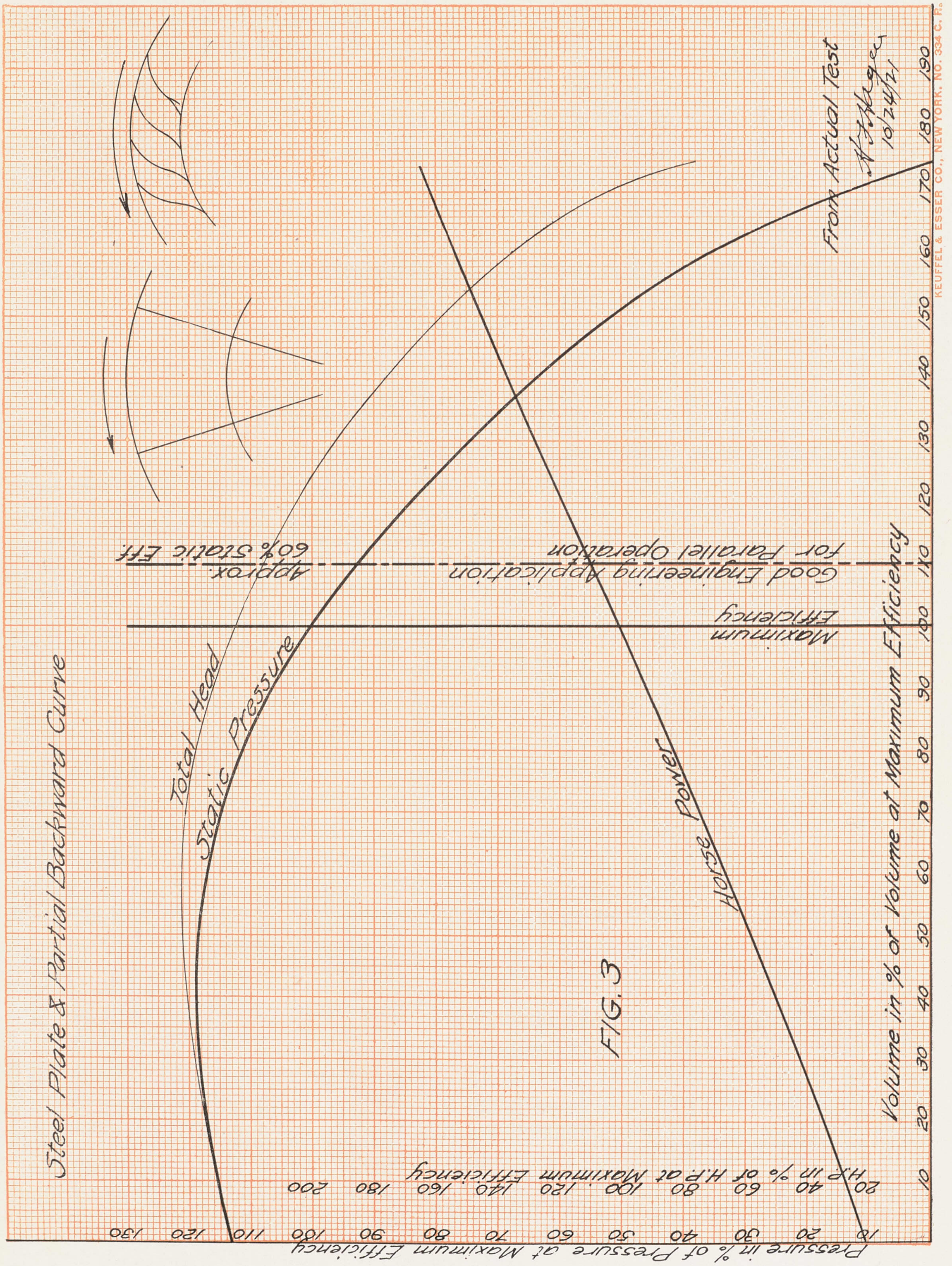
The second requirement, parallel operation, practically eliminates the forwardly curved blade fan. If two or more fans blow into a common duct, it is apparent that the only determining factor in dividing the load will be a common pressure at the fan outlets, which pressure will be that of the main duct. And with the characteristics of this fan there is no assurance that one of the fans will not lie down on the job, especially if the flow of


Pressure in % of Pressure at Maximum Efficiency
10 20 30 40 50 60 70 80 90 100 110 120 130 140 150
20 40 60 80 100 120 140 160 180 200
H.P. in % of H.P. at Maximum Efficiency
Volume in % of Volume at Maximum Efficiency
10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190
FIG. 4
Horse Power
Static Pressure
Total Head
Maximum Efficiency 65 to 70% Static Eff.
& Perfect Parallel Operation
Full Backward Curve
From Actual Test
10/24/21
KEUFFEL & ESSER CO., NEW YORK, NO. 334 C. R.

Fans with the steel-plate paddle-wheel characteristics are suitable for stokers. They will operate fairly successfully in parallel. They have an increase in pressure with a decrease in volume, which I have elsewhere termed reserve pressure, throughout the efficient working range of the fan. The speed of the steel plate is too low for modern drive, but the partial backward curve has a reasonably high speed making it a practicable apparatus. It fails of complete desirability in that its efficiency is not so high as that of the full backward curve, and its horse-power curve is the straight line and not self limiting.

With any of these types already discussed it is impossible to give unreservedly the horse power required in a given installation. We can say accurately that when they are delivering a certain volume against a certain pressure they will take a definite horse power; but it is seldom that the exact volumes and pressures required on the job are known beforehand. The amount of leakage, which is seldom accurately estimated, the skill with which the stoker will be operated, the number of fans run for a given load — all have an important bearing on the volume and pressure required from each individual fan. It is desirable under these conditions to have a fan with a self-limiting horse-power curve. This almost essential feature is possible only in a well designed backward-curved blade fan. The maximum possible horse power can be made to coincide very closely with the peak of the efficiency curve. With this combination of characteristics we can say definitely of this type of fan that it will take a certain horse power. We can select our driver for the horse power and be absolutely sure that it can never be overloaded. This fan has a higher speed than any of the others. Its efficiency is inherently the highest, as a large part of the energy is already static pressure when the air leaves the impeller, and it requires, therefore, only a relatively small conversion of velocity into static in the housing. This conversion in the housing is a very difficult matter and must be so from the nature of the flow of air from the impeller.

This type gives us a greater reserve pressure than any of the others. This reserve pressure, indicated by the slope of the pressure curve, is of utmost importance to successful stoker practice. Most of the resistance to the flow of air is in the fuel bed itself, and this is changing continuously and rapidly. If the fuel settles, or clinkers start to form, a greater resistance is offered to the air flow. The prayer of the operator, then, is for more pressure. The comparatively slow action of any automatic regulator, while useful, of course, may not come until the fire has lost some of its clearness. When you are operating at 300% of rating it does not do to lag at all on the fire. If, however, the fan of itself without change of speed will supply a higher pressure and a considerable increase in pressure coincidentally with any decrease in volume, the fuel bed will be opened up immediately, and a drop in boiler pressure averted. This feature the full backwardly curved blade fan possesses to a greater degree than any other type. It furthermore operates perfectly in parallel, for in addition to the inherent property of load balance, which is also a property of the steel plate, it has the further advantage of the self-limiting horse power. This latter is a fool-proof feature. I remember a case in a textile mill which had just completed the


Steel Plate & Partial Backward Curve
Total Head
Static Pressure
Horse Power
FIG. 3
Maximum Efficiency
Good Engineering Application for Parallel Operation
Approx. 60% Static Eff
From Actual Test
Volume in % of Volume at Maximum Efficiency
Pressure in % of Pressure at Maximum Efficiency
H.P. in % of H.P. at Maximum Efficiency
KEUFFEL & ESSER CO., NEW YORK, NO. 334 C. F.

air to the inlets of the two fans is not equally unobstructed. With steam turbine or variable-speed motor drive another element enters. The fans, I may safely say, never continue all at the same speed, and any change in relative speed greatly increases the inherent unbalancing of this type. However, this forward-curve fan does not need either a restricted air supply or a change in relative speeds to perform erratically when in parallel and should never be so used for any purpose. Aside from failure to deliver the air against the pressure required, the horse-power curve with its upward bend makes a very ready and ever-ready method of overloading the driver. In addition to this unbalancing, which will occur under the most careful operation, there will be in most plants some bright operator who will start up only one fan for the whole system, thereby taking a very large volume from that one fan and burning out the driving motor if at all possible. It is possible if the horse-power curve of the fan, as is the case with the type we are considering, will permit the overload.

This type would also be the last selection from the standpoint of the high R.P.M. necessary for direct connection. It has the slowest tip speed for a given pressure of any type. This slow tip speed makes it first choice for some work, but practically rules it out of forced-draft service. Also, it has no reserve pressure, that is, a reduction in volume is not accompanied by an increase in pressure. The desirability of this feature will be commented on in the discussion of another type. Further, its efficiency is fundamentally low. In general it would be difficult to design a fan more unsuitable to the work required.

In spite of all this, these forwardly curved blade fans have been installed successfully on stokers, although the percentage of satisfactory jobs is low. I know of one stoker salesman, who is at this convention, who was in the main successful with them. He is a man of wide experience and knows enough about fans and their characteristics to adapt even this unsuitable type to his work. He would select a size fan that would enable him to work on the falling portion of the curve. The efficiency was very low, under 50%, and he would then put in the neighborhood of 50% excess capacity in his motor or turbine to take care of the probable increases in horse power. However, at the present time power-plant engineers are rightly demanding very much higher efficiencies than these, making it impossible to sell on this basis, the only correct basis for the type. This same salesman was one of the first to realize the possibilities of other designs and to use them in his installations. The forward-curved fans have their good points, adapting them to certain work, but that work most emphatically is not forced draft for underfeed stokers.

The radial-tip fan also does not fulfill the requirements. It leaves much to be desired when operating in parallel, as a slight change in relative speeds means a large change in volumes. This serious unbalancing between two or more fans can always be expected if the pressure curves are flat. Even a slight change in speed produces a change in pressures, and with flat characteristics a small difference in pressure is sufficient to cause a wide variation in volumes before equilibrium is again established. This fan further has little reserve pressure; its speed is low, as is likewise its efficiency.

Development of Centrifugal Fans

The type and proportions of a fan depend upon the work it has to do, and as the earlier fans were used mostly for the ventilation of mines, they were designed to handle large quantities of gas against comparatively low heads, producing slight vacuums in the shafts of the mine and causing a flow of air through them. They seldom had a casing or housing except when necessary to protect them from the weather or injury from external sources.

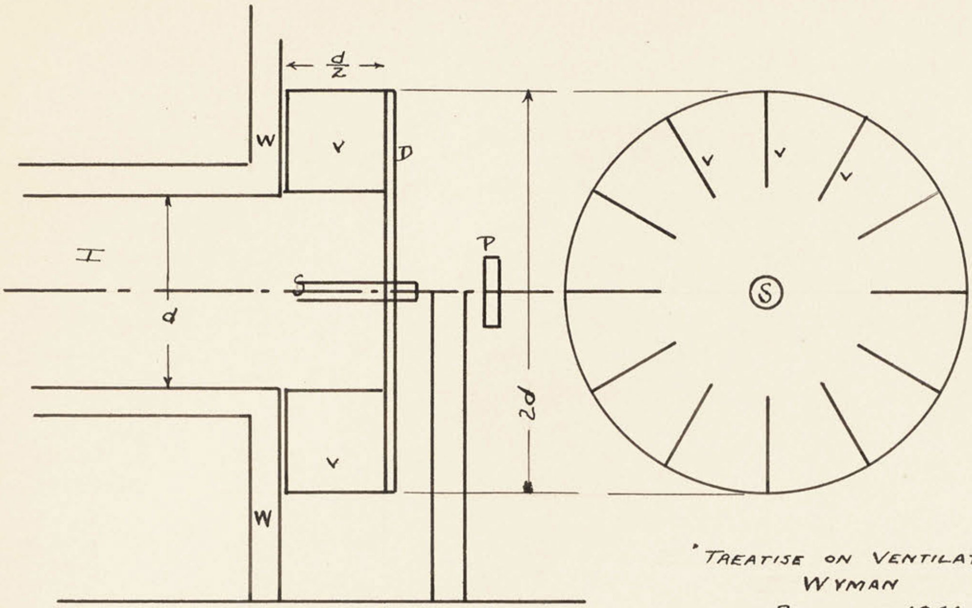
This first type was followed by a second which did have a casing enabling them to be used either as exhausters, taking air slightly below atmospheric pressure and discharging it to the atmosphere, or as blowers, taking atmospheric air, increasing its pressure and discharging it at a higher pressure.

Next followed another type in which the shape of the housing was changed. This was the forerunner of the modern centrifugal fan used for heating and ventilating work, and producing mechanical draft whenever it is necessary to handle large quantities of air against slight resistances either as exhalators or blowers.

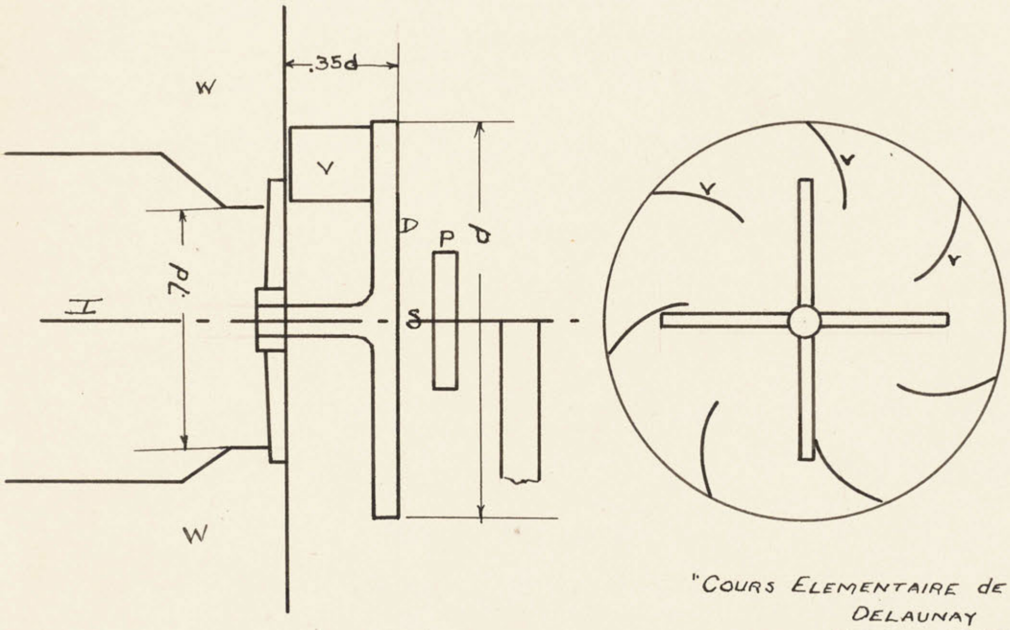
The first type is represented by the fans used in England and Europe previous to 1830, the second may be credited to Guibal of France, the third, an outgrowth of the Guibal fan is sometimes referred to as the Schiele type.

The first type of fan, used to replace the ventilation by building a fire at the bottom of a mine shaft, was mounted on the surface at the head of the shaft, sucking in air at its center and discharging in all directions to the atmosphere. The wheels were usually large, some being twenty five feet in diameter, having an inlet with a diameter about six-tenths that of the wheel, located on one side of the wheel only. The wheels were mounted either with their axis vertical or horizontal. When vertical they were located directly over the shaft, when horizontal they were connected by a horizontal duct from the top of the shaft. The vanes of the first fans were commonly straight placed radially and held by a revolving disk fastened to the shaft. The figures on the following page show two of these earlier fans, one having straight radial vanes and the other curved vanes. In each figure, I is the inlet, W the wall, V the vane fastened to the movable disk D revolving on shaft S which is driven thru pully B. The radial vane fan can be revolved in either direction, the curved vane fan was to turn as shown by the arrow. Tests made upon some of the earlier types show the radial vane fan to be the more efficient.

The second type, shown on following page has a housing which fit closely about the vanes, and discharged at a single point on the periphery. Each fan was so built that the efficiency would be a maximum for a certain discharge area. Since calculation of this best area was impracticable the area was varied variable by means of a screen E which worked in groove F. The vanes were bent forward slightly at the periphery and back about 45^{o} at the inlet. The discharge flue was tapered to prevent a drop in head by sudden decreases in velocity, and this feature is often credited to Guibal. Because of ability to use this fan as either an exhauster or blower it was located either at the top of

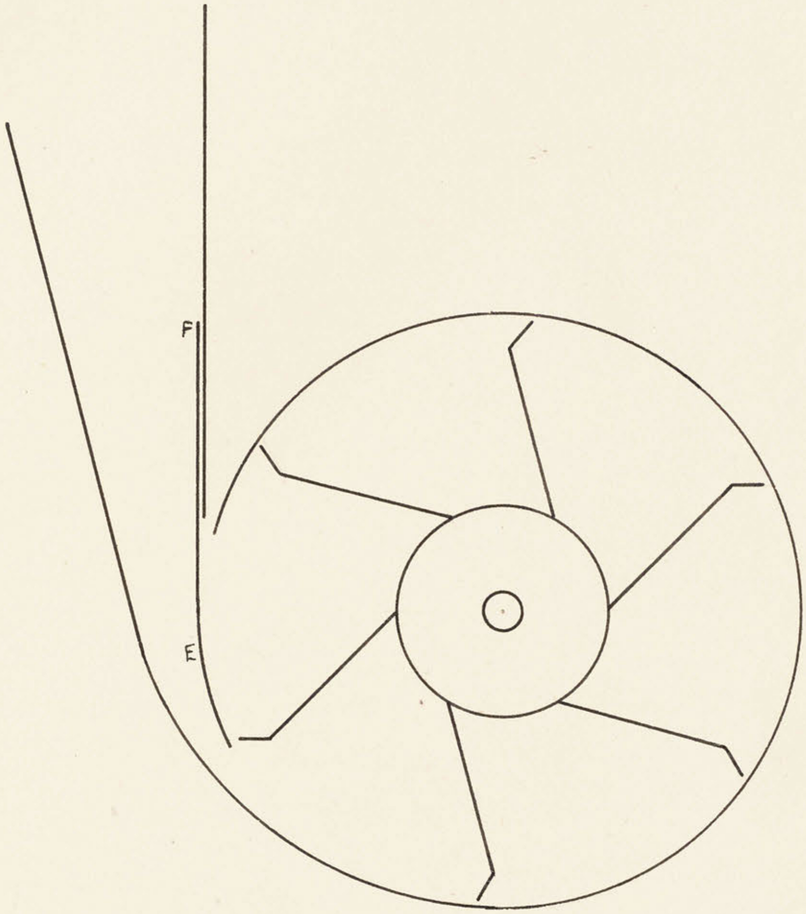


"TREATISE ON VENTILATION"
WYMAN
BOSTON — 1846.



"COURS ELEMENTAIRE de MECHANIQUE"
DELAUNAY
PARIS — 1854.

FANS OF FIRST TYPE.

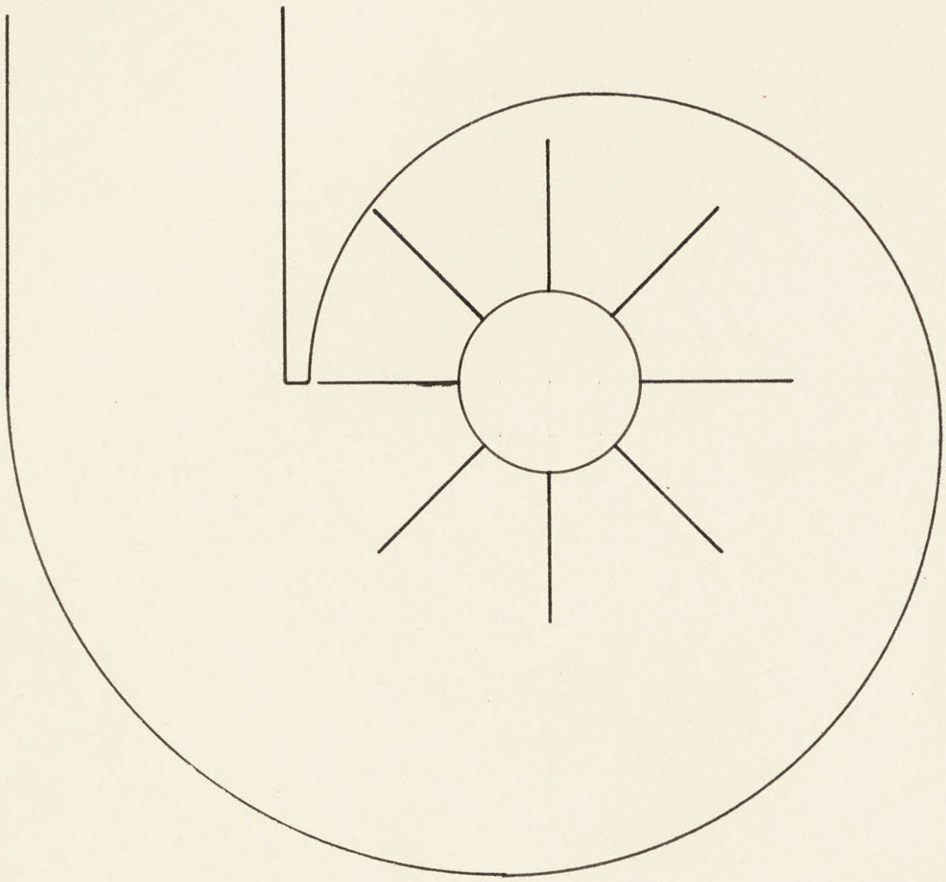


THE GUIBAL FAN AND CHIMNEY

the mine shaft or near the bottom, whichever was the most convenient.

One disadvantage of this second type was its intermittent action. Air would flow thru the space between two vanes only when it could escape thru the outlet. In the third type of fan the flow was made continual by moving the housing away from the tips of the fan, starting with clearance room just after the outlet and gradually increasing to a maximum at the outlet. The figure of one of these types shown on a following page makes it appear that the form of the casing was an Archimedes spiral, but actually the form was either circular or merely an approximation of the sprial, being classed as a "two radius," "three radius" or "four radius" spiral depending upon the number of arcs used. These fans were usually made smaller than the first two types and run at a higher speed. Sometimes an inlet was provided on each side of the casing the blades being connected to a rotating disk in the center.

Modern fans are merely developments of the third type. Attempts have been made to use both the spiral casing and the Guibal Chimney, to prevent losses due to sudden reduction of the velocity head. The earlier designs has few vanes. In 1884 Professor Ter of Paris designed one of the first multivane fans, consisting of thirty-two curved vanes mounted upon a central revolving disk. Two features were claimed for this fan, both of which gained considerable prominence in the patent granted the Davidson fan in 1899, the unobstructed air in inlet, and the advancement of the tip of the curved blade before the heel.



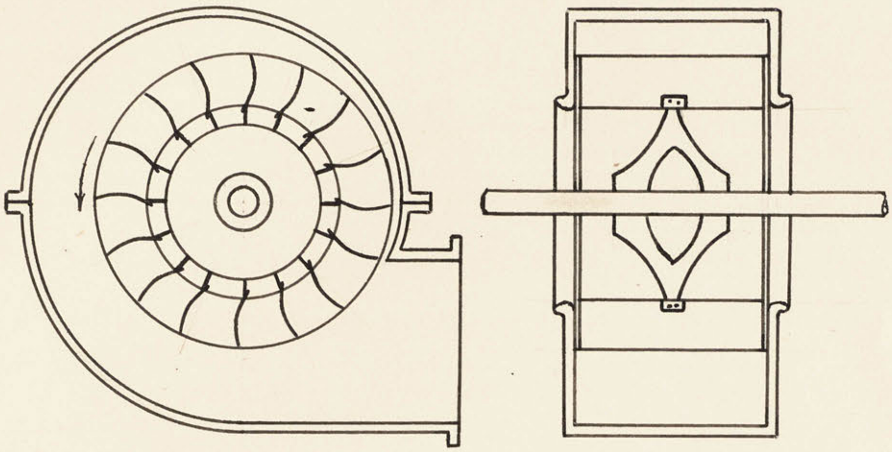
THIRD TYPE OF FAN

The immediate predecessor of our modern narrow blade multivane type was patented in France in 1896. The fan consists of a wheel having curved blades mounted on a central revolving disk. This fan also has the unobstructed air inlet. According to the patent the blades of this fan are twice as long as their radial dimension, and are spaced the blade depth apart.

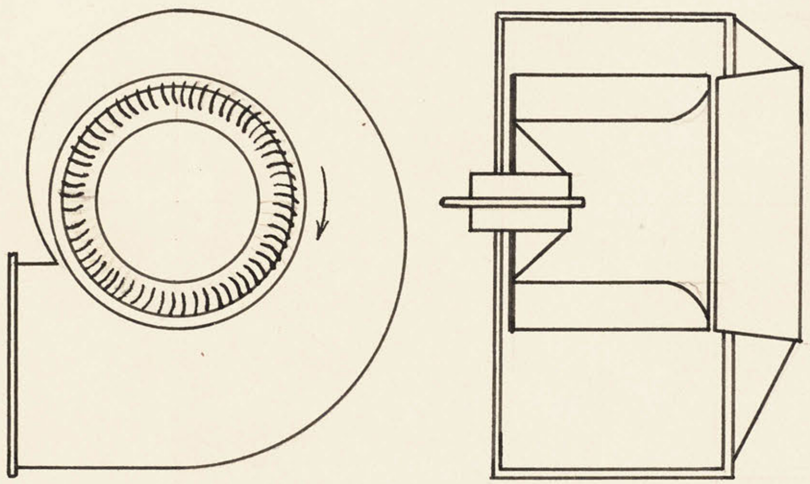
The fan of the Davidson patent, 1890, more commonly known as the Sirocco Fan, had blades much narrower radially and much closer spaced than the previous fans. The blades are specified as at least three times their depth in length and spaced apart two thirds of their depth. In the earliest design the blades projected their entire depth into the inlet circle. The design has been modified so that today the blades do not project into the inlet circle at all. The pipe connected to the outlet is made some 150 percent of the discharge area, decreasing the discharge velocity and increasing the statical pressure at discharge.

The shapes of the fan blades now used depend upon whether large quantities of air are to be handled against low resistances or smaller quantities against greater resistances. The fan having few radial blades offers less obstruction to the passage of air than does the multivane fan and should prove more efficient. The speed at which the fan is to run will also determine the curvature of the blades, high speed without correspondingly high pressure may be accomplished by bending the blade backward, low speed with accompanying high pressure by bending the blade forward.

Air leaving the fan has a velocity resulting from



FOURNIER & CORNU PATENT - 1896.



"SIROCCO"

THE DAVIDSON FAN - 1900.

components due to centrifugal force and the peripheral velocity. This velocity produces kinetic energy for moving the air and also sufficient excess to overcome the pipe resistance after its conversion to potential energy. To properly care for this change the housing must be properly designed, ever increasing in cross section as it nears the outlet, and sometimes the size of the outlet has a gradually increasing cross section.

Modern practice tends to favor the multivane fan "the real advantage of which is the attainment of fairly high efficiency in a more limited space, which makes it of great commercial value for certain classes of work. In case an increasing pressure is desired with an increasing resistance, it has been proved from the pressure curves that the forward curved type is not applicable unless operated beyond its most economical point. On the other hand, it is frequently desired to maintain a constant or increasing pressure with an increasing in capacity. In such cases the forward curved type is the only fan capable of accomplishing desired results. Another important advantage of the multivane type is the fact that its higher speed makes it more suitable for direct connection to motors or at least gives better pully ratios than may be obtained with radial blade fans." The foregoing statement appears in the conclusion of a paper written by Frank L. Busey for presentation to the American Society, Heating and Ventilating Engineers, at annual meeting January, 1915.

The Effect of Various Shaped Inlets on Fan Performance

The authors of this paper conducted a series of tests 42 in number to determine the effect, if any, of various shaped inlets on the performance of a fan. The fan used was a three feet diameter multivane fan with forward curved blades. The fan was motor driven and the data was obtained in the usual manner as described before under a typical fan test. The inlets were three in number.

(1) A plain circuler inlet as is ordinarily found on fans.

(2) A straight cylindrical inlet eighteen inches long.

(3) A conical inlet eighteen inches long with a ten inch increase in diameter over the short end.

The output of the fan was determined by means of a Pitot tube. While the input was determined by finding the motor output; the motor losses being accurately determined. The results of the tests were as follows:

(a) The capacity of the fan was increased about 100 cubic feet per minute on the working range by using a conical inlet. A cylindrical inlet caused a decrease in capacity of about the same amount.

(b) The effect of various inlets on the dynamic and static heads was negligable, the variation not being enough beyond the accuracy of the readings.

Conical Inlet Full opening A= 1.97 ft'							
RPM	I_a	I_f	V_L	T	D.P.	V.P.	$\sqrt{VP}$
210	8.25	1.38	110	76	0.20	0.20	0.448
351	22.5	1.35	185	76.5	0.57	0.52	0.719
404	28.5	1.31	208	77.	0.77	0.68	0.822
449	36.9	0.97	202	77	0.95	0.84	0.914
487	48.6	0.79	192	77	1.15	1.00	0.985
542	67.5	0.63	178	77	1.39	1.19	1.090

Without Inlet Full Opening A = 1.97 □.							
R.P.M.	I_a	I_f	V_L	T	D.P	V.P	$\sqrt{V.P.}$
212	11.3	1.33	107	77.	.20	.20	.445
375	22.5	1.32	183	77.	.60	.55	.743
402	27.9	1.31	207	77.	.78	.69	.813
485	49.5	0.78	190	77.	1.17	1.03	.931
535	61.5	0.68	179	77.	1.35	1.15	1.073
558	87.9	0.52	161	76.5	1.54	1.32	1.150

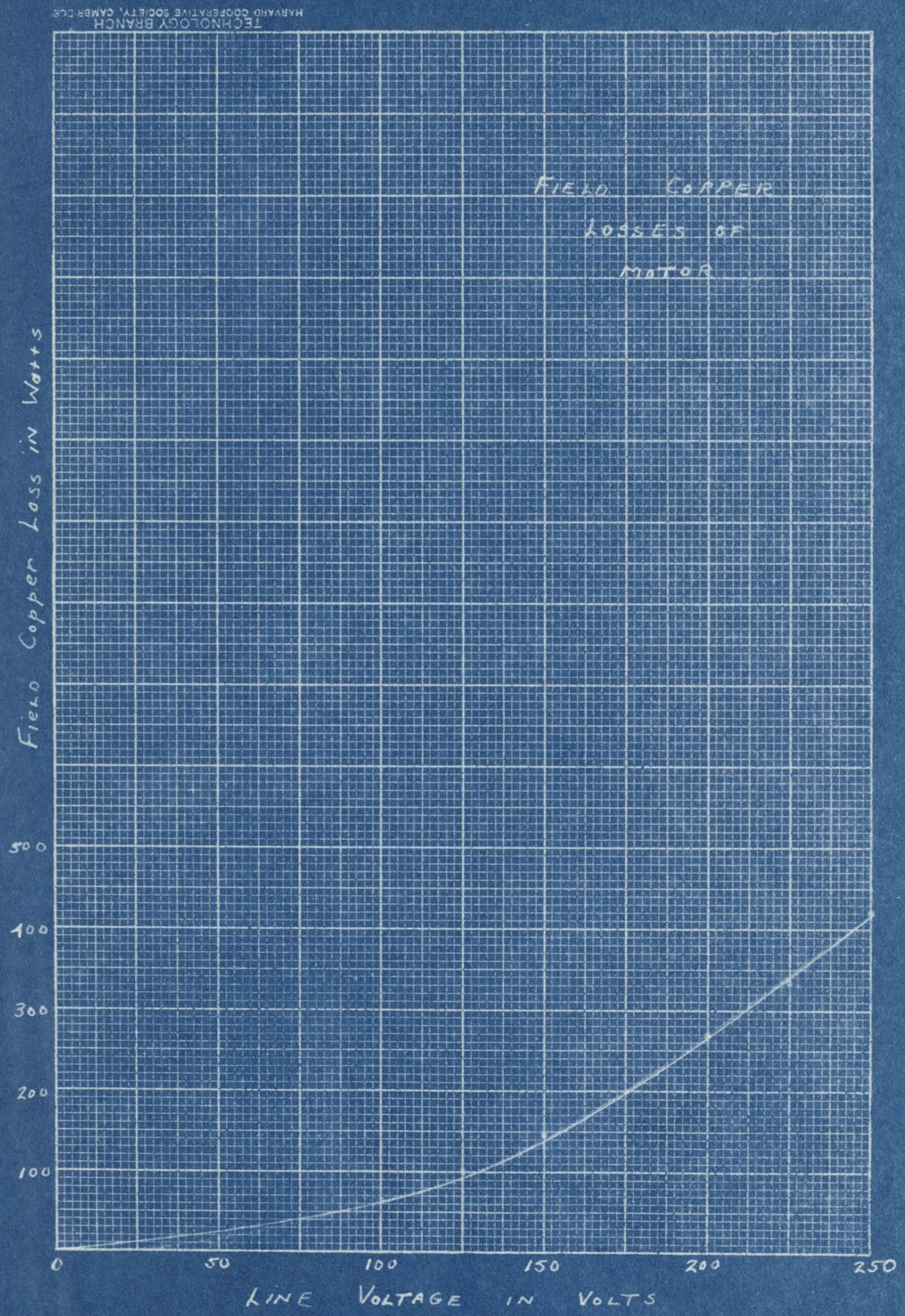
Straight Cylindrical Inlet Full Opening $A = 1.97$ □"							
RPM	I_a	I_f	V_L	T	D.P.	V.P.	$\sqrt{V.P.}$
207	11.4	1.47	110	71.	.17	.19	.440
263	15.9	1.42	140	72.	.30	.29	.537
303	19.5	1.45	163	73.5	.40	.38	.614
354	24.0	1.39	187.5	73.5	.55	.50	.708
369	25.5	1.39	198	74.0	.62	.55	.737
395	28.5	1.38	209	73.5	.70	.64	.795

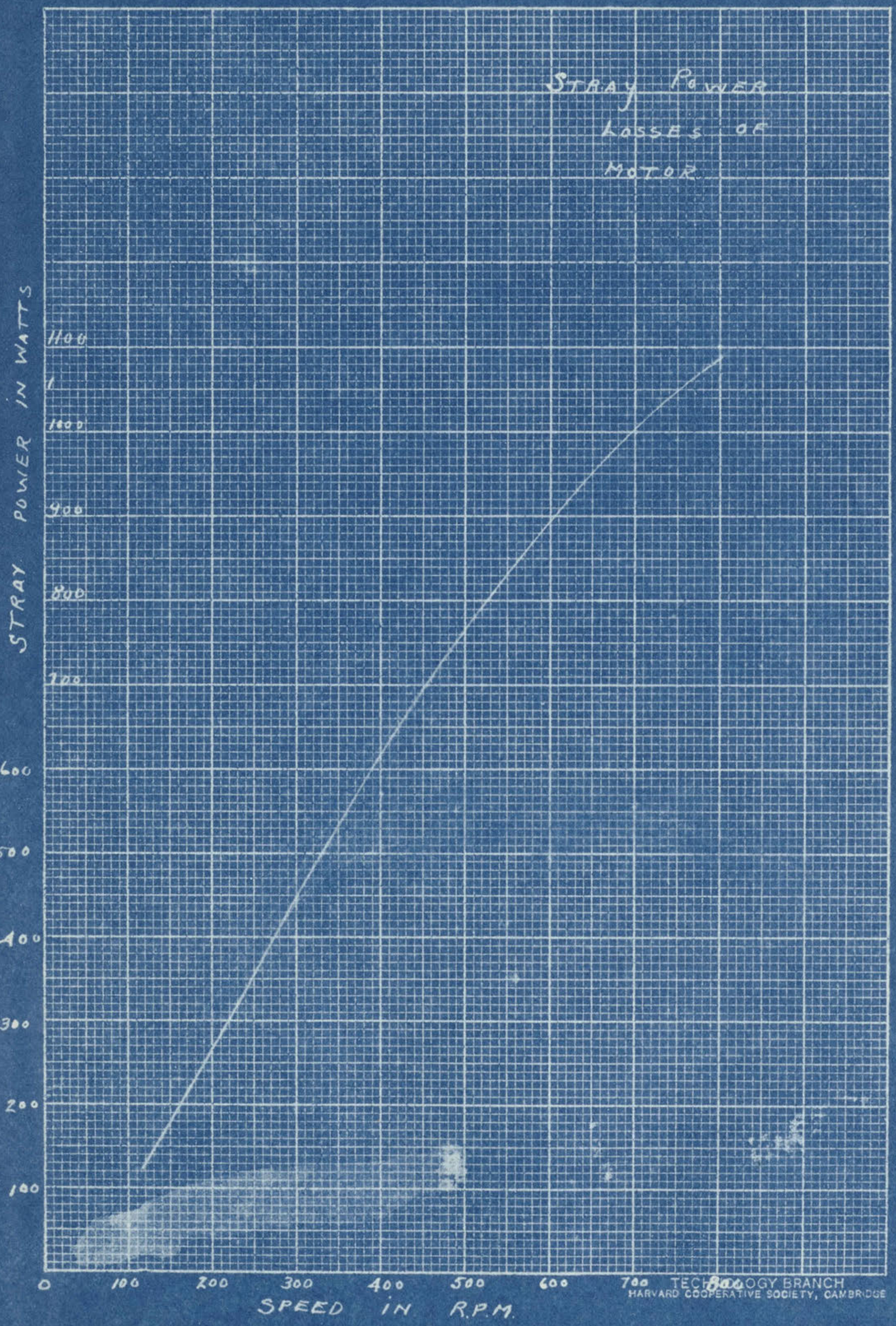
Straight Cylindrical Inlet Full Opening A = 1.974							
RPM	I_a	I_f	V_L	T	D.P.	V.P.	$\sqrt{V.P.}$
408	30.5	1.26	210	74.5	0.76	0.67	.819
451	37.5	1.00	205	74.0	0.88	0.80	.895
514	55.5	0.75	192	74.0	1.18	1.02	1.007
556	76.5	0.59	175	75.0	1.41	1.20	1.091

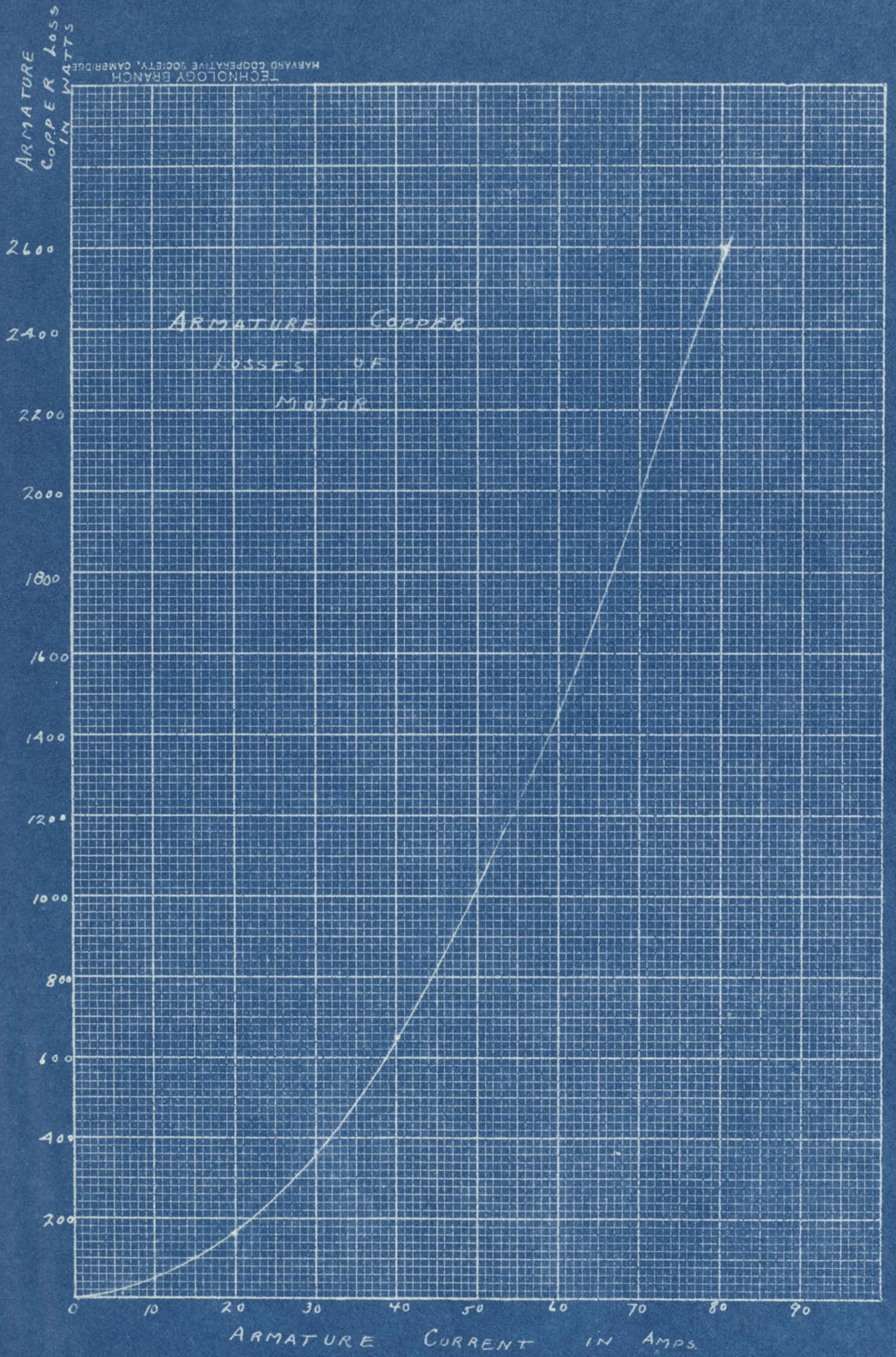
Straight Cylindrical Inlet $\frac{1}{2}$ Opening $A = .892$ □'							
RPM	I_a	I_f	V_L	T	D.P.	V.P.	$\sqrt{V.P.}$
250	10.5	1.39	130	75.5	.31	.08	.276
347	15.	1.38	181	76	.63	.11	.335
393	18.	1.36	209	76	.83	.14	.376
409	19.5	1.36	218	76	.92	.15	.387
523	31.5	0.81	208	76	1.47	.22	.466
644	54.	0.58	192	76.5	2.11	.30	.551

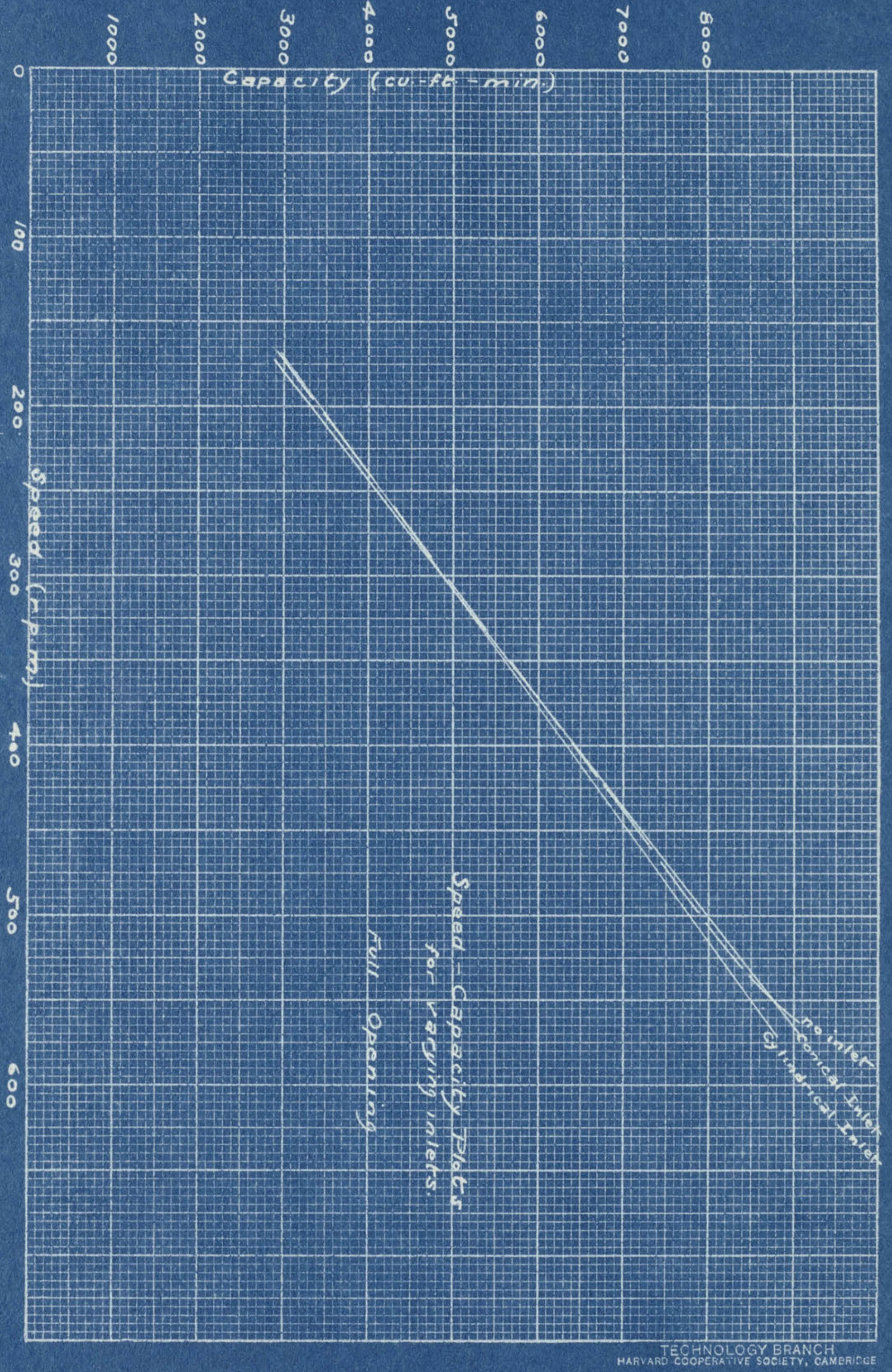
Without Inlet $\frac{1}{2}$ Opening A = .892 □'							
RPM	I_a	I_f	V_L	T	DP	VP	$\sqrt{VP}$
242	10.5	1.36	127	76.5	.31	.08	.274
343	15.6	1.35	179	77	.62	.11	.337
402	19.5	1.35	207	77	.84	.14	.375
426	20.3	1.28	215	77.5	.94	.15	.391
532	34.5	0.88	202	77	1.53	.23	.480
634	58.5	0.53	184	77.	2.13	.31	.559

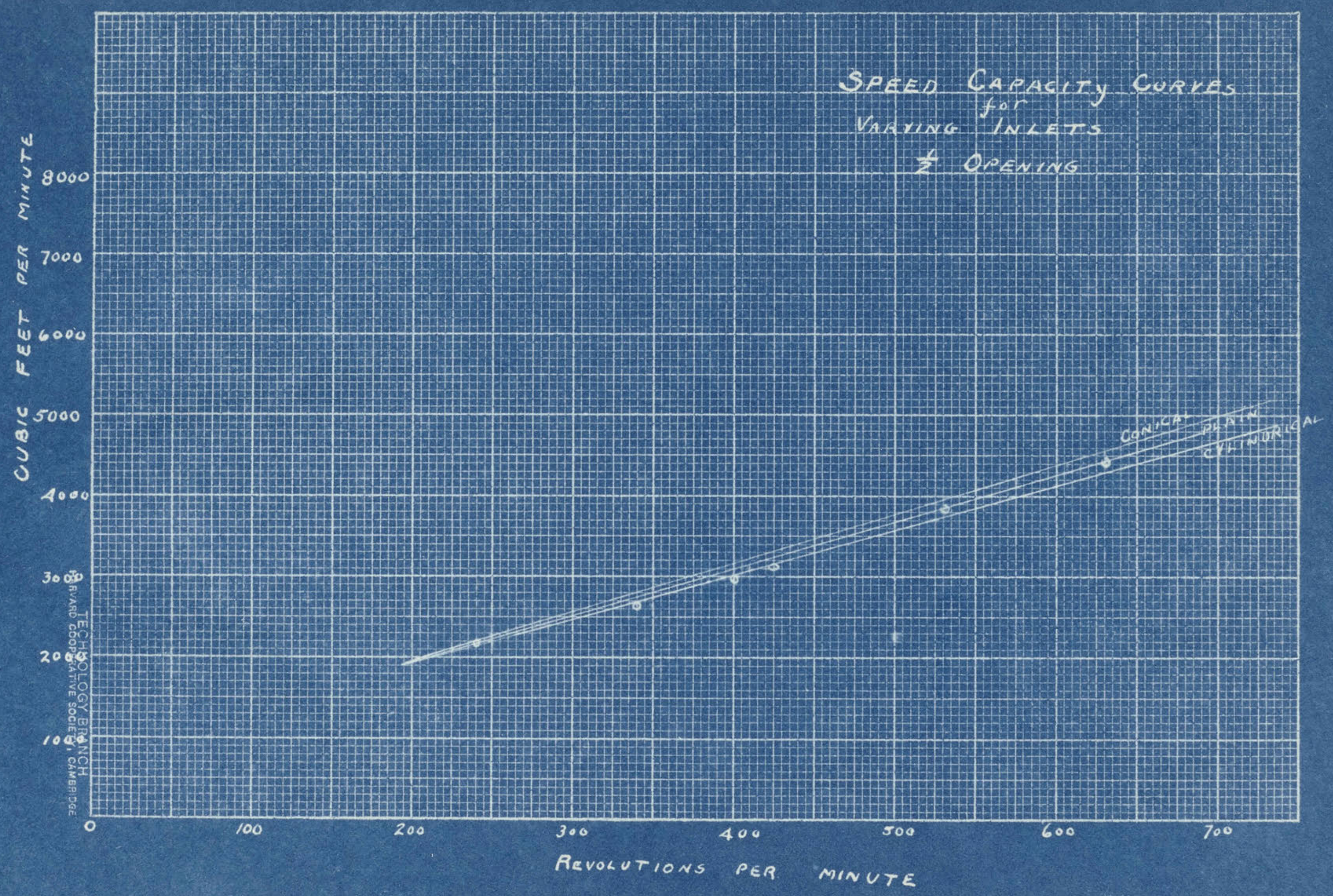
Conical Inlet $\frac{1}{2}$ opening $A = .892 \square_1$							
R.P.M.	I_a	I_f	V_L	T	D.P.	V.P.	$\sqrt{V.P.}$
247	11.5	1.35	126	77	0.30	0.08	.286
384	16.5	1.35	200	78	0.78	0.13	.365
419	19.5	1.29	218	78	0.95	0.15	.390
534	32.3	0.81	208	78	1.49	0.23	.478
575	42.6	0.68	201	78.5	1.79	0.27	.516
634	57.9	0.55	190	78.5	2.20	0.33	.570


HARVARD COOPERATIVE SOCIETY, CAMBRIDGE
TECHNOLOGY BRANCH
FIELD COPPER
LOSSES OF
MOTOR
Field Copper Loss in Watts
500
400
300
200
100
0
50
100
150
200
250
LINE VOLTAGE IN VOLTS

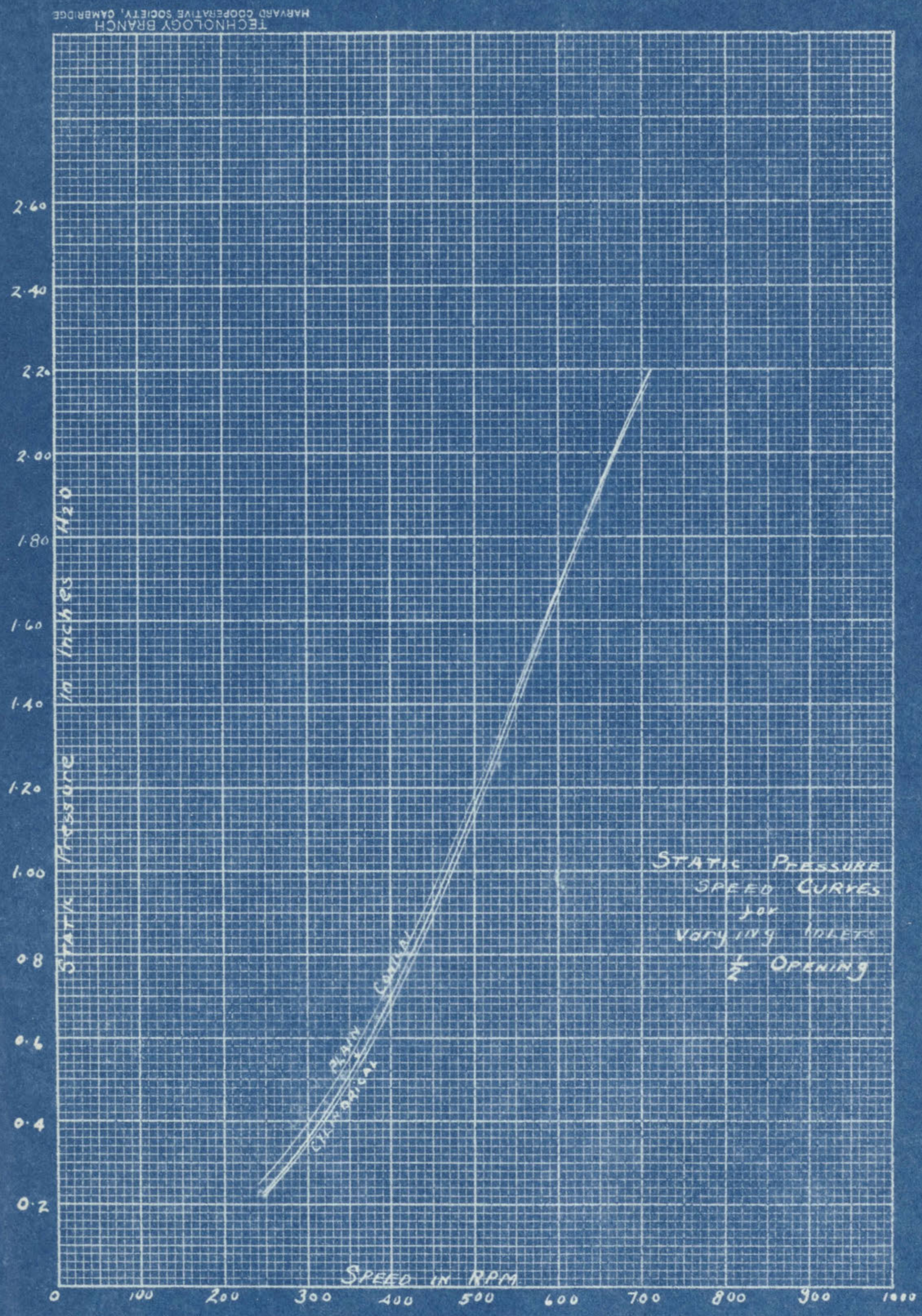

STRAY POWER
LOSSES OF
MOTOR
STRAY POWER IN WATTS
1100
1000
900
800
700
600
500
400
300
200
100
0
100
200
300
400
500
600
700
800
SPEED IN R.P.M.
TECHNOLOGY BRANCH
HARVARD COOPERATIVE SOCIETY, CAMBRIDGE

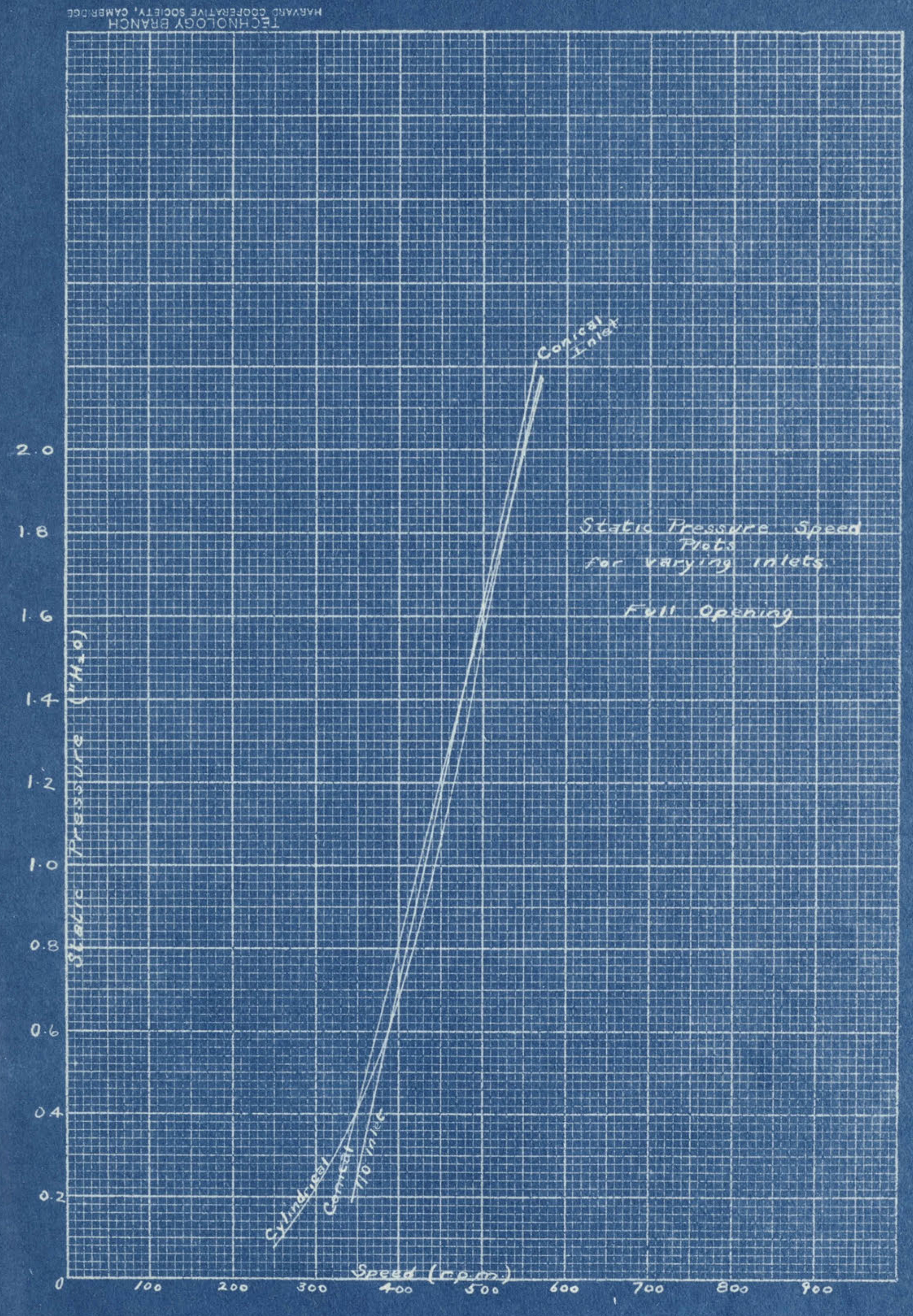

TECHNOLOGY BRANCH
HARVARD COOPERATIVE SOCIETY, CAMBRIDGE
ARMATURE COPPER LOSS IN WATTS
ARMATURE COPPER LOSSES OF MOTOR
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
0
10
20
30
40
50
60
70
80
90
ARMATURE CURRENT IN AMPS.

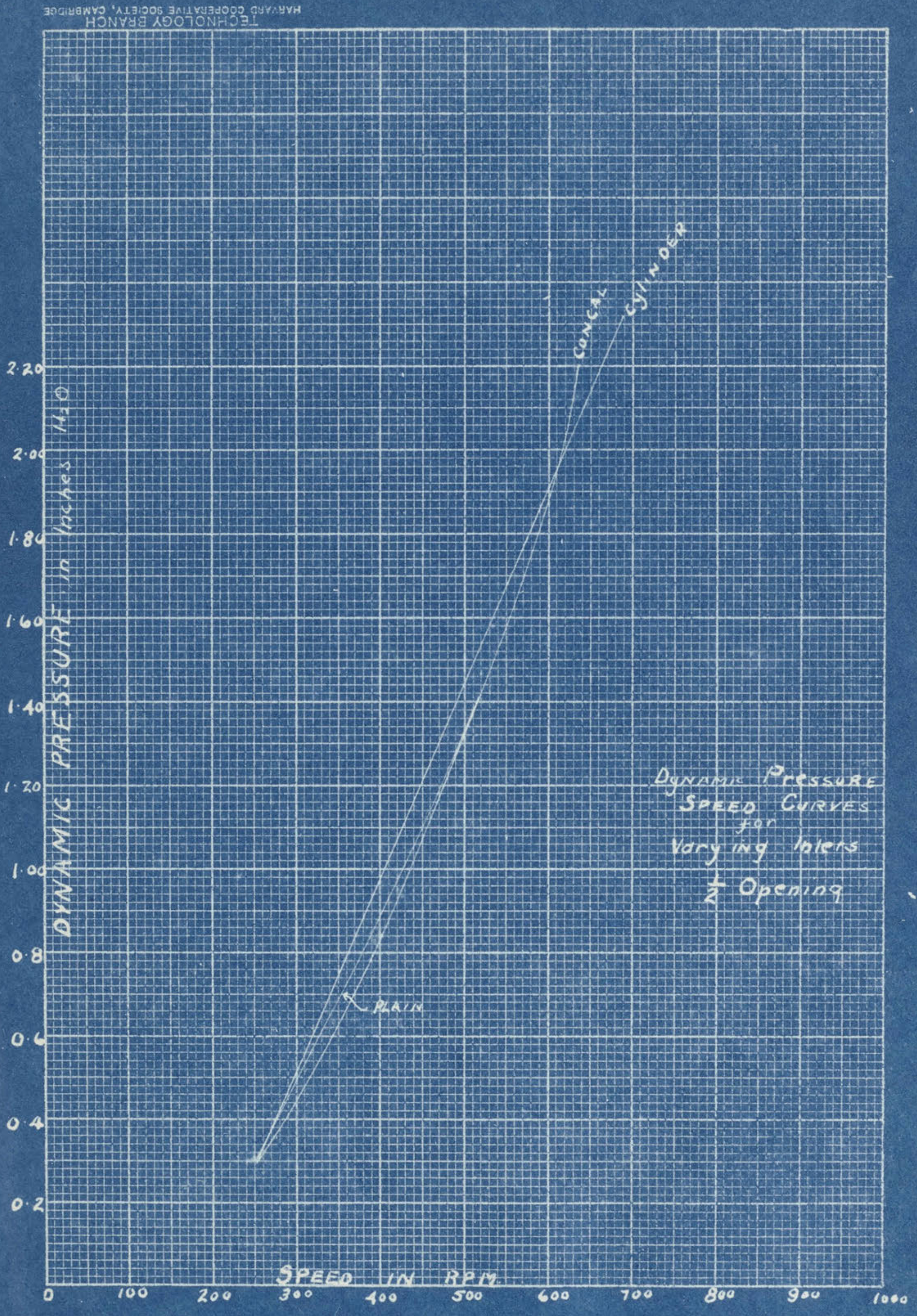

Capacity (cu-ft-min.)
0
1000
2000
3000
4000
5000
6000
7000
8000
Speed (r.p.m.)
100
200
300
400
500
600
Speed - Capacity Plots
for varying inlets.
Full Opening
no inlet
conical inlet
cylindrical inlet
TECHNOLOGY BRANCH
HARVARD COOPERATIVE SOCIETY, CAMBRIDGE



Speed Capacity Curves
for
Varying Inlets
½ Opening
Conical
Plain
Cylindrical
Cubic Feet per Minute
1000
2000
3000
4000
5000
6000
7000
8000
0
100
200
300
400
500
600
700
Revolutions per Minute
Technology Branch
Harvard Cooperative Society, Cambridge


TECHNOLOGY BRANCH
HARVARD COOPERATIVE SOCIETY, CAMBRIDGE
Static Pressure in Inches H2O
Speed in RPM
0
100
200
300
400
500
600
700
800
900
1000
0.2
0.4
0.6
0.8
1.00
1.20
1.40
1.60
1.80
2.00
2.20
2.40
2.60
Plain
Conical
Cylindrical
Static Pressure
Speed Curves
for
Varying Inlets
1/2 Opening


TECHNOLOGY BRANCH
HARVARD COOPERATIVE SOCIETY, CAMBRIDGE
Conical Inlet
Static Pressure Speed Plots for varying inlets
Full Opening
Static Pressure ("H_2O)
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
0
Cylindrical
Conical
1/10 Inlet
Speed (r.p.m.)
100
200
300
400
500
600
700
800
900


TECHNOLOGY BRANCH
HARVARD COOPERATIVE SOCIETY, CAMBRIDGE
DYNAMIC PRESSURE in Inches H_2O
SPEED IN RPM
CONICAL
CYLINDER
PLAIN
Dynamic Pressure
Speed Curves
for
Varying Inlets
$\frac{1}{2}$ Opening
2.20
2.00
1.80
1.60
1.40
1.20
1.00
0.8
0.6
0.4
0.2
0
100
200
300
400
500
600
700
800
900
1000

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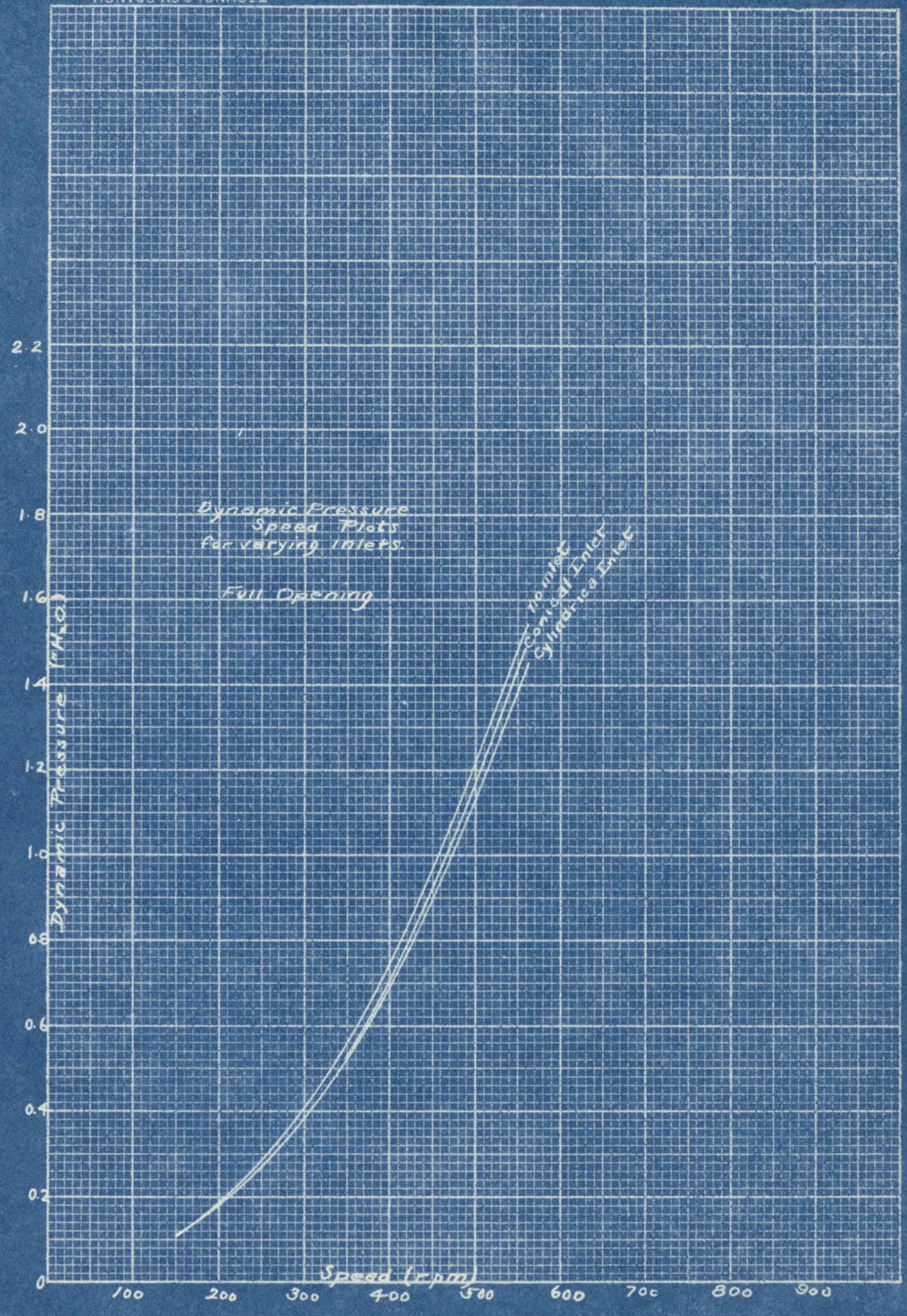
Dynamic Pressure
Speed Plots
for varying inlets.
Full Opening
No inlet
Conical Inlet
Cylindrical Inlet
Dynamic Pressure ("H2O)
Speed (rpm)
0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
100
200
300
400
500
600
700
800
900

Summary of Current Literature on "The Development and Characteristics of Centrifugal Fans"

"Forced and Induced Fraft with Mechanical Stokers"

By Mr. H.F. Hagen, Research Engineer, B.F. Sturtevant Co.

Mr. Hagen is a Stevans graduate and has been actively engaged in fan design work for over twenty years.

"The advent of the underfeed stoker calling for large volumes at higher pressures than had been previously required of fans necessitated new designs. Previously designers limited by the R.P.M. of the steam engine driving the fan, strove to secure as high a pressure as possible for a given blade tip speed, great stress being laid on the manometric efficiency. At present however, the case is just the reverse, the object being to obtain the highest practicle R.P.M. for the pressures and volumes desired. Fans with either forward or backward curved blades will not operate successfully in parellel, hence these fans should never be used for forced draft work, the best fans to use for this type of work are fans with steel plate paddle wheel characteristics. The effect of the various types of blades are shown on plots obtained as a result of tests by Mr. Hogen. The full backward curved fan is also suitable for stoker work and is practically universally used for such service, it meets all the requirements, reliability, perfect parellel operation, pressure reserve, high speed and efficiency. The worst sort of ducts to sue for fan work are those of concrete due to leakage of air thru the concrete. Tests have shown that air under a pressure of six to seven inches of water, will leak thru a concrete wall six inches thick at the rate of 38 percent of the total sent thru.

Where both forced and induced draft are used the induced should be only called upon to overcome the resistance to the flow of gases caused by the heat absorbing surfaces of boiler and economizer, and the necessary flues and further to furnish a slight draft in the firebox. A forced draft fan is not suited for the type of work as the resistances are practically constant. The dirt in the gases and the high speed of the fan do not go well together. Undoubtedly the paddle wheel fan of proper efficient proportions is by far the most suitable of all. Its speed is so low that an accumulation of duston the wheel does not cause vibration. The passages between the blades are large and any depos it will be so thin comparatively that there will be little or no loss in capacity or efficiency. This loss due to deposits is very serious in multibladed fans and necessitates frequent cleaning. To engineers who insist upon a direct connection however, must be sold the multibladed fans. It frequently happens that the forwardly curved blade fan is the only one to be used for an induced job. With high boiler ratings the velocities and corresponding draft losses are high. At the same time temperatures may run up to 600° or 700° at the fan, expecially when the boilers need cleaning. The elastic limit of steel decreases markedly at these temperature. The pressures desired may run as high as six inches which is in necessary speed equivilent to over twelve inches with cold air. Simply from strength considerations alone we are compelled at times to select the fan which will give the highest pressure for the lowest tip speed. So that for many of the ultra-modern power stations (December 1921) the forwardly curved blade fan has been selected for the induced draft. It is not generally appreciated that it costs more in power to create a given draft from the hot than from the cold end, although handling in each case the same weight of gases. Hence it is desirable to have the forced draft fan do all the work possible."

Some Developments in Centrifugal Fan Design

This article was written in May 1921 by Messrs. Bailey and Criqui. Mr. Bailey is the chief draftsman of the Buffalo Forge Co., and Mr. A.A. Criqui is a mechanical engineer. The following interesting points are brought out:

"The pressure curve of a radial blade fan rises continually with a decrease in capacity, and the power curve increases constantly with increasing capacity. The pressure continues to rise with the radial blade fan clear up to no load, while with the forward curve blade fan the pressure stops rising when a certain point is reached and with further decrease in capacity shows instead a slight drop in pressure. While with the radial blade fan the power increases constantly with the quantity of air, the power required for the forward curved blade fan increases at a much more rapid rate. This combination in the forward curved blade fan, of the pressure curve which droops or even only flattens with decreasing capacity within the working range of the fan, with the power curve which has a more abrupt rise as the load increases is sometimes the cause of serious trouble in fan installations. The flat portion of the pressure curve makes the fan very sensitive to resistance variations and if used at a capacity corresponding to this portion of the curve may make the fan run over or under the estimated capacity. This is particularly so if the friction in the system is slightly different from that as calculated, or if an existing duct system has been changed consquently changing the resistance. With a directly connected unit when a fan which has an abruptly rising power curve, runs above the estimated capacity there is a decided danger in over loading the motor. With a motor driven forward curved blade fan run at the critical speed point in its range, it is necessary to supply an excess of motor power with the probability of running the motor at reduced capacity and thereforereduced efficiency, to guard against any possibility of the fan being overloaded. This represents a perpetual insurance payment to protect the motor when using a forward curved blade fan under these conditions. The old radial blade fan was more self adjusting to these conditions and would vary but slightly from the desired capacity and power when the resistance proved different from that estimated. The forward curved blade fan is essentially a velocity fan. Air leaves the periphery of the wheel with but little static but with a high velocity pressure. It is the function of the fan housing to transform the velocity pressure of this air into static pressure. Within limits the larger the scroll or cone effect of the housing the greater the transformation and the higher the efficiency of the fan, but the greater the transformation the greater the hump in the pressure curve which means a more pronounced droop as the fan approaches no load. As has been stated before a droop or even a flat spot in the pressure curve has decided disadvantages. The design of the housing of the forward curved blade fan is therefore a compromise between efficiency and reliability of operation. The most desirable combination of fan characteristics would be to retain the good features of the old radial blade fan, that is the rising pressure curve from free delivery to no load with a moderately increasing power curve and at the same time to have the high capacity range, insuring a small housing, together with the higher efficiency of the forward curved blade fan. During the past few years a new type of multiblade fan has been developed which combines these good features. This new multi-blade fan has blades which curve forward at the heel to meet the incoming air, but curve backward at the tip to discharge the air. Thus while the blade of the forward curve fan is a portion of the surface of one cone or one cylinder, the blade of the backward curved blade fan is formed from the surfaces of two cylinders or two cones."

Pitot Tubes For Gas Measurement

This article is of value to anyone having to do with fan work as the question frequently arises as to how accurate readings obtained by the Pitot tube may be. Mr. C.W. Rowse, the author of the article, is a member of the American Society of Mechanical Engineers and the following conclusions were obtained by him as a result of tests at the University of Wisconson in 1913.

a. The Pitot tube as a means of measuring gases is reliable within appromimately one percent when the static pressure is correctly obtained and when all readings are taken with a sufficient degree of refinement; in order to obtain this degree of refinement and accuracy the pitot tube should be preceeded by a length of pipe 20 to 38 times the pipe diameter in order to make the flow of gas as nearly uniform across the section of pipe as possible.

b. The most reliable and accurate means of obtaining the static pressure is the piezometer or its equivilent, the resutls of 138 separate tests using the piezometer, static pressures agreeing with the Thomas meter within one-third of one percent were obtained. These results show beyond any doubt that the static pressure is constant across any section of pipe in which gas is flowing at a uniform rate.

c. Of the methods of obtaining the static pressure by pitot tube itself, the most reliable and accurate is by means of a very small hole in a perfectly smooth surface. The long slots for obtaining the static pressure are not reliable and give results which are in error from 3.5 to ten percent. The length of the slots or the thickness of the outer tube do not appear to affect the accuracy of the tube.

d. The tube should point directly upstream, the effect of allowing the tube to point at an angle with the direction of flow is seen when it is noted that if the tube is off 20° in one direction an error of 85 percent in the velicity head is introduced.

From Minutes of Proceedings, Institute of Civil Engineers.

"The Desing and Testing Of Centrifugal Fans" by Hammersley Heenan, M. Inst. C.E. and William Gilbert, Wh. Sc., Assoc. M. Inst. C.E. Paper No. 2857, December 17, 1895.

The object of this paper was:

"(1) To determine the best shape of the fan blade and fan casing, in order to secure a minimum expenditure of power when producing any given output of air, i.e. to find the best type of fan.

"(2) The standard type being selected to obtain the data whereby the proper diameter of the standard fan and its most economical speed could be determined for any required output of air at a given pressure.

"The experiments proved that a fan with a few simple blades fives the best results, provided the form of blade and dimensions of the casing are designed to suit the kind of work required. Fans of more complex design have too large an internal resistance to give the highest mechanical efficiency, although they may have to be used if high pressures are essential.

In the discussion of the paper, there were brought out the following facts:

(1)"That theory must be well supplemented by testing of each design in the development of fans.

(2) "That the most important thing in designing a fan was not so much the form of the blade, but that the air should enter the

fan with as little shock as possible, and be reduced to the lowest velocity upon escaping."

(3) Capacity varied as the speed of the fan, the velocity head as the square of the speed, and the horsepower output as the cube of the speed.

(4) To get larger capacity when speed and space were limited, use radial blades and increase speed between them.

(5) That it might be possible to sue the high discharge velocity of a small quantity of air to impart a lower velocity to a large quantity of air by the injector principle, thus utilizing the momentum of the air leaving a fan.

Latest Developments in Fan Design

Up till twenty years ago fans were designed by experiment, that is, a certain shaped blade was tried, then another, and so on until the best type was obtained. This was all right with the straight blade fan,,but when the curved blade fan is treated in this manner the combinations are infinite and consequently fan design has to be attacked from a different angle. If air is treated as a perfect fluid incompressible and without any or with a constant viscosity the path of the air thru a fan casing can be reasonably plotted. The next thing is to find a surface which will offer the least resistance to the flow of air, and this surface takes the shape ~~of~~ similar to the wings used on airplanes. With these surfaces as blades a fan has been constructed which has actually given an efficiency of 82.5 percent. This is a remarkable performance. From a manufacturing standpoint these wing shaped blades are too expensive to make so the wing shape is approximated in practice by its center line and with this arrangement an efficiency of 80 percent is obtained. This fan is housed in a special type of casing. Previously it was stated that the usual casings are made to approximate the Archimedian spiral. In this new type of casing the scroll follows a logerithmic spiral. The basis for this is as follows:

The velocity of the air leaving the tips of the fan blades is made up of two components. A tangential and a radial. The radial component is some function of the velocity, while the tangential is also another function of the radius. Hence the resultant must be a function of the radius , hence if we desire to keep conditions at the casing the same as at the blade tips the casing should follow a logerithmic spiral.

Bibliography of Centrifugal Fans

Very little worth while has been written in this country in regards to fans and fan design, which is worth while from a scientific standpoint, most of the good work being unknown to the general public. This is probably due to the fact that the fan business in this country is so fiercly competitive, and fan companies keep their work a secret. However the following sources of information may prove of interest.

"Neue Theorie and Berechnung der Krieselradär " by Dr. Hand Lorenz. This is an excellent book covering fan design.

"Die Gebläse" by Von Ohring treats of the historical part of the subject.

"Centrifugal Fans" by Kinealey

"Fans" by Snell

"Fan Engineering" by Buffalo Forge Co.

"Fan Testing" by American Blower Co.

"Fan Design" Mark's Handbook

"Mechanical Draft" Gebhardt

"Heating & Ventilation" Carpenter

"Heating & Ventilation" F.B. Sturtevant Co.

"Transactions of American Society of Heating & Ventilating Engineers"

"Coal Age" This deals with fans used in mine work.

"Transactions of the American Society of Mechanical Engineers"

"Transactions of the British Institute of Mechanical Engineers"

"Proceedings of British Institute of Civil Engineers"

"Power"

"Heating & Ventilation" by Prof. Miller

"Lectures of Prof. E.B. Wilson on Aero-dynamics"

"Engineering" London