An Experimental Study on the Performances of Drum Type Centrifugal Ventilators of Varying Axial Lengths

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By

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(Received June 28, 1924)

ABSTRACT.

The forms of the runner of the multiblade drum type centrifugal ventilators, of which the "Sirocco" is the best known, and most popularly adopted in our collieries seem somewhat disadvantageous for a uniform flow of air. The air axially sucked in must suddenly change its direction radially in order uniformly to go out from the periphery, but this is impossible unless the air has no moving inertia.

Thus more or less eddy formations due to this cause are inevitable with the runner, and the writer observed that the eddy phenomena decreased by shortening the axial length of the runner. But, then, the ratio, between the amount of frictional surfaces for the air, to the cross rectional area in the casing, would increase, thus causing more frictional loss of energy, if assumed that the quantity of air treated is proportional to the axial length of the fan.

The purpose of this experiment is to investigate how the fan performrnces change with fans of varying axial lengths, owing to these two opposite influences, keeping other conditions as unchanged as possible.

INTRODUCTION.

Of this type of fan of single inlet, it has been suggested,[1] when driven without casing, that at the back end the air rushes out while at the inlet end the air is sucked in as shown in fig. 1. I have tried to observe the phenomena with a fan of this type which is installed at the Central Laboratory of the Kyoto Imperial University, and have learnt about these unfavorable eddy formations, by bringing a light paper strip close to the periphery of the runner.

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1) Charles H. Innes, "The Fan". 259 (1919)
T. Suhara, "Photographic Study on the flow of air through a centrifugal fan". XXIV, 66 (1922).

Referring to fig. 1, the air rushes out considerably at the back end and the velocity seems gradually to diminish to zero up to the middle section, while at the inlet end the paper strip is strongly pushed to the periphery and it appears as if the air were sucked into the fan, and this gradually diminishes up to the middle section.

Fig. 1.

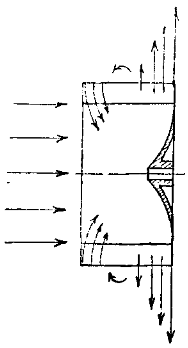
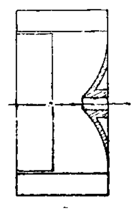


Fig. 2.



These eddy phenomena were always observed whatever the number of revolutions. I also tried to observe them, by the method above descrived, at various reduced axial lengths of the fan. Strapping a thick paper ribbon around the inner periphery of the runner as shown in fig. 2, and increasing the width of the ribbon stepwisely, thus approximately obtaining several fans of different axial lengths, other consitions remaining the same.

Then I observed that the unfavorable phenomena decreased more and more with the fans of shorter axial lengths, and at last the eddy formations apparently ceased and the air was blown out of all parts of the periphery.

Even with fans with casing, I believe that such like phenomena occur.

If it is assumed that the air quantity treated is proportional to the effective axial length of the runner, and consequently to the width of the casing, the velocity of the air at the outlet of the casing should theoretically be a constant, independent of the length of the runner and the width of the casing, other conditions remaining the same. The area of the surfaces along the current, and the cross sectional area in the casing both decrease by shortening the axial length of the fan, but the former does not so rapidly decrease as the latter.

Hence, from the standpoint of the energy losses of the air by surface friction, the fan of longer axial length would be better.

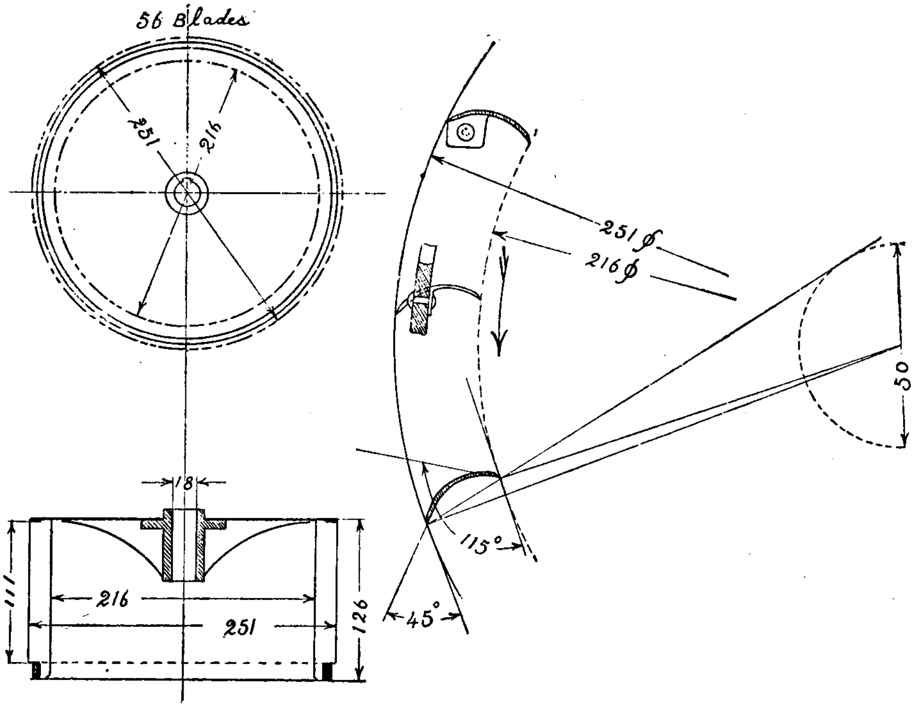
Interested in the question how the fan performances change, owing to these two opposite influences, I have carried out the following experiments.

As the form and number of blades, and the diameter of a fan could not be easily changed, I had to be satisfied with a given fan for my experiments.

1. EXPERIMENTAL APPARATUS AND MEASURING INSTRUMENTS.

§ <u>Fan.</u> "Sirocco" type as shown in fig. 3.

Fig. 3.



Runner :	Outer diameter	0.251 m
	Inner diameter	0.216 m
	Axial length (original)	0.111 m
	Number of blade	56

Thickness of blade 0.001 m

Exit angle of blade as shown in the figure 45°

Inlet angle of blade as shown in the figure 115°

(The above is a model of a Sirocco runner of ordinary proportions.)

Casing: Rectangular cross section.

Profile as shown in fig. 4

Instead of thick paper ribbons strapped around the inner periphery of the runner to change the axial length of the latter as mentioned in the preliminary observations, two short pipes of sheet metal of different diame-

Fig. 4.

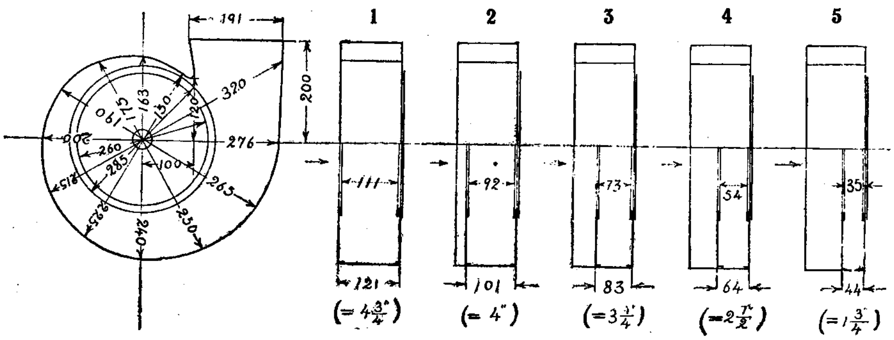
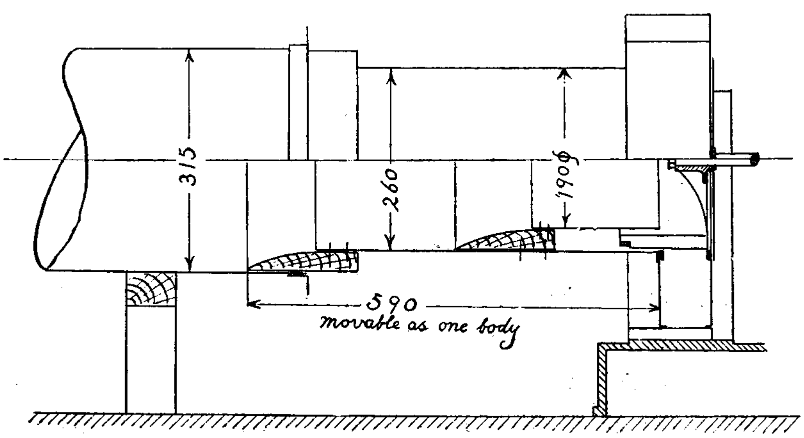


Fig. 5.



ters, corresponding to the outer and inner diameters of the runner with a certain amount of clearance, were secured to one side plate of the casing, which is arranged to be movable as ahown in figs. 4 and 5.

By this apparatus the axial width of the casing is also changed at the same time with that of the runner.

As it was difficult to attach evasée of proper side inclinations to the casings of variable widths, the following experiments were proceeded without evasée so the fan outlets were always opened to the atmosphere.

§ Suction pipe. Circular section as shown in fig. 6.

Inner diameter	0.315 m
Length over all	5.73 m
Length from pipe inlet to the measuring position	3.48 m

The inlet end was made very much like a camera blind and any desired opening is obtained which serve to regulate the air quantity as well as the fan resistance. Without the wire mesh shown in the figure, the meniscus in the manometer oscillated so sensibly that observation was almost impossible, while once the mesh was inserted the oscillation practically ceased so long as the motor speed was constant. Due to this mesh,

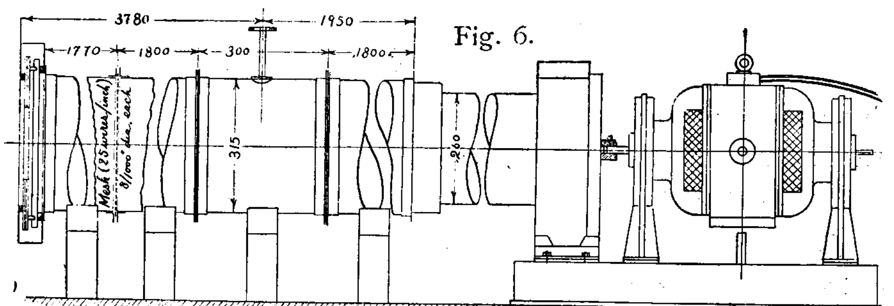


Fig. 6.

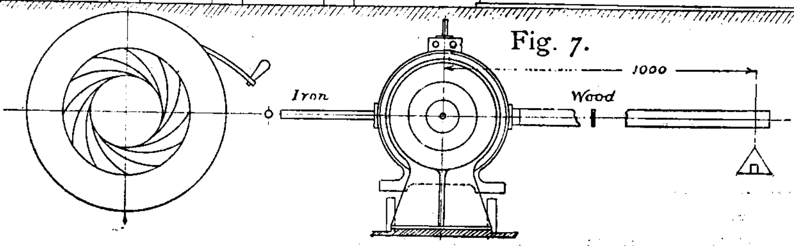


Fig. 7.

the resistance in the suction pipe was, of course, somewhat increased.

§ Driving motor. Direct Current electric motor as shown in fig. 7.

Voltage	110
Rated horsed power	2
Rated speed	1450 rev/mn

The motor is one directly coupled to the fan (the latter had been originally fitted with a suitable expanding chimney and a delivery chamber) by the maker, and made as a cradle dynamometer. The stator of the motor is supported by ball bearings.

§ Manometer.

Type———R. Fuess Micromanometer,[2] minimum reading $\frac{1}{100}$ mm. of alcohol column.

§ Measuring tube.

Type———Prandtl[3] as ahown in fig. 8.

§ Revolution counter.

Type———Hasler automatic speed indicator.

§ Psychrometer.

Type———Schubert's " Schleuderpsychrometer ".

2. PROCESS OF EXPERIMENT.

In order to carry out the whole series of experiments, the air volume was measured in the suction pipe and the pressure, which the fan generates, was measured also in the suction pipe, for convenience sake.

Regarding the speed of the fan, I have treated it as constant as possible at 1450 rev/mn throughout all experiments, which is the rated speed of the driving motor. As the current for the motor is supplied from the distributing station at the University, the voltage of the supplying main, and consequently the speed of the motor fluctuate within a considerable range, even when the operating condition of the fan is kept the same.

2) Ostertag, " Kolben-und Turbokompressoren " 217 (1919)
3) Gramberg, " Technische Messungen ", 111 (1923)

So, it was necessary to correct all readings, in the following calculations, by using the general rules[4] that :

when a fan is operating at the same restriction,

i) the volume dilivered will vary directly with the revolution per minute N; and

ii) the various pressures will vary with N^2; and

iii) the air horse power will vary with N^3. (Accordingly, assuming that the mechanical efficiency remains the same, the shaft horse power will also vary with N^3.)

The rheostat was regulated, when the speed fluctuation had surpassed ± 50 rev/mn.

The air volume was measured with the measuring tube inserted into the suction pipe. Equally dividing the cross sectional area at the

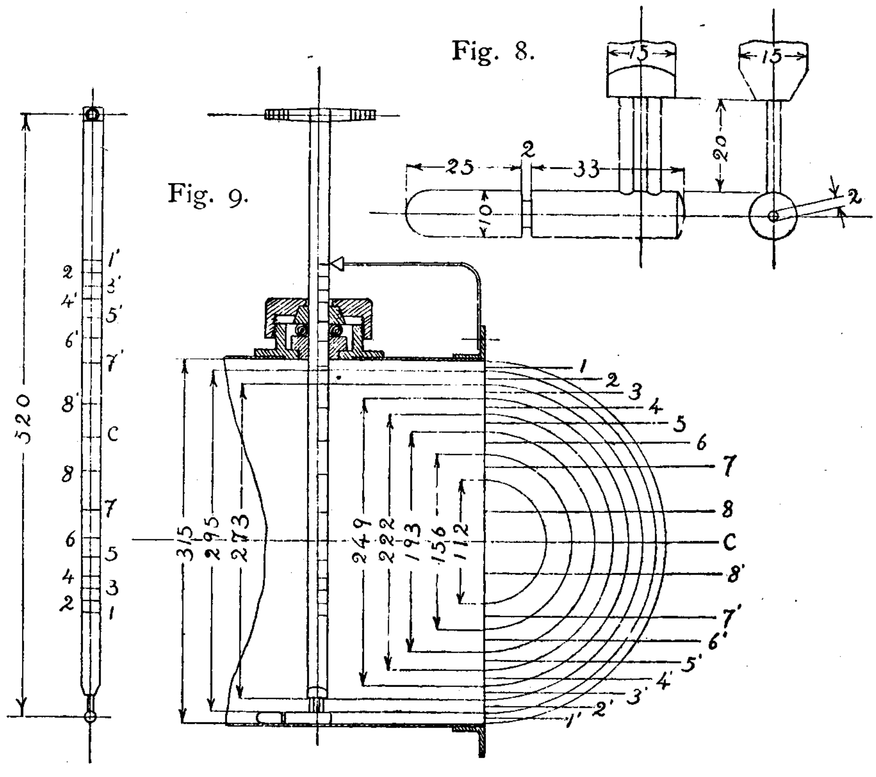


Fig. 8.

Fig. 9.

4) Lionel S. Marks, "Mech. Eng. Handbook", 1545 (1916)

measuring section by 8 concentric areas and in each ring the velocity heads were observed at two points which lie symmetrically, to the center, on a diameter as shown in fig. 9.

To take the center velocity into account, velocity heads at three points were observed, in the center circle. Thus, referring to the figure, if the letters 1, 2, 3, 4,......C......4′, 3′, 2′, 1′ represent the velocities of the air at the points above mentioned, then

$$\text{Volume} = \frac{\pi d^2}{4\times 8}\left(\frac{1+1'}{2}\right)+\frac{\pi d^2}{4\times 8}\left(\frac{2+2'}{2}\right)+\cdots\cdots\cdots\cdots$$

$$+\frac{\pi d^2}{4\times 8}\left(\frac{7+7'}{2}\right)+\frac{\pi d^2}{4\times 8}\left(\frac{8+C+8'}{3}\right)$$

$$=\frac{\pi d^2}{4}\left(\frac{1+2+3+\cdots\cdots+7+7'+6'+\cdots\cdots+1'}{2\times 8}+\frac{8+C+8'}{3\times 8}\right)\cdots\cdots\cdots\cdots(1)$$

where d is the diameter of the suction pipe at the measuring section = 0.315 m. By the way, the term in the bracket represent the mean volocity.

The suction inlet opening was regulated at seven stages, namely, 0, 7, 11, 14, 17, 20, and 30 cm diameters which were determined so in the following way; driving the fan, the effective axial length of the runner of which is 0.111 m$\left(=4\frac{3''}{8}\right)$ at the speed above mentioned, and regulating the inlet opening at seven stages with trials, so as to approximately obtain equal amounts of increase of air volume. To measure the volume then, an anemometer was used at the discharge end of the fan.

The diameters of the suction inlet opening above mentioned were kept the same at other axial lengths of the fan.

To avoid confusion, let them be symbolized in the experiments as follows;

Suction inlet diametres	cm 0	cm 7	cm 11	cm 14	cm 17	cm 20	cm 30
Symbols	0	I	II	III	IV	V	VI

and for the axial lengths of the runner, and consequently the widths of the casing (refer to fig. 4.), let them be denoted as follows;

Width of casing outlet in	inch	$4\frac{3}{4}$	4	$3\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{3}{4}$
	cm	12.1	10.1	8.3	6.4	4.4
effective axial length of runner in	inch	$4\frac{3}{8}$	$3\frac{5}{8}$	$2\frac{7}{8}$	$2\frac{1}{8}$	$1\frac{3}{8}$
	cm	11.1	9.2	7.3	5.4	3.5
Symbols.		**1**	**2**	**3**	**4**	**5**

Thus, for example, by denoting the experiment with **4** at II, the effective axial length of the runner, casing outlet width, and the suction inlet opening correspond to 5.4 cm, 6.4 cm, and 11 cm respectively.

Throughout my experiments, the temperature of the air in the suction pipe as well as that at delivery end were practically equal to the atmospheric temperature; the differences between these three were so small that I could not detect the differences, with ordinary Centigrade thermometers, so in all cases in my experiments these three temperatures were assumed to be the same.

As the density of the air, though in small range, varies with the sucsion resistance, and in pretty large range, with barometric conditions, atmospheric temperature and humidity, the volume obtained were reduced, in drawing the characteristic curves, to a standard condition[5] of 735.5 mm Hg and 15°C for convenience in comparing the observed results.

3. CALCULATION EXAMPLES.

The formulae, employed for the calculations in my experiments, are summarized as follows, with corresponding numerical examples for the case of **1** at II.

§ BAROMETER CORRECTION.

$$b = b'(1-0.000162 \times t)^{6)} \quad \ldots\ldots(2)$$

where b = corrected barometric height in mm Hg.

5) Gramberg, "Tech. Messungen" 112

6) "Hütte" 20 Auf, I, 313

$b' =$ barometer reading (the scale is of brass)

$t =$ temperature in °C

Example 1. $t = 6°C$ $b' = 765$

$\therefore$ $b = 764.3$ mm Hg

absolute pressure $= 13.596 \times 764.3 = 10391.42$ kg/m²

or 10391.42 mmAq ($\because$ 1 mmAq $=$ 1kg/m²)

Note Second decimal figure corresponds to the minimum reading of the micromanometer.

§ HUMIDITY IN ATMOSPHERE.

$$\varphi_0 h_0 = h_f - kb(t_0 - t_f)^{7)} \qquad \text{...(3)}$$

where $\varphi_0 =$ relative humidity.

$t_0 =$ temperature in °C. given by dry thermometer.

$t_f =$ ditto given by wet thermometer.

$h_0 =$ water vapour tension at t_0 in mm Hg[8]

$h_f =$ ditto at t_f

$b =$ as before.

$k =$ a constant, taking 0.0007 following the consideration in Watson's text,[7]

Example 2. $t_0 = 6°C$ $t_f = 4°C$

$$\varphi_0 = \frac{6.1 - 0.0007 \times 764.3(6-4)}{7.0} = 0.72$$

§ DENSITY OF MANOMETER LIQUID.

Alcohol is used as manometer liquid, and its density was measured with a hydrometer as 0.8265 at 15°C, and the densities at other temperatures may be given by the formula below;

$$\gamma_t = \frac{1}{[1 + \alpha(t - 15)]}\gamma_{15} \qquad \text{...................................(4)}$$

where $t =$ temperature in °C.

7) Watson, "Practical Physics," 248 (1917)

8) "Hütte" 20 Auf. I. 322

γ_t = density of the alcohol at t°C. in g/cm^3

γ_{15} = ditto at 15° C = 0.8265

a = expansion coefficient of alcohol, taking 0.0011[9)]

Example 3. $t = 6$°C

$\gamma_t = 0.8265/[1+0.0011(6-15)] = 0.835$

§ PRESSURE DEPRESSION IN SUCTION PIPE. D.

Connecting the side opening of the measuring tube to the manometer, the static depressicn is obtained. As the difference in reading was not noticeable by placing the tube at the various positions in the measuring section, the tube was always inserted at the center of the pipe for the purpose.

Example 4. Manometer inclination = 15°

density of alcohol = 0.835 at 6°C reading = 164mm

speed of the fan = 1450 rev/mn (exactly)

static depression D = 164 × sin 15° × 0.835 = 35.44 mmAq or kg/m^2

§ HUMIDITY OF AIR IN SUCTION PIPE.

Let φ_s = relative humidity of air in suction pipe.

φ_0 = ditto of atmospheric air as before,

h_0 = water vapour tension at atmospheric temperature in kg/m^2,

h_s = ditto at the temperature of air in suction pipe,

p_0 = atmospheric pressure in kg/m^2,

p_s = absolute static pressure in suction pipe at measuring point in kg/m^2,

then

$$\varphi_s \frac{h_s}{p_s} = \varphi_0 \frac{h_0}{p_0} \quad {}^{10)}$$

but by the assumptions regarding the temperatures above described, $h_0 = h$

9) "Hütte" 20 Auf. I. 297

10) "Hütte" 20 Auf. I. 322

and the formula becomes much simpler so that,

$$\varphi_s = \varphi_0 \frac{p_s}{p_0} \quad \text{..........(5)}$$

Example 5. $\varphi_0 = 0.72$ $p_0 = 10391.42 \text{ kg/m}^2$

$p_s = 10391.42 - 35.44 = 10355.98 \text{ kg/m}^2$

$\varphi_s = 0.72(10355.98/10391.42) = 0.72$

As seen in this example, the influence of the pressure drop on the humidity is negligible in my cases.

§ AIR CONSTANTS.

Let R_0 = air constant for atmospheric air,

R = air constant for dry air = 29.27

p_0, h_0 and φ_0 = as before

then,

$$R_0 = \frac{R}{1 - 0.377\ \varphi_0 \frac{h_0}{p_0}} \text{[11]} \quad \text{..........(6)}$$

Example 6. $\varphi_0 = 0.72$ $p_0 = 10391.42 \text{ kg/m}^2$

$h_0 = 13.596 \times 7 (= \text{tension at } 6^\circ\text{C in mm Hg})$

$= 95{,}2 \text{ kg/m}^2$

$$R_0 = \frac{29.27}{1 - 0.377 \times 0.72 \frac{95.2}{10391.42}} = 29.34$$

Similarly let R_s = air constant for the air in suction pipe,

φ_s, h_s. p_s and R = as before, then

$$R_s = \frac{R}{1 - 0.377\ \varphi_s \frac{h_s}{p_s}} \quad \text{..........(7)}$$

Example 7. $\varphi_s = 0.72$ $p_s = 10355.98 \text{ kg/m}^2$ $h_s = h_0 = 95.2 \text{ kg/m}^2$

$$R_s = \frac{29.27}{1 - 0.377 \times 0.72 \frac{95.2}{10355.98}} = 29.34$$

As in these examples, R_0 is practically equal to R_s throughout my experiments.

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11) Ostertag, "Kolben-und Turbokompressoren" 10 (1919)

§ DENSITY OF AIR.

Atmospheric air. $$\gamma_0 = \frac{p_0}{R_0 T_0} \quad \text{...(8)}$$

where γ_0 = specific weight of air in kg/m³

R_0 and p_0 = as before

T_0 = absolute temperature of air in °C.

Example 8. $T_0 = 273+6 = 279°C$ $p_0 = 10391.42$ kg/m²

$R_0 = 29.34$ then $\gamma_0 = 1.269$ kg/m³

Air in suction pipe. $$\gamma_s = \frac{p_s}{R_s T_s} \quad \text{..................................(9)}$$

where γ_s = specific weight of air in suction pipe in kg/m³

R_s and p_s = as before.

T_s = absolute temperature °C in suction pipe.

Example 9. $T_s = T_0 = 279°C$ $p_s = 10355.98$ kg/m³

$R_s = R_0 = 29.34$, then $\gamma_s = 1.265$ kg/m³.

Air at delivery end. The density of air at the delivery end is assumed to be equal to that of atmospheric air.

§ AIR VELOCITY IN SUCTION PIPE.

Let h_v = velocity head in mmAq. v_s = air velocity in m/s

g = gravity acceleration = 9.80 m/s²

γ_s = density of air in kg/m³

ξ = measuring tube constant, always assuming $\xi = 1$,

then, modifying the general formula $h_v = \xi\gamma_s v_s^2/2g$, we obtain

$$v_s = \sqrt{\frac{2g}{\gamma_s}}\sqrt{h_v} \quad \text{..................................(10)}$$

Example 10. At the pipe center, the reading of velocity head was 26 mm alcohol, inclination of the manometer = 1°10′, density of alcohol = 0.835 g/cm³, $\gamma_s = 1.265$ kg/m³, speed of the fan = 1440 rev/mn.

then, $h_v = 26 \times \sin 1°10 \times 0.835 = 0.435$ mmAq.

$$h_v \text{ reduced to 1450 rev/mn} = 0.435 \times \frac{1450^2}{1440^2} = 0.441$$

$$v_s = \sqrt{\frac{2\times 9.80}{1.265}} \cdot \sqrt{0.441} = 2.62 \text{ m/s}$$

§ VOLUME REDUCED TO (735.5 mmHg, 15°C)

$$V_r = V_s \frac{273+15}{273+t} \cdot \frac{p_s}{735.5\times 13.596} \quad ^{12)} \quad \ldots\ldots(11)$$

where V_r = reduced volume

V_s = volume measured

t = air temperature in suction pipe in °C.

p_s = as before.

Example 11. $V_s = 0.1948\ m^3/s$ (refer to Table 4_1)

$t = 6°C$

$\therefore\ V_r = 0.208\ m^3/s$

§ THEORETICAL ISOTHERMAL POWER, E and E′.

$$\text{Power E in P. S.} = p_1 V_s ln\frac{p_2}{p_1}/75 \quad \ldots\ldots(12)$$

where p_1 = absolute suction pressure in kg/m^3

= absolute static suction pressure + suction velocity pressure.

p_2 = absolute delivery pressure in kg/m^3

= absolute static delivery pressure + delivery velocity pressure.

When the effective isothermal power E′ is required, p_2 is put equal to the absolute static delivery pressure,[13] that is, in my case, it is equal to the atmospheric pressure p_0, as the delivery velocity pressure may be assumed to be totally lost.

Example 12. Let $p_0 = 10391.42\ kg/m^2$, $p_s = 10355.98\ kg/m^2$, mean velocity in suction pipe = 2.44 m/s (refer to Table 4_1), outlet area of fan casing = 0.0231 m^2.

12) Gramberg, "Tech. messungen" 112

13) Gramberg, "Masch. untersuchungen" 471 (1921)

Then, suction velocity pressure $= \frac{(2.44)^2}{2 \times 9.80} \times 1.265 = 0.38$ kg/m².

$\therefore \quad p_1 = 10355.98 + 0.38 = 10356.36$ kg/m².

The volume delivered at discharge end

$= V_s \times \frac{\gamma_s}{\gamma_0} = 0.1948 \times 1.265/1{,}269 = 0.1942$ m³/s, so the

mean delivery volocity $= 0.1942/0.0231 = 8.41$ m/s, then,

delivery velocity pressure $= (8.41^2/2 \times 9.80) \times 1.269 = 4.58$ kg/m²

$\therefore \quad p_2 = 10391.42 + 4.58 = 10396.00$ kg/m²

or when the effective power is required,

$p_2 = 10391.42$ kg/m².

Total power E in P. S. $= 10356.36 \times 0.1948 \; ln \frac{10396.00}{10356.36} / 75 = 0.102$

Effective power E′ in P. S. $= 0.091$

§ THEORETICAL ADIABATIC POWER.

$$\text{Power in P. S.} = p_1 \; V_s \frac{x}{x-1} \left[\left(\frac{p^2}{p_1} \right)^{\frac{x-1}{x}} - 1 \right] / 75 \;\text{...........(13)}$$

where $x = \frac{C'_p + mC''_p}{C'_v + mC''_v}$ [14]

and C'_p = specific heat of air at constant pressure = 0.2375

C'_v = ditto at constant volume = 0.1685

C''_p = specidic heat of superheated steam at constant pressure = 0.4805

C''_v = ditto at constant volume = 0.3695

and $m = 0.623 \, p_d/p_l$

where p_d = vapour tention at a temperature in kg/m²

p_l = (partial) pressure of dry air in kg/m² at the same temperature.

Example 13. All data are as in the last example.

$p_l = p_s - p_d = p_s - 13.596 \times$ vapour tention at 6°C in mm Hg $\times$ relative humidity φ_s

14) Ihering, "Die Gebläse" 550 (1903)

$$= 10355.08 - 13.596 \times 7 \times 0.72 = 10355.98 - 68.5$$
$$= 10287.48 \text{ kg/m}^2$$
$$m = 0.623 \times 68.5/10287,48 = 0.00414$$
$$x = \frac{0.2375 + 0.00414 \times 0.4805}{0.1685 + 0.00414 \times 0.3695} = 1.409$$
$$\frac{x}{x-1} = 3.445 \qquad \frac{x-1}{x} = 0.2903$$
$$\text{Power in P. S.} = 10356.36 \times 0.1948$$
$$\times 3.445 \left[\left(\frac{10396}{10356.36} \right)^{0.2903} - 1 \right] / 75 = 0.102$$

As in these examples, the isothermal and adiabatic powers are practically equal to each other in such small ranges of pressure differences as in my cases, hence for simplicity, the theoritical horse powers are calculated always as isothermal.

§ PRESSURE DIFFERENCES BETWEEN SUCTION AND DELIVERY, P and P′.

Although the pressure difference can be obtained as $(p_2 - p_1)$ using the above described figures, but to avoid such large numerical values as (10396.00 − 10356.36) it may be more convenient to calculate as follows:

As p_2 = Atmospheric pressure + static delivery pressure + delivery velocity pressure, and

p_1 = atmospheric pressure − static suction depression + suction velocity pressure.

$p_2 - p_1$ = static delivery pressure + delivery velocity pressure + static suction depression − suction velocity pressure.

Example 14.

As static delivery pressure (over atmospheric) has assumed to be zero, the total pressure difference $P = p_2 - p_1 = 4.58 + 35.44 - 0.38 = 39.64$ kg/m² or mmAq, and the effective pressure difference $P' = p_2 - p_1 = 35.44 - 0.38 = 35.06$ kg/m²

Note: If an ideal frictionless evasée were attached to the casing, it might be possible, (by reducing the inlet area of the suction pipe) to treat an equal amount of air which the fan without evasée has treated, against certain higher ventilating resistance (the driving power remaining the same). In such a condition the delivery velocity head which had been otherwise lost would converge towards null and serve to create more depression in the suction pipe if necessary; and in the ideal limit, the total pressure difference obtained as in the above example would all be utilized in the effective suction depression. Thus the total pressure difference is the ideal maximum possible limit of effective pressure difference with evasée.

§ THEORETICAL HEAD WHICH THE FAN GENERATES.

The theoretical maximum possible pressure difference between suction and delivery is obtain by the formula:

$$H = \frac{u^2}{g} + \frac{u}{g} C.\cot\beta \text{[15]} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (14)$$

which represents the best case and the formula is called that for "Normal" condition,

where $H =$ head the fan creates in *m* air column.
$u =$ outer peripheral velocity in m/s.
$C =$ radial exit velocity in m/s
$\beta =$ outer vane-angle as shown in fig. 3.
$g =$ gravity acceleration $= 9.80$ m/s².

Example 15. Suction air volume $= 0.1948$ m³/s (reduced to the fan speed of 1450 rev/mn),
outer diameter $= 0.251$ m,
$\beta = 45°$ $\therefore$ $\cot\beta = 1$,
number of blades $= 56$,
thickness of blades $= 0.001$ m,
axial length of blade $= 0.111$ m.

Then the net area of outflow at the outer periphery is

$$\pi \times 0.251 \times 0.111 \times [1 - 0.001 \times 56/(\pi \times 0.251 \times \sin 45°)]$$

15) Ostertag, "Kolben-und Turbokompressoren", 155

$$= 0.0876 \times 0.9 \text{ m}^2$$

and assuming the coefficient of contraction of flow to be 0.92 following Ostertag, and moreover assuming that it is constant at various axial lengths of blades, then the effective exit area of flow becomes,

$$0.0876 \times 0.9 \times 0.92 = 0.0725 \text{ m}$$

$$C = (\text{suction air volume} \times \gamma_s/\gamma_0) \div 0.0725$$

$$= (0.1948 \times 1.265/1.269) \div 0.0725 = 2.678 \text{ m/s}$$

Strictly speaking, the volume change must not be referred to γ_0, but to the density of air at the exit of the runner. But in my case, as the casing directly opens in the atmosphere, it may be allowed to do so by approximation.

$$u = (\pi \times 0.251 \times 1450) \div 60 = 19.06 \text{ m/s}$$

$$u^2/g = 37.07 \qquad u/g = 1.945$$

$$H = 37.07 + 1.945 \times 2.678 = 42.27 \text{ m air column.}$$

As the fan has created 42.27 m of head, treating the atmospheric air whose density is 1.269 kg/m^3; converting which to the pressure difference H_w, we obtain

$$H_w = 1.269 \times 42.27 = 53.64 \text{ kg/m}^2 \text{ or mmAq}$$

§ FUN INPUT POWER E_f AND ITS MEASUREMENT.

$$E_f = \text{Power in P. S.} = M_e\omega/75 \quad \ldots\ldots\ldots\ldots\ldots\ldots(15)$$

where M_e = moment for net output of motor in mkg.

ω = angular velocity in rad/s

As the fan is driven by a motor which was made as a cradle dynamometer as ahown in figs. 6 and 7,

$$M_e = M_d - (V + R_1 + R_2)$$[16)]

where V = moment for motor windage.

R_1 = moment for motor shaft friction.

R_2 = moment for the ball bearing friction of the stator.

M_d = moment measured in mkg.

As it is difficult to measure the value V, R_1, and R_2 individually,

16) Gramberg, "Tech. Messungen" 318.

the aggregate values was measured by driving the motor with no load at 1450 rev/mn, that is, without the fan, and the value of momont was 15 grammes weight on the pan at 1m leverage from the motor center; moreover, the moment were practically constant from a speed of 1400 up to 1500 rev/mn.

To avoid confusion, the leverage was kept constant at 1m, and a weight of 15 g was secured to the pan during the whole experiment. Then the weight newly put on corresponds to M_e.

Example 16. (additional) weight = 78 g.

motor speed = 1455 rev/mn

then, $M_e = 0.078 \times 1 = 0.078$ mkg.

$\omega = 2 \times 1455/60 = 152.33$ rad/s

Power in P. S. $= 0.078 \times 152.33/75 = 0.159$

$E_f =$ Power reduced to $1450 = 0.159 \times 1450^3/1455^3$

$= 0.158$

§ MECHANICAL EFFICIENCIES.

Let η_m = total mechanical effiency, and

η'_m = effective mechanical efficiency, then

$$\eta_m = \frac{\text{theoretical total isothermal power } E}{\text{fan input power}} \quad \text{.........(16)}$$

$$\eta'_m = \frac{\text{theoretical effective isothermal power } E'}{\text{fan input power}} \quad \text{...(17)}$$

Example 17.

$$\eta_m = \frac{0.102}{0.158} = 0.652$$

$$\eta'_m = \frac{0.091}{0.158} = 0.575$$

§ PRESSURE EFFICIENCIES.

Let η_p = total pressure efficiency, and

η'_p = effective pressure efficiency, then

$$\eta_p = \frac{\text{observed total pressure difference } P}{(14)\times\gamma_0} \quad \text{.........(18)}$$

$$\eta'_p = \frac{\text{observed effective pressure difference } P'}{(14)\times\gamma_0} \quad \text{...(19)}$$

Example 18.

$$\eta_p = \frac{39.64}{53.64} = 0.74$$

$$\eta'_p = \frac{35\ 06}{53.64} = 0.653$$

§ VOLUMETRIC CAPACITY.

It is the ratio of the suction air volume treated, during one revolution, to the cubical content of the runner.[17]

$$\eta_v = V_s \Big/ \frac{\pi d^2}{4}.\ b.\ n \quad \text{..................................(20)}$$

where η_v = volumetric capacity.

d = outer diameter of the runner.

b = axial length of the runner.

n = speed of the fan in rev/s.

Example 19.

$d = 0.251$ m, $b = 0.111$ m $V_s = 0.1948$ m³/s $n = 1450/60$

then $\eta_v = 1.45$

4. SUMMARY OF EXPERIMENTAL AND CALCULATED DATA.

Table 1 shows the atmospheric conditions, etc.

Tables 2 and 3 show the static pressures, humidities, and densities of air in the suction pipe.

Tables 4_1, 4_2, 4_3, 4_4 and 4_5 show the air velocities and volumes in the suction pipe.

17) Lionel S. Marks, "Mech. Eng. Handbook." 1542.

Table 5 shows velocity pressures of suction and delivery, and the theoretical isothermal powers.

Table 6 shows pressure differences between suction and delivery.

Table 7 shows fan input powers.

Table 8 shows pressure efficiencies, mechanical efficiencies and volumetric capacities.

Plates 1, 2, 3, 4, and 5 show the performances of the fans 1, 2, 3, 4, and 5.

Plate 6 is the summary of the curves of the static depressions D and those of the effective pressure differences P'.

Plate 7 is the summary of the curves of the effective isothermae powers E'.

Plate 8 is the summary of the curves of the total isothermal powers E and those of the fan input powers E_f.

Plate 9 is the summary of the curves of total pressure differences P.

Plate 10 is the summary of the curves of the theoretical head H_w.

Plate 11 is the summary of the curves of the total mechanical efficiencies η_m.

Plate 12 is the summary of the curves of the effective mechanical efficiencies η'_m.

Plate 13 is the summary of the curves of the total pressure efficiencies η_p.

Plate 14 is the summary of the curves of the effective pressure efficiencies η'_p.

Plate 15 is the summary of the curves of the volumetric capacities η_v.

5. CONCLUSIONS.

Although my experiments were carried out under the conditions of variable barometric heights, atmospheric temperatures and humidities, and the various data are not reduced to certain standards, excepting that the characteristic curves are plotted on the reduced volumes as abscissae, yet the general tendencies of performances of the fans of variable axial lengths without evasée might be indicated.

By shortening the axial length of the fan (whose dimensions, forms of blades, and the profile of the casing are shown in figs. 3 and 4), and on the other hand, regulating the inlet area of the suction pipe and driving the fan at the constant speed of 1450 rev/mn, the following conclusions result so far as my experiments are concerned:

1, In a certain range of suction restrictions, for instance, between those of II and III, it is noticeable that the total pressure differences (plate 9), the total pressure efficiencies (Pl. 13), and the mechanical efficiencies (Pl. 11 and 12), of the fans of shorter axial lengths are higher, in a certain extent, than those of the fans of longer axial lengths.

2. But in the majority of suction restrictions, the above mentioned characteristics are better in the fan of longer axial length.

3. As regards the effective pressure efficiencies (Pl. 14), the fan of maximum axial length always shows the best results and the longer the length the better become these two.

4. As regards the theoretical heads (Pl. 10) and the volumetric capacities (pl. 15), the fan of shorter axial lengths show better results in the majority of suction restrictions. These may indicate that the eddy formations of air at the runner, before stated, gradually decrease more and more with the fans of shorter axial lengths.

Combining these items and taking the other characteristics into consideration, it may be said further that:

5. If the eddy formations, before stated, decrease and consequently the loss of energy by this cause decrease, by reducing the length of the fan, the surface frictional resistance in the casing, on the contrary, might have the predominating influence on the internal resistance of the fan, in general.

6. If a fan of this type has to be operated in a wide range of suction restrictions, that of maximum length gives the best results in general aspects, and consirdering the tendencies shown in the experiments, it might be allowed to conjecture that, regarding fan performances, a fan of this type of longer axial length than ordinary is, to a certain extent,

better for the purpose.

May 25th, 1924.

Mining Laboratory

Kyoto Imperial University.

Table 1. Atmospheric Conditions etc.

Fan		1	2	3	4	5	
Date		Feb. 5	Feb. 7	Feb. 9	Feb. 20	Feb. 23	1924
Duration of test		1-4 P.M.	2-5 P.M.	1-4 P.M.	1-4 P.M.	9-12 P.M.	
Weather		cloudy	rainy	cloudy	cloudy	fine	
Barometer reading		765.0	758.4	760.8	756.5	754.0	mm Hg
Ditto corrected by (2)		764.3	757.5	759.8	755.6	753.4	
Temperature by	dry thermometer	6	7.5	8	8	5	C°
	wet thermometer	4	6	5	5	2	
Ralative humidity by (3)		0.72	0.80	0.61	0.61	0.57	
Air constant by (6)		29.34	29.36	29.34	29.34	29.33	
Density of air by (8)		1.269	1.251	1.253	1.246	1.222	kg/m³
Density of alcohol by (4)		0.835	0.834	0.833	0.833	0.836	g/cm³

Table 2. Static pressures in suction pipe.

Fan resistance	Fan speed in rev/mn					Reading of static depression in mm of alcohol, by manometer inclination=15°					Static depression D reduced to 1450 rev/mn in mm Aq or kg/m²					Absolute pressure in mm Aq or kg/m²				
	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
○	1430	1440	1450	1460	1460	174	157	145	130	103	38.68	34.36	31.26	27.60	22.00	10352.74	10264.61	10298.98	10245.54	10221.23
I	1450	20	20	30	50	169	147	134	124	105	36.52	33.06	30.11	27.50	22.72	10354.90	265.91	300.13	245.64	220.51
II	1450	20	20	30	25	164	155	145	118	76	35.44	34.90	32.58	25.10	17.00	10355.98	264.07	297.66	248.04	226.23
III	1440	35	75	20	15	165	149	131	89	46	36.09	32.85	27.29	20.00	10.45	10355.33	266.12	302.95	253.14	232.78
IV	1425	50	60	10	40	140	119	94	58	27	31.33	25.72	19.98	13.20	5.92	10360.09	273.25	310.26	259.94	237.31
V	1410	20	60	20	30	99	77	58	35	15	22.69	17.30	12.33	7.87	3.34	10368.73	281.67	317.91	265.27	239.89
VI	1420	10	40	70	15	45	32	23	15	6	10.16	7.35	5.03	3.14	1.36	10381.26	291.62	325.21	270.00	242.87

Table 3. Humidities and Densities of air in suction pipe.

Fan resistance	Relative humidity φ_s by (5)					Air constant R_s by (7)					Air density γ_s by (9) in kg/m³					$\sqrt{\frac{2g}{\gamma_s}}$ in (10)				
	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
○											1.265	1.246	1.249	1.243	1.219					
I											1.265	1 246	1.249	1 243	1.219	3.936	3.966	3.961	3.971	4.010
II											1.265	1.246	1.249	1.243	1.219	3.936	3.966	3.961	3.971	4.010
III	0.72	0.80	0.61	0.61	0.57	29.34	29.36	29.34	29.34	29.23	1.265	1.247	1.250	1.244	1.220	3.936	3.965	3.960	3.969	4.008
IV											1.266	1.247	1.251	1.244	1.221	3.935	3.965	3.958	3.969	4.006
V											1.267	1.248	1.252	1.245	1.221	3.934	3.963	3 957	3.968	4.006
VI											1.268	1.250	1.253	1.246	1.222	3.931	3.961	3.955	3.966	4.005

Table 4-1. Air velocities and Volumes measured in Suction pipe. (Fan 1.)

Measuring position in fig. 9	Fan speed in rev/mn						Reading in mm of alcohol (0.835 density), by manometer inclination=					
	I	II	III	IV	V	VI	0° 35′ I	1° 10′ II	1° 10′ III	5° 45′ IV	5° 45′ V	5° 45′ VI
1′	1440	1465	1440	1480	1420	1410	10	16	35	12	16	21
2′	1440	65	40	80	20	20	11	16	40	20	24	30
3′	1440	60	10	80	20	50	11	23	52	22	27	35
4′	1480	60	80	80	20	50	14	25	61	22	30	39
5′	1480	40	80	70	50	50	14	26	64	23	31	42
6′	1480	40	80	70	50	50	15	26	65	23	32	44
7′	1480	40	80	70	50	30	15	26	66	24	32	42
8′	1460	40	80	70	50	30	15	26	66	23	32	42
C	1460	40	40	50	50	50	15	26	62	22	32	44
8	1460	35	40	50	50	50	15	26	62	22	32	44
7	1440	35	40	40	50	50	14	26	61	21	32	44
6	1440	35	40	40	50	50	14	26	60	21	32	44
5	1440	35	40	40	60	50	14	26	60	21	31	44
4	1430	40	65	40	60	50	12	25	58	20	30	42
3	1430	40	65	40	60	50	11	22	56	19	28	39
2	1410	40	70	60	50	50	10	16	48	18	24	35
1	1480	40	40	60	50	50	11	16	35	12	17	25

Measuring position in fig. 9	h_v in (10) reduced to 1450 rev/mn in mm Aq or kg/m²						Air velocity, by (10) in m/s.					
	I	II	III	IV	V	VI	I	II	III	IV	V	VI
1′	0.085	0.262	0.593	0.958	1.391	1.850	1.15	2.02	3.03	3.85	4.64	5.35
2′	0.093	0.262	0.678	1.596	2.085	2.610	1.20	2.02	3.24	4.97	5.69	6.35
3′	0.093	0.378	0.917	1.756	2.344	2.923	1.20	2.42	3.76	5.22	6.02	6.72
4′	0.112	0.413	0.973	1.756	2.610	3.257	1.32	2.53	3.89	5 22	6.35	7.10
5′	0.112	0.441	1.022	1.869	2.589	3.507	1.32	2.62	3.98	5.38	6.33	7.36
6′	0.120	0.441	1.037	1.869	2.672	3.675	1.36	2.62	4.01	5.38	6.43	7.54
7′	0.120	0.441	1.054	1.950	2.672	3.608	1.36	2.62	4.04	5.49	6.43	7.47
8′	0.124	0.441	1.054	1.869	2.672	3.608	1.38	2.62	4.04	5.38	6.43	7.47
C	0.124	0.441	1.050	1.837	2.672	3.675	1.38	2.62	4.04	5.33	6.43	7.54
8	0.124	0.444	1.050	1.837	2.672	3.675	1.38	2.62	4.04	5.33	6.43	7.54
7	0.119	0.444	1.032	1.778	2.672	3.675	1.36	2.62	3.99	5.24	6.43	7.54
6	0.119	0.444	1.016	1.778	2.672	3.675	1.36	2.62	3.97	5.24	6.43	7.54
5	0.119	0.444	1.016	1.778	2.556	3.675	1.36	2.62	3.97	5.24	6.29	7.54
4	0.103	0.425	0.949	1.693	2.470	3.507	1.26	2.57	3.84	5.12	6.22	7.36
3	0.095	0.373	0.916	1.608	2.305	3.257	1.21	2.40	3.76	4.99	5.98	7.10
2	0.088	0.271	0.781	1.484	2.004	2.923	1.17	2.05	3.48	4.80	5.57	6.72
1	0.083	0.271	0.593	0.989	1.420	2.087	1.13	2.05	3.03	3.91	4.69	5.68

Calculated results.

	Reduced volume V_r by (11) in m³/s	Volume V_r by (1) in m³/s	Mean velocity by (1) in m/s. Suction area at measuring section = $\frac{\pi}{4} \times 0.315^2 = 0.079$ m²
I	0.111	0.104	1.30
II	0.208	0.195	2.44
III	0.320	0.300	3.76
IV	0.431	0.403	5.05
V	0.515	0.481	6.02
VI	0.601	0.562	7.03

Table 4-2. Air velocities and Volumes measured in suction pipe. (Fan 2.)

Measuring position in fig. 9	Fan speed in rev/mn						Reading in mm of alcohol (0.834 density), by manometer inclination=						h_v in (10) reduced to 1450 rev/mn in mm Aq or kg/m²						Air velocity, by (10) in m/s.					
	I	II	III	IV	V	VI	0° 35′ I	1° 10′ II	1° 10′ III	1° 10′ IV	5° 45′ V	5° 45′ VI	I	II	III	IV	V	VI	I	II	III	IV	V	VI
1′	1450	1475	1475	1465	1470	1420	8	15	30	46	13	15	0.067	0.241	0.484	0.752	1.056	1.307	1.02	1.95	2.76	3.44	4.07	4.53
2′	1450	75	75	60	70	20	8	19	44	71	18	21	0.067	0.306	0.698	1.167	1.460	1 825	1.02	2.19	3.32	4.28	4.79	5.34
3′	1450	75	65	60	70	30	9	23	52	79	22	25	0.075	0.371	0.850	1.298	1.785	2.140	1.08	2.42	3.66	4.52	5.29	5.78
4′	1450	75	50	70	60	30	12	26	54	85	23	29	0.100	0.418	0.902	1.380	1.892	2.490	1.25	2.57	3.77	4.66	5.45	6.25
5′	1450	90	55	70	60	50	12	27	57	92	24	30	0.100	0.427	0.945	1.493	1.970	2.502	1.25	2.59	3.86	4.85	5.57	6.26
6′	1450	60	55	70	60	50	12	26	57	93	24	30	0.100	0.428	0.945	1.510	1.970	2.502	1.25	2.60	3.86	4.88	5.57	6.26
7′	1460	60	70	70	60	50	12.5	26	58	94	24	31	0.140	0.428	0.943	1.525	1.970	2.585	1.29	2.60	3.85	4.90	5.57	6.37
8′	1460	55	70	70	45	50	12.5	26	58	94	24	31	0.140	0.430	0.943	1.525	2.014	2.585	1.29	2.61	3.85	4.90	5.62	6.37
C	1450	40	70	65	45	50	12.5	26	58	94	24	31	0.142	0.440	0.943	1.537	2.014	2.585	1.30	2.63	3.85	4.92	5.62	6.37
8	1450	40	70	60	45	50	12.5	26	58	92	24	31	0.142	0.440	0.943	1.513	2.014	2.585	1.30	2.63	3.85	4.88	5.62	6.37
7	1455	40	65	60	45	50	12.5	26	57	92	24	30	0.141	0.440	0.933	1.513	2.014	2.502	1.29	2.63	3.83	4.88	5.62	6.25
6	1455	50	65	60	45	50	12	26	56	92	24	30	0.099	0.434	0.915	1.513	2.014	2.502	1.25	2.61	3.79	4.88	5.62	6.25
5	1455	50	65	40	45	50	12	26	56	87	23	30	0.099	0.434	0.915	1.470	1.930	2.502	1.25	2.61	3.79	4.82	5.51	6.25
4	1460	50	65	40	45	50	11	25	53	86	23	29	0.090	0.417	0.867	1.455	1.930	2.420	1.19	2.56	3.69	4.78	5.51	6.15
3	1460	60	65	35	60	50	9	22	50	75	21	26	0.074	0.363	0.817	1.278	1.724	2.170	1.08	2.39	3.58	4.48	5.22	5.83
2	1460	60	65	25	60	40	8	18	43	69	18	22	0.066	0.296	0.704	1.190	1.480	1.860	1.02	2.16	3.33	4.33	4.83	5.40
1	1460	60	70	60	55	40	8	15	31	47	13	16	0.066	0.246	0.503	0.773	1.077	1.352	1.02	1.97	2.82	3.49	4.11	4.61

Calculated results.

	Reduced volume V_r by (11) in m³/s	Volume V_s by (1) in m³/s	Mean velocity by (1) in m/s. Suction area at measuring section = $\frac{\pi}{4} \times 0.315^2 = 0.0799$ m²
I	0.099	0.094	1.18
II	0.204	0.194	2.44
III	0.304	0.288	3.60
IV	0.388	0.365	4.56
V	0.452	0.428	5.35
VI	0.498	0.472	5.90

Table 4-3. Air velocities and Volumes measured in suction pipe. (Fan 3.)

Measuring position in fig. 9	Fan speed in rev/mn I	II	III	IV	V	VI	Reading in mm of alcohol (0.833 density), by manometer inclination= 0° 35′ I	0° 35′ II	1° 10′ III	1° 10′ IV	1° 10′ V	5° 45′ VI
1′	1455	1460	1460	1480	1450	1440	6	26	23	39	44	10
2′	1455	40	60	65	50	40	6.5	33	33	55	65	15
3′	1480	40	55	65	50	40	7.5	40	40	61	77	17
4′	1480	35	70	65	60	40	10	44	42	65	81	18
5′	1480	40	70	60	60	40	10	46	44	68	86	19
6′	1480	40	70	60	60	40	10	46	44	69	87	20
7′	1460	40	60	60	60	40	10	46	44	69	87	20
8′	1460	40	60	50	60	40	10	46	44	67	86	20
C	1480	40	40	70	60	40	10	46	43	69	86	20
8	1480	40	40	75	60	40	10	46	43	69	86	20
7	1480	40	30	75	60	40	10	46	42	69	86	20
6	1480	40	25	90	60	40	10	46	41	69	86	20
5	1470	40	25	60	60	30	10	46	41	66	86	20
4	1470	25	30	60	60	30	10	43	41	63	81	20
3	1470	25	30	60	60	40	7.5	42	38	58	77	18
2	1460	25	30	45	60	40	6.5	30	31	50	65	15
1	1460	25	30	45	65	40	6	27	23	37	46	10

Measuring position in fig. 9	h_v in (10) reduced to 1450 rev/mn in mm Aq or kg/m² I	II	III	IV	V	VI
1′	0.050	0.214	0.378	0.621	0.733	0.845
2′	0.054	0.279	0.542	0.898	1.083	1.267
3′	0.060	0.338	0.662	0.996	1.283	1.436
4′	0.080	0.374	0.680	1.061	1.331	1.520
5′	0.080	0.389	0.713	1.118	1.414	1.605
6′	0.080	0.389	0.713	1.130	1.430	1.689
7′	0.082	0.389	0.723	1.130	1.430	1.689
8′	0.082	0.389	0.723	1.162	1.414	1.689
C	0.080	0.389	0.726	1.119	1.414	1.689
8	0.080	0.389	0.726	1.111	1.414	1.689
7	0.080	0.389	0.720	1.111	1.414	1.689
6	0.080	0.389	0.707	1.088	1.414	1.689
5	0.081	0.389	0.707	1.085	1.414	1.712
4	0.081	0.371	0.702	1.035	1.331	1.712
3	0.061	0.362	0.651	0.953	1.265	1.520
2	0.053	0.259	0,531	0.838	1.068	1.267
1	0.049	0.233	0.394	0.620	0.760	0.845

Measuring position in fig. 9	Air velocity, by (10) in m/s. I	II	III	IV	V	VI
1′	0.88	1.83	2.44	3.12	3.39	3.64
2′	0.92	2.12	2.91	3.75	4.12	4.45
3′	0.97	2.31	3.22	3.95	4.49	4.74
4′	1.12	2.42	3.27	4.08	4.56	4.87
5′	1.12	2.47	3.25	4.18	4.71	5.02
6′	1.12	2.47	3.25	4.21	4.74	5,14
7′	1.14	2.47	3.37	4.21	4.74	5.14
8′	1.14	2.47	3.37	4.27	4.71	5.14
C	1.12	2.47	3.38	4.19	4.71	5.14
8	1.12	2.47	3.38	4.17	4.71	5.14
7	1.12	2.47	3.36	4.17	4.71	5.14
6	1.12	2.47	3.33	4.17	4.71	5.14
5	1.13	2.47	3.33	4.16	4.71	5.17
4	1.13	2.41	3.32	4.03	4.56	5.17
3	0.98	2.39	3.19	3.87	4.45	4.87
2	0.92	2.05	2.88	3.63	4.09	4.45
1	0.88	1.91	2.49	3.12	3.45	3.64

Calculated results.

	Reduced volume V_r by (11) in m³/s	Volume V_s by (1) in m³/s	Mean velocity by (1) in m/s. Suction area at measuring section = $\frac{\pi}{4} \times 0.315^2$ 0.0779 m²
I	0.089	0.084	1.05
II	0.196	0.186	2.32
III	0.267	0.253	3.16
IV	0.333	0.315	3.94
V	0.374	0.354	4.43
VI	0.404	0.382	4.78

Table 4-4. Air velocities and Volumes measured in Suction pipe. (Fan 4.)

Measuring position in fig. 9	Fan speed in rev/mn						Reading in mm of alcohol (0.833 density), by manometer inclination =						h_v in (10) reduced to 1450 rev/mn in mm Aq or kg/m²						Air velocity, by (10) in m/s.					
							0° 35′	1° 10′	1° 10′	1° 10′	1° 10′	1° 10′												
	I	II	III	IV	V	VI	I	II	III	IV	V	VI	I	II	III	IV	V	VI	I	II	III	IV	V	VI
1′	1465	1445	1460	1470	1490	1475	6	11	17	24	29	31	0.049	0.184	0.280	0.389	0.457	0.498	0.89	1.70	2.10	2.48	2.68	2.80
2′	1440	45	60	70	80	75	6	14	25	36	41	42	0.051	0.235	0.410	0.584	0.653	0.676	0.89	1.92	2.54	3.03	3.21	3.26
3′	1450	55	60	70	80	70	7	17	29	39	46	48	0.058	0.282	0.478	0.633	0.732	0.779	0.96	2.11	2.74	3.16	3.39	3.50
4′	1475	55	40	55	80	70	9	18	31	42	50	54	0.072	0.298	0.524	0.695	0.796	0.876	1.07	2.17	2.88	3.31	3.54	3.71
5′	1475	55	40	40	80	80	10	19	32	43	53	58	0.080	0.315	0.541	0.726	0.844	0.924	1.13	2.23	2.92	3.39	3.64	3.77
6′	1475	55	40	40	80	90	10	19	32	43	53	60	0.080	0.315	0.541	0.726	0.844	0.947	1.13	2.23	2.92	3.39	3.64	3.85
7′	1460	55	40	30	80	90	10	19	32	44	55	60	0.082	0.315	0.541	0.754	0.876	0.947	1.14	2.23	2.92	3.45	3.71	3.85
8′	1460	55	40	30	80	90	10	19	32	44	55	60	0.082	0.315	0.541	0.754	0.876	0.947	1.14	2.23	2.92	3.45	3.71	3.85
C	1460	70	40	30	70	80	10	19	32	44	53	60	0.082	0.309	0.541	0.754	0.859	0.956	1.14	2.20	2.92	3.45	3.68	3.88
8	1460	70	40	40	70	80	10	19	32	44	53	60	0.082	0.309	0.541	0,743	0.859	0.956	1.14	2.20	2.92	3.42	3.68	3.88
7	1450	70	40	40	70	80	10	19	32	44	53	60	0.083	0.309	0.541	0.743	0.859	0 956	1.15	2.20	2.92	3.42	3.68	3.88
6	1450	80	40	40	70	80	10	19	31	44	53	60	0.083	0.303	0.524	0.743	0.859	0.956	1.15	2.18	2.88	3.42	3.68	3.88
5	1450	80	40	40	70	80	10	19	31	43	53	60	0.083	0.303	0.524	0.726	0.859	0.956	1.15	2.18	2.88	3.38	3.68	3.88
4	1450	80	50	50	70	90	10	18	30	42	50	60	0.083	0.287	0.500	0.700	0.811	0.947	1.15	2.13	2.81	3.32	3.57	3.85
3	1450	55	50	50	70	90	8	16	27	38	47	54	0.067	0.265	0.450	0.633	0.762	0.852	1.02	2.05	2.66	3.16	3.46	3.66
2	1440	55	50	50	70	90	7	14	22	37	38	43	0.059	0.231	0.366	0.617	0.616	0.678	0.97	1.91	2.40	3.12	3.09	3.26
1	1440	55	50	50	70	80	6	12	18	25	30	33	0.051	0.199	0.300	0.416	0.487	0.526	0.90	1.77	2.18	2.56	2.77	2.88

Calculated results.

	Reduced volume V_r by (11) in m³/s	Volume V_s by (1) in m³/s	Mean velocity by (1) in m/s. Suction area at measuring section = $\frac{\pi}{4} \times 0.315^2 = 0.0799$ m²
I	0.089	0.085	1.06
II	0.175	0.167	2.09
III	0.229	0.218	2.73
IV	0.270	0.257	3.21
V	0.289	0.275	3.44
VI	0.305	0.290	3.63

Table 4-5. Air velocities and Volumes measured in suction pipe. (Fan 5.)

Measuring position in fig. 9	Fan speed in rev/mn						Reading in mm of alcohol (0.836 density), by manometer inclination=						h_v in (10) reduced to 1450 rev/mn in mm Aq or kg m²						Air velocity, by (10) in m/s.					
	I	II	III	IV	V	VI	0° 35′ I	0° 35′ II	0 35′ III	1° 10′ IV	1° 10′ V	1° 10′ VI	I	II	III	IV	V	VI	I	II	III	IV	V	VI
1′	1460	1460	1420	1460	1450	1420	5	13	14	11	11	11	0.041	0.107	0.122	0.182	0.184	0.192	0.81	1.32	1.40	1.71	1.72	1.75
2′	1460	90	20	60	70	40	5	18	21	14	16	14	0.041	0.143	0.183	0.231	0.260	0.237	0.81	1.51	1.71	1.93	2.04	1.95
3′	1460	70	30	60	70	60	5.5	19	25	17	18	17	0.045	0.155	0.215	0.280	0.292	0.280	0.85	1.58	1.86	2.12	2.17	2.12
4′	1460	70	30	25	70	60	7.5	20	29	17	20	20	0.062	0.163	0.246	0.294	0.325	0.330	1.00	1.62	2.00	2.17	2.28	2.30
5′	1460	70	35	25	70	60	8	21	29	18	20	21	0.066	0.171	0.247	0.311	0.325	0.346	1.03	1.67	1.99	2.24	2.28	2.35
6′	1460	70	35	50	60	60	8	21	29	19	20	21	0.066	0.171	0.247	0.317	0.330	0.346	1.03	1.67	1.99	2.26	2.30	2.35
7′	1470	40	35	50	60	40	8	21	29	19	20	21	0.065	0.178	0.247	0.317	0.330	0.356	1.02	1.69	1.99	2.26	2.30	2.39
8′	1470	60	40	50	60	40	8	22	30	19	20	21	0.065	0.182	0.255	0.317	0.330	0.356	1.02	1.71	2.02	2.26	2.30	2.39
C	1470	60	60	30	60	40	8	22	31	18	20	21	0.065	0.182	0 256	0.309	0.330	0.356	1.02	1.71	2.02	2.23	2.30	2.39
8	1470	60	60	30	60	40	8	22	31	18	20	21	0.065	0.182	0.256	0.309	0.330	0.356	1.02	1.71	2.02	2.23	2.30	2.39
7	1455	60	60	30	60	30	8	21	31	18	20	21	0.066	0.173	0.256	0.309	0.330	0.361	1.03	1.67	2.02	2.23	2.30	2.40
6	1455	60	60	30	60	30	7.5	21	31	18	20	21	0.062	0.173	0.256	0.309	0.330	0.361	1.00	1.67	2.02	2.23	2 30	2.40
5	1455	60	45	20	60	30	7.5	21	30	18	20	21	0.062	0.173	0.253	0.313	0.330	0.361	1.00	1.67	2.01	2.24	2.30	2.40
4	1455	50	45	50	55	55	7	20	29	17	19	20	0.058	0.167	0.244	0.284	0.315	0.332	0.97	1.64	1.98	2.14	2.25	2.31
3	1440	50	45	40	55	55	6	19	25	16	18	19	0.051	0.159	0.210	0.271	0.298	0.315	0.90	1.60	1.84	2.08	2.19	2.25
2	1440	50	65	40	70	40	5.5	17	21	13	15	15	0.047	0.142	0.172	0.220	0.244	0.255	0.87	1.51	1.66	1.88	1.98	2.02
1	1440	50	65	40	70	40	5	13	17	11	12	12	0.042	0.109	0.139	0.187	0.195	0.203	0.83	1.32	1.50	1.73	1.77	1.80

Calculated results.			
Reduced volume V_r by (11) in m³/s	Volume V_s by (1) in m³/s	Mean velocity by (1) in m/s. Suction area at measuring section $= \frac{\pi}{4} \times 0.315^2 = 0.0799$ m²	
0.080	0.076	0.95	I
0.135	0.128	1.60	II
0.159	0.150	1.88	III
0.179	0.169	2.11	IV
0.184	0.174	2.18	V
0.189	0.178	2.22	VI

Table 5. Theoretical isothermal powers.

Fan resistance.	Suction velocity pressure as in example 12. im mm Aq or kg/m².					Absolute suction pressure p_1 in (12) in mm Aq or kg/m².					Delivery velocity pressure as in example 12. in mm Aq or kg/m². Delivery area in m².=				
											0.023	0.019	0.016	0.012	0.0085
	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
I	0.11	0.09	0.07	0.07	0.06	10355.01	10266.00	10300.20	10245.71	10220.57	1.31	1.52	1.81	3.05	4.94
II	0.38	0.38	0.34	0.28	0.16	10356.36	264.45	298.00	248.32	226.39	4.58	6.50	8.90	11.80	14.05
III	0.91	0.66	0.64	0.48	0.22	10356.24	266.78	303.59	253.62	233.00	10.84	14.24	16.50	20.15	19.35
IV	1.65	1.32	1.00	0.66	0.28	10361.74	274.57	311.26	260.60	237.59	19.63	22.80	25.60	28.10	24.43
V	2.34	1.82	1.25	0.75	0.30	10371.07	283.49	319.16	266.02	240.19	27.98	31.50	32.40	32.30	25.98
VI	3.20	2.22	1.46	0.85	0.31	10384.46	293.84	326.67	270.85	243.18	38.17	38.60	37.70	36.20	27.40

Table 5. *Continued.*

Fan resistance.	Absolute delivery pressure p_2 in (12) in mm Aq or kg/m².					Total isothermal power E by (12) in P.S.					Effective isothermal power E' as in example 12. in P.S.				
	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
I	10392.73	10300.49	10332.05	10276.19	10248.17	0.052	0.043	0.036	0.034	0.028	0.050	0.041	0.034	0.031	0.023
II	10396.00	305.47	339.14	284.94	257.28	0.102	0.105	0.101	0.082	0.053	0.091	0.089	0.079	0.055	0.029
III	10402.25	313.21	346.74	293.29	262.58	0.183	0.178	0.146	0.115	0.059	0.142	0.123	0.089	0.056	0.020
IV	10411.05	321.25	356.84	301.24	267.66	0.265	0.227	0.191	0.139	0.068	0.156	0.119	0.079	0.043	0.012
V	10419.40	330.47	362.64	305.44	269.21	0.310	0.266	0.204	0.144	0.067	0.131	0.088	0.051	0.026	0.006
VI	10429.59	337.57	367.94	309.34	270.63	0.338	0.274	0.210	0.158	0.065	0.052	0.032	0.013	0.009	—

Table 6. Pressure differences between suction and delivery.

Fan resistance	Total pressure difference P as in example 14. in mm Aq or kg/m².					Effective pressure difference P' as in example 14. in mm Aq or kg/m².					Theoretical head H_w as in example 15. in mm Aq or kg/m².				
	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
O	38.68	34.36	31.26	27.60	22.00	38.68	34.36	31.26	27.60	22.00	47.04	46.37	46.45	46.18	45.27
I	37.72	34.49	31.84	30.45	27.56	36.41	32.97	30.04	27.43	22.62	50.56	50.18	50.75	51.85	53.15
II	39.64	41.02	41.14	36.62	30.89	35.06	34.52	32.24	24.82	16.84	53.64	54.23	55.92	57.35	58.60
III	46.02	46.43	43.15	39.67	29.58	35.18	31.86	26.65	19.52	10.23	57.23	57.97	59.35	60.87	60.85
IV	49.31	47.20	44.58	40.64	30.07	29.68	24.40	18.98	12.54	5.64	60.74	61.14	62.55	63.63	62.80
V	48.33	46.98	43.48	39.42	29.02	20.35	15.48	11.08	7.12	3.04	63.40	63.64	64.51	64.94	63.35
VI	45.13	43.73	41.27	38.49	28.45	6.96	5.13	3.57	2.29	1.05	66.15	65.30	65.96	65.96	63.80

Table 7. Fan input powers.

Fan resistance	Moment M_e in (15) in mkg, by leverage = 1 m.					Fan speed in rev/mn.					Power E_f reduced to 1450 rev/mn as in example 16. in P.S.				
	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
O	0.023	0.01	—	—	—	1460	1440	—	—	—	0.047	0.021	—	—	—
I	0.038	0.033	0.030	—	—	1455	55	40	—	—	0.077	0.066	0.064	—	—
II	0.078	0.075	0.068	0.065	—	1455	55	40	1440	—	0.158	0.150	0.139	0.134	—
III	0.130	0.120	0.105	0.098	0.063	1460	45	40	60	1440	0.260	0.244	0.215	0.196	0.129
IV	0.175	0.160	0.135	0.115	—	1445	40	60	45	—	0.357	0.328	0.270	0.234	—
V	0.213	0.195	0.160	0.125	—	1450	50	45	40	—	0.432	0.395	0.326	0.256	—
VI	0.268	0.233	0.185	0.140	0.080	1445	45	60	45	1450	0.547	0.480	0.368	0.286	0.162

Table 8. Efficiencies and Volumetric Capacities.

Fan resistance.	Total pressure efficiency η_p by (18), in %.					Effective pressure efficiency η_p' by (19) in %.					Total mechanical efficiency η_m by (16) in %.				
	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
O	82	74	68	60	49	82	74	68	60	49					
I	75	69	63	59	52	72	66	60	54	43	68	65	56	—	—
II	74	76	74	64	53	65	64	58	43	29	65	71	73	61	—
III	79	80	73	65	49	62	55	45	32	17	71	73	68	59	46
IV	82	77	71	64	48	49	40	30	20	9	72	70	71	60	—
V	76	74	68	61	46	32	24	17	11	5	72	68	63	56	—
VI	68	67	63	59	45	11	8	5	3	2	62	57	57	55	40

Table 8. *Continued.*

Fan resistance.	Effective mechanical efficiency η_m' by (17) in %.					Volumetric Capacity η_v by (20).				
	1	2	3	4	5	1	2	3	4	5
O										
I	65	63	54	—	—	0.78	0.86	0.96	1.30	1.82
II	58	59	57	41	—	1.45	1.77	2.12	2.56	3.06
III	46	51	41	29	16	2.24	2.62	2.88	3.32	3.60
IV	45	36	29	18	—	3.01	3.32	3.60	3.93	4.05
V	30	22	16	10	—	3.60	3.88	4.05	4.21	4.17
VI	10	7	4	3	—	4.20	4.28	4.37	4.44	4.26

PLATE 1. Characteristics of fan I.

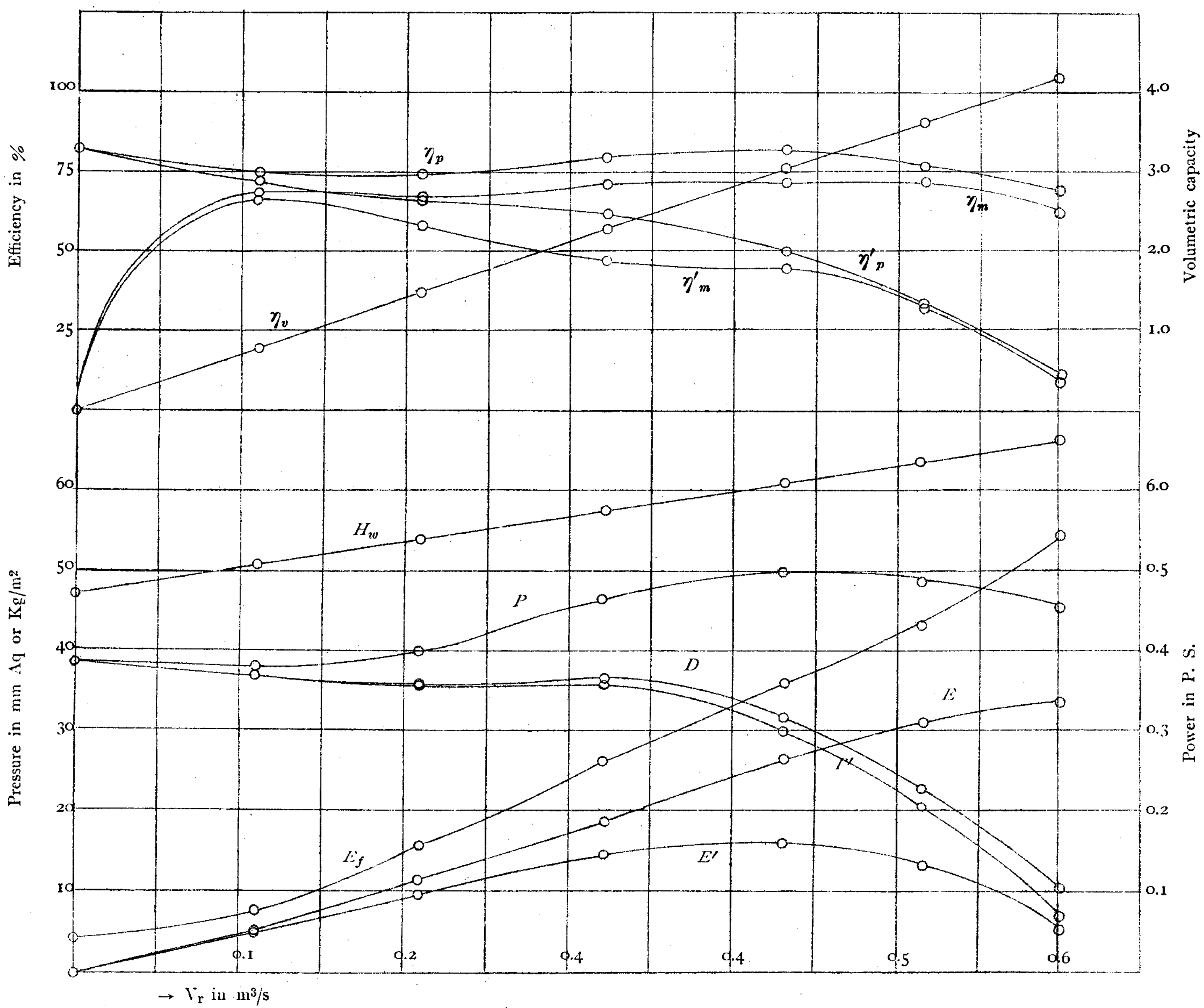

PLATE 2. Characteristics of fan 2.

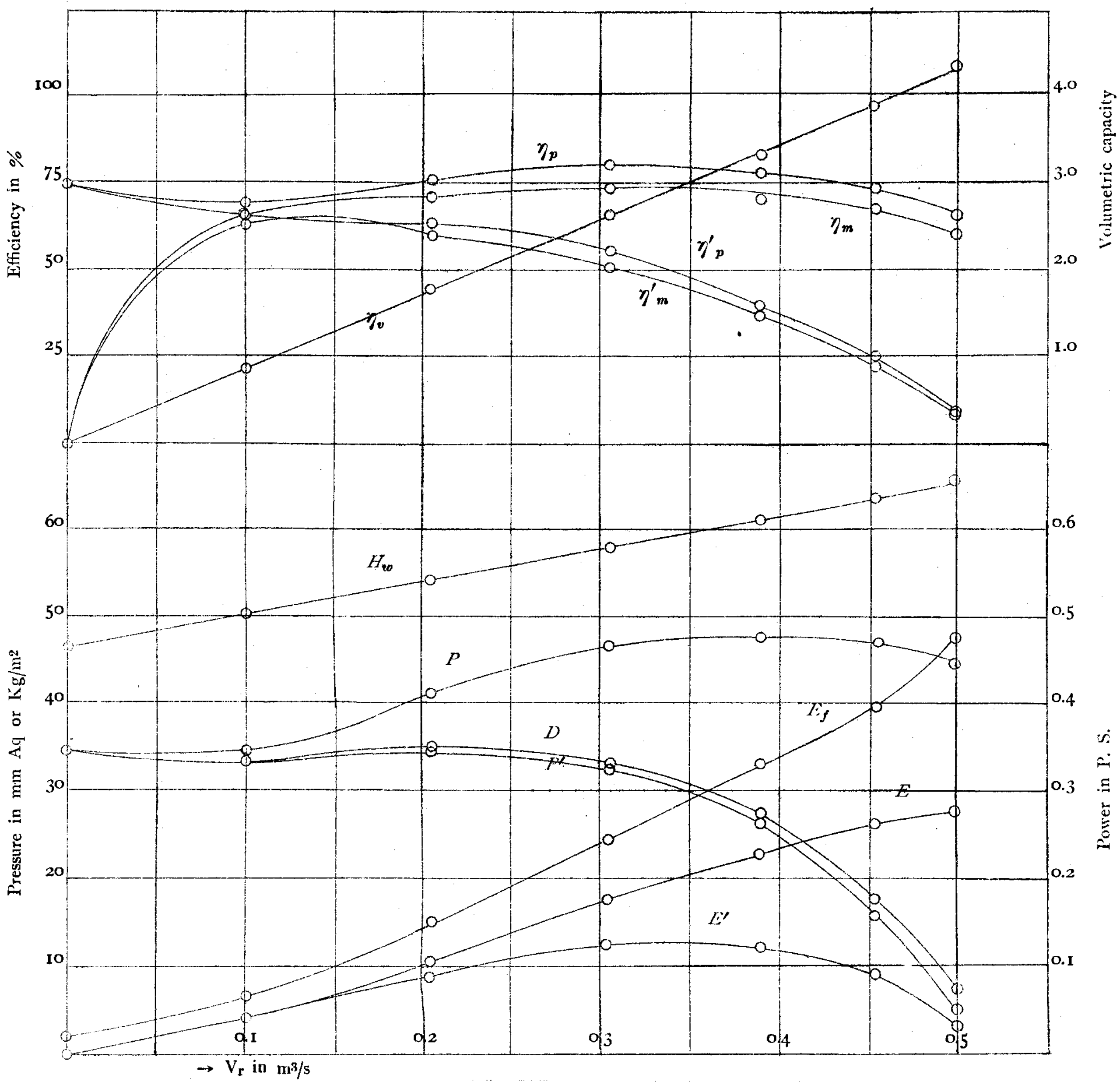

PLATE 3. Characteristics of fan 3.

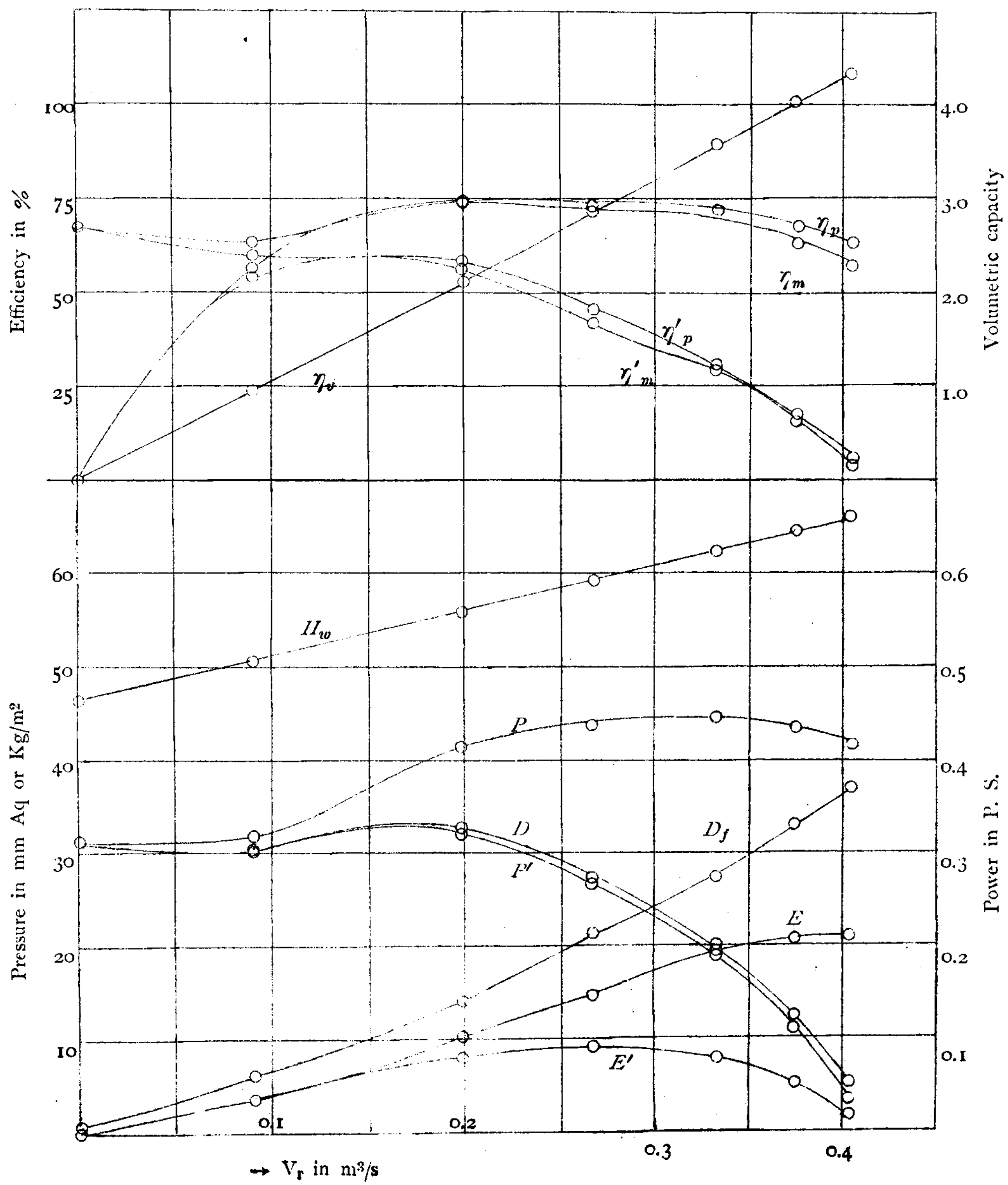

PLATE 4. Characteristics of fan 4.

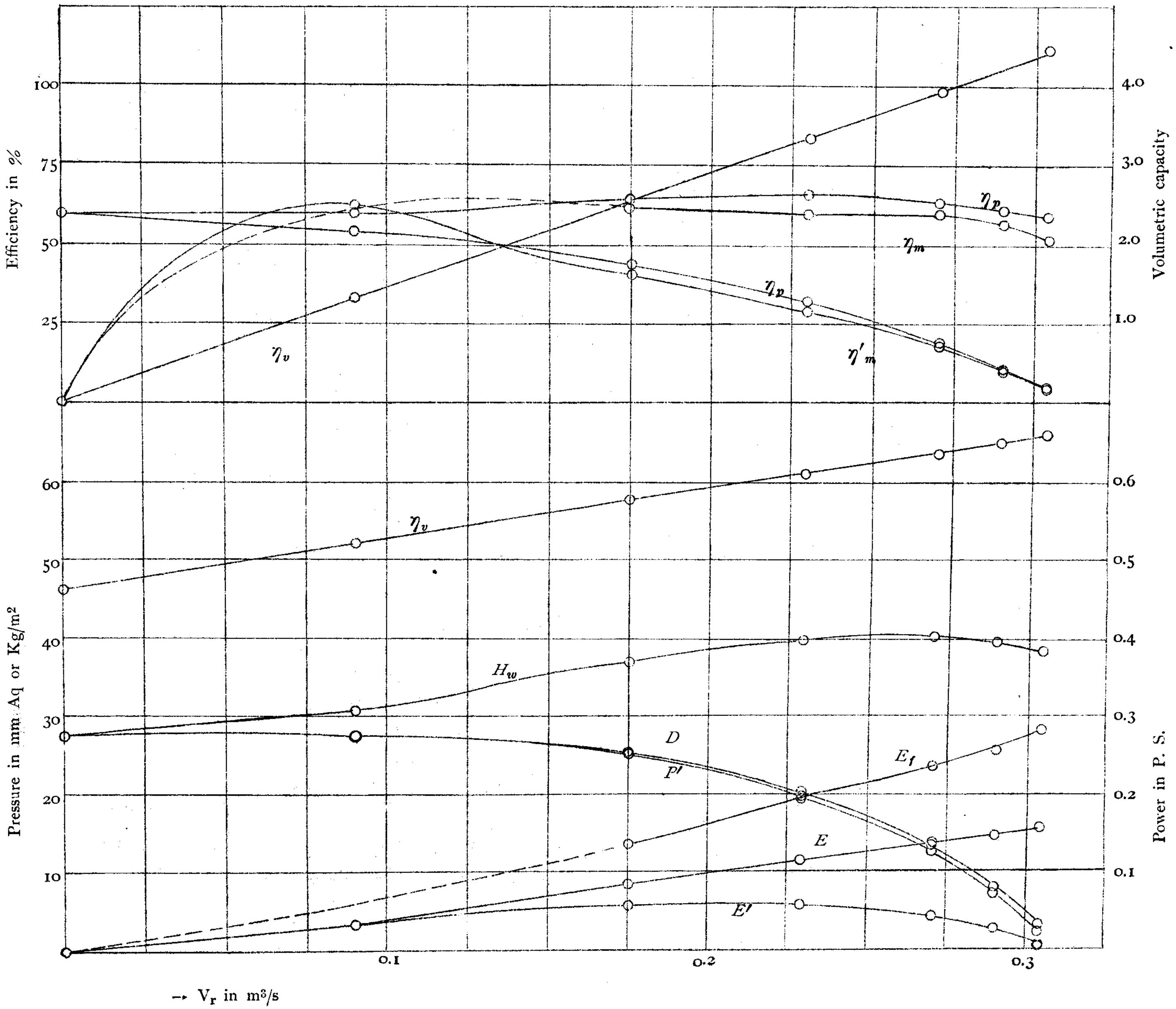

PLATE 5. Characteristics of fan 5.

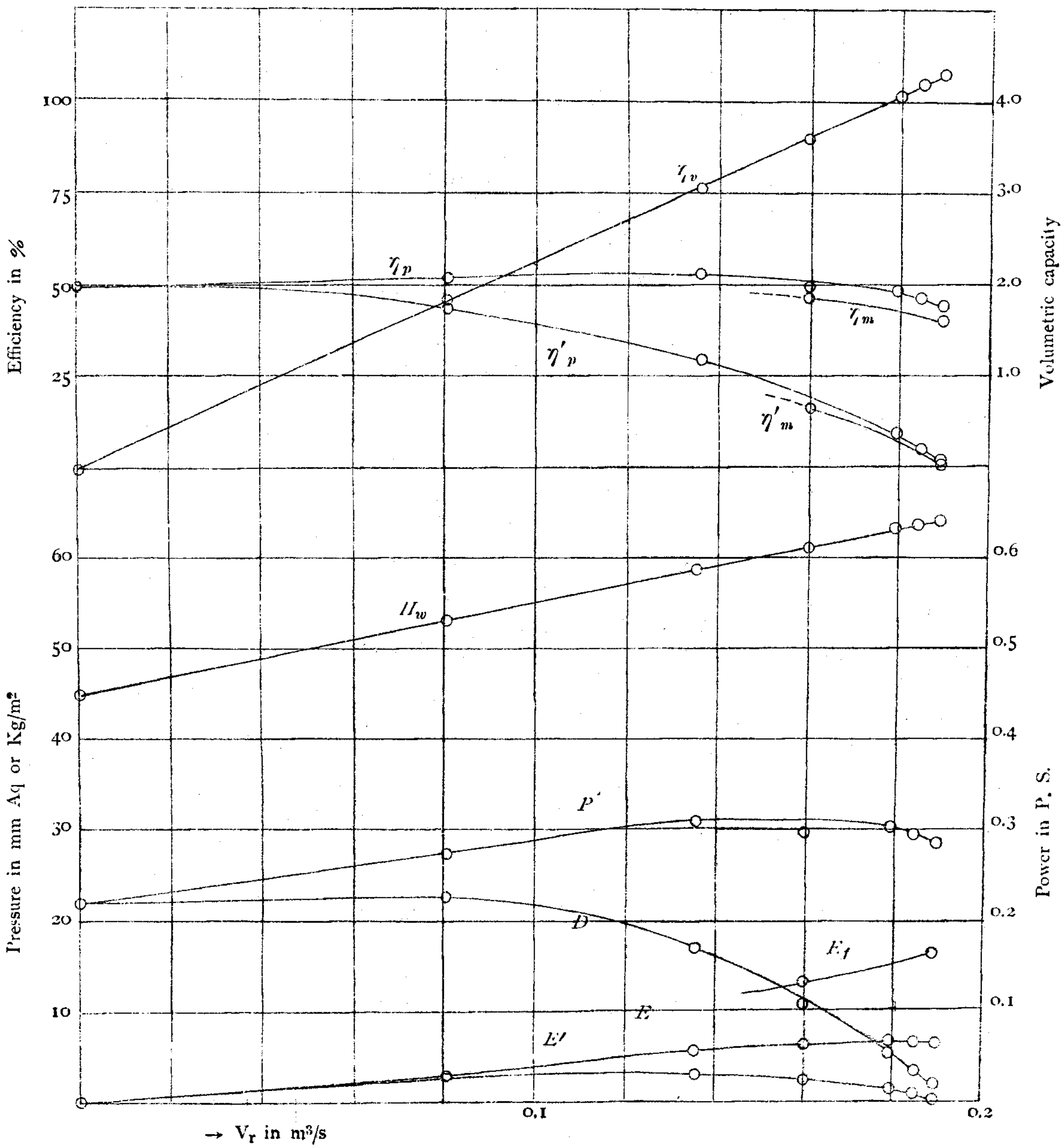

PLATE 6. Summary of curves D and *P′*

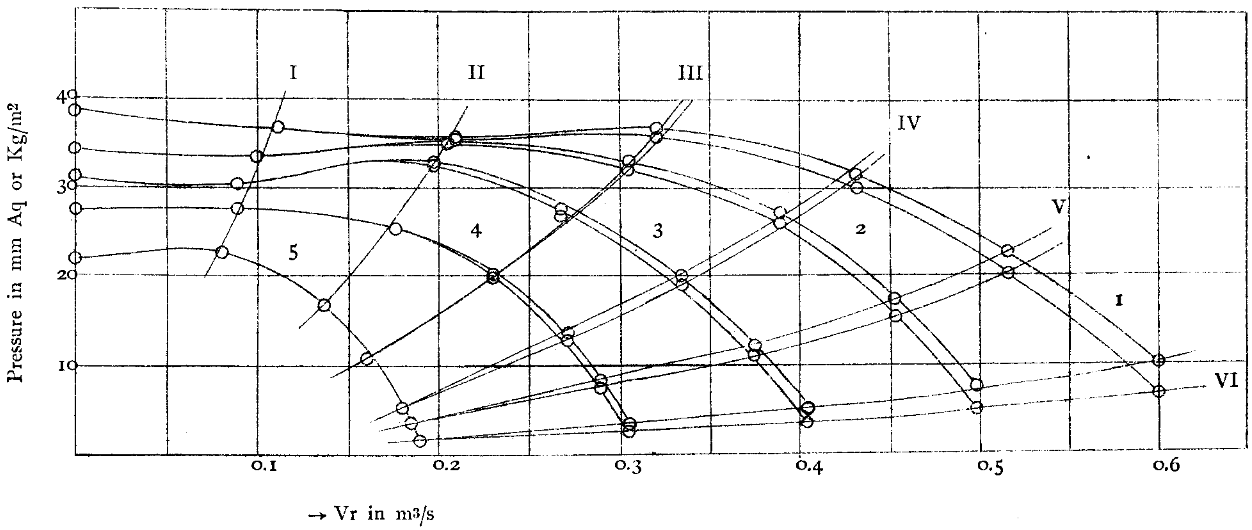

PLATE 7. Summary of curves *E′*

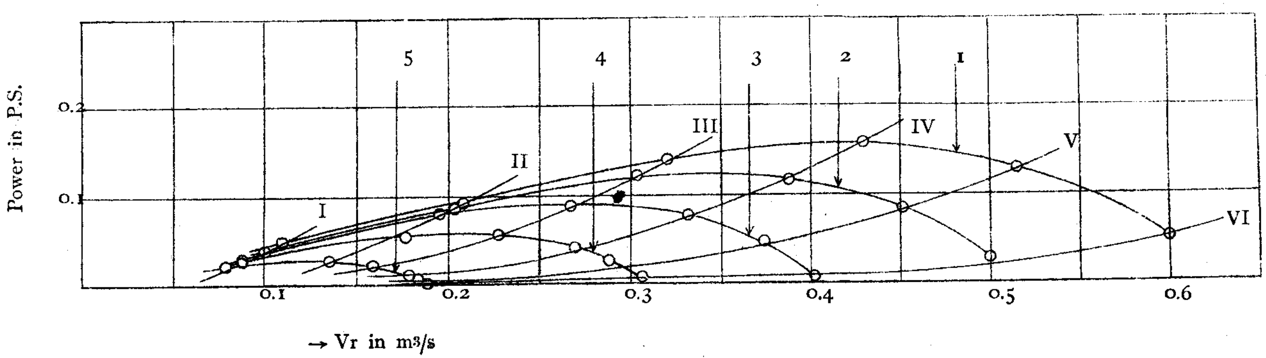

PLATE 8. Summary of curves E and E_f

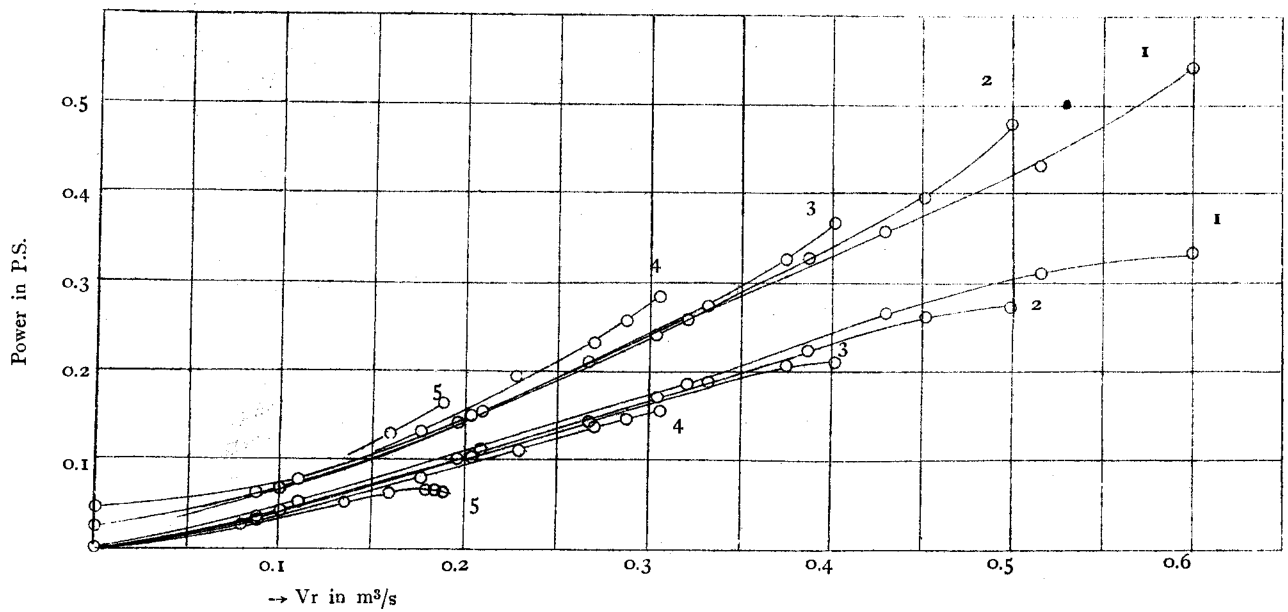

PLATE 9. Summary of curves *P*

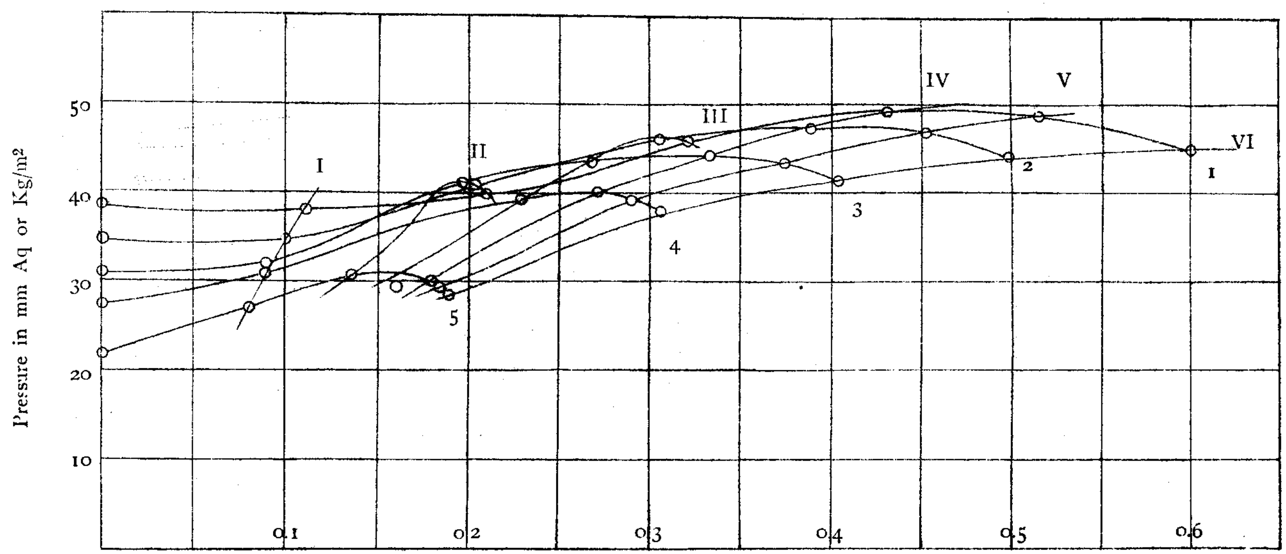

PLATE 10. Summary of curves H_w

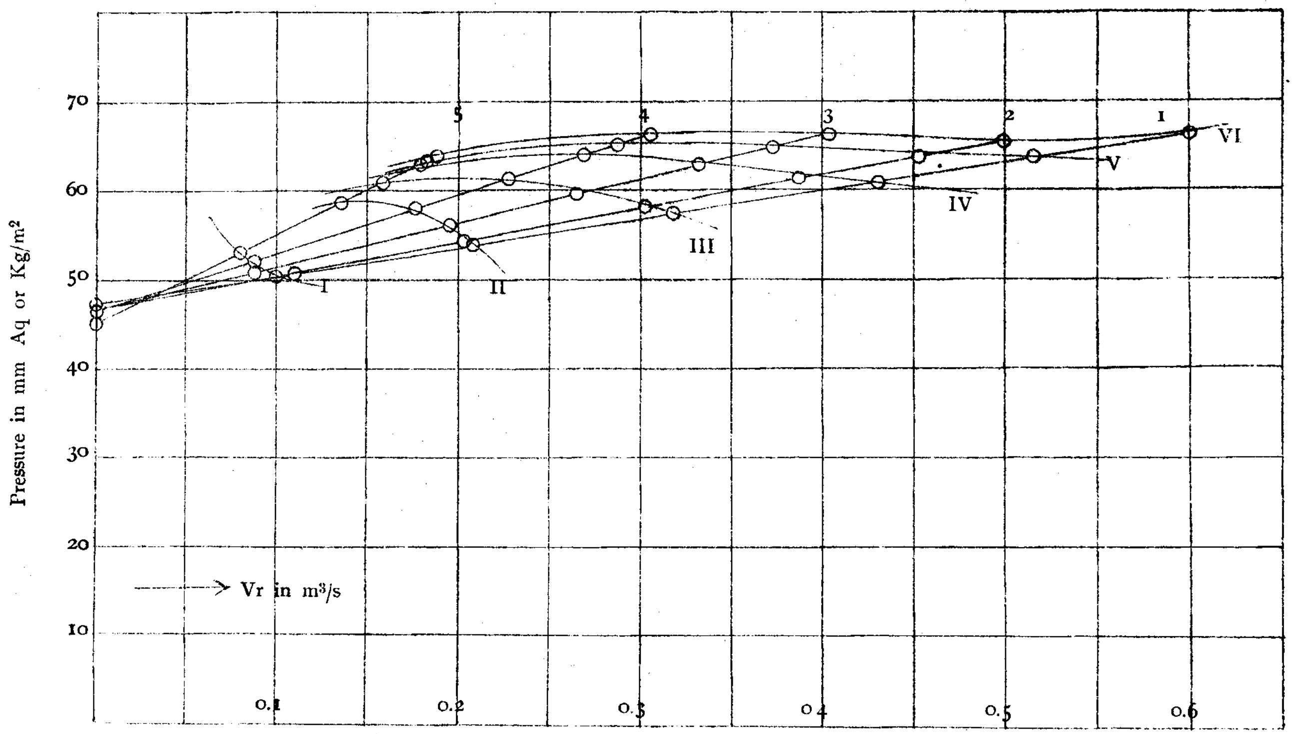

PLATE 11. Summary of curve η_m

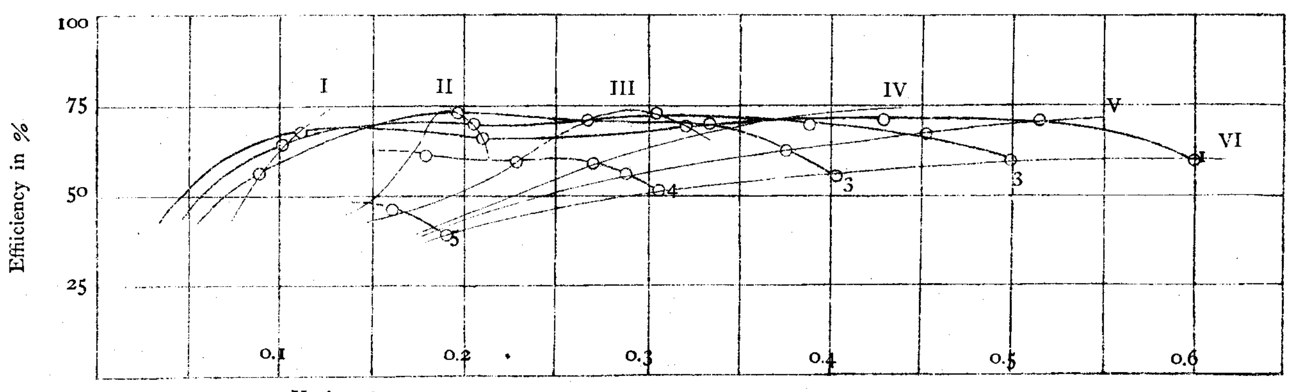

PLATE 12. Summary of corves η'_m

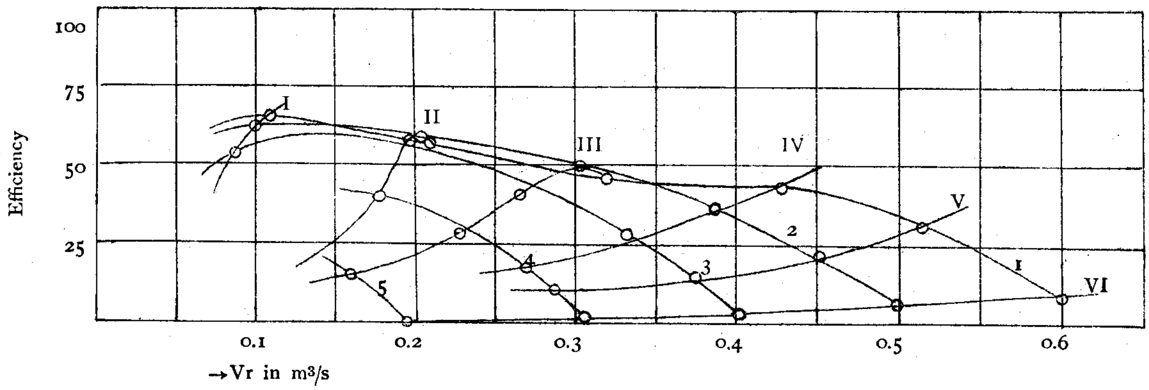

PLATE 13. Summary of curves η_p

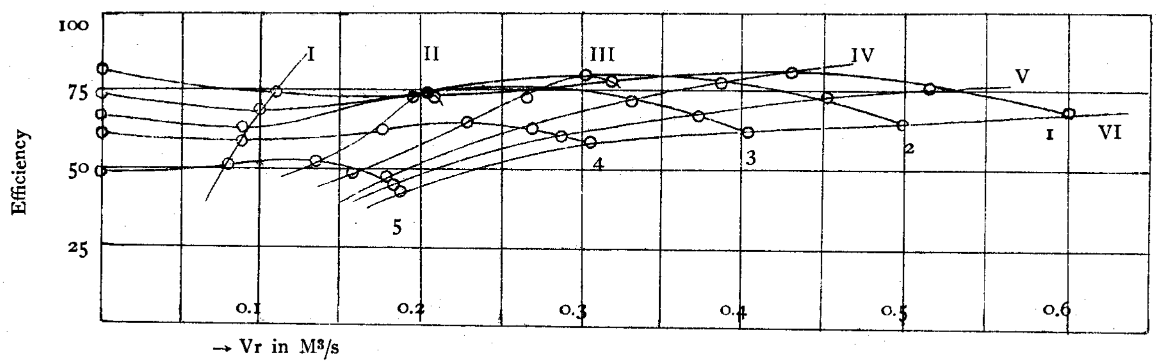

PLATE 14. Summary of curves η'_p

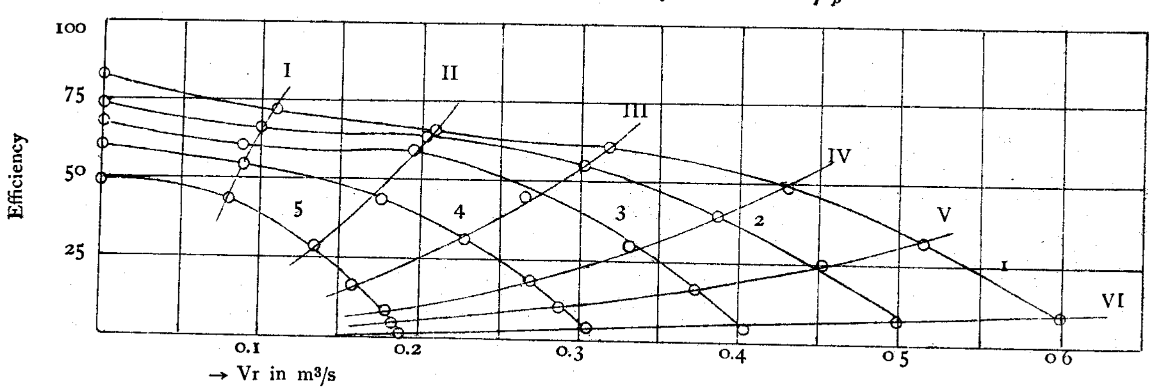

PLATE 15. Summary of curves η_v