THE

A I R E J E C T O R.

by

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ABSTRACT.

This paper deals with the general characteristics of the air ejector. Except for a few published results on the performance of standard units, information on the subject is very scarce, and the experimental lines followed in the investigation have been arranged to provide a systematic examination of the effect of the various dimensions on the stability and efficiency of operation.

Diffuser losses are deduced by analysis from the tests, and the main questions arising in the combining of the operating and induced fluids are discussed. Using a glass-sided diffuser, several aspects of the fluid action are illustrated by the wave formation set up at the nozzle outlet.

THE AIR EJECTOR.

INTRODUCTION.- The ejector may be defined as a pump in which the fluid to be compressed is entrained by one or more jets of fluid which pass through the entrainment space at a very high velocity, the combined fluid then being compressed up to the delivery pressure at the expense of its kinetic energy.

Used for the ejection of air, it is now an indispensable adjunct to the high-efficiency steam turbine unit. With it the very high condenser vacuum necessary is made possible. It is very simple and reliable, can be placed in any convenient position, and does not require special foundations. It involves no moving machinery, requires little attention, and can be operated to its fullest capacity without stressing. According to most authorities, it also compares favourably with the reciprocating pump in the matter of steam consumption, and the only reason why it was not adopted earlier was the lack of interest in the production of a very high vacuum, the special benefits of which were not then capable of exploitation. Steam is generally the operating fluid since it is so readily available, though some claim that the water jet has its advantages when working at high vacua.

From the first the difficulty in designing a suitable diffuser has been a serious obstacle to the development of the ejector. It was soon realised that a single ejector could not be safely employed for compression ratios above 1/8, for the diffuser throat diameter required for starting is considerably greater than that necessary for working under these conditions. A compromise between the two might be made in an effort to ensure stability, but this inevitably results in a reduction of efficiency. The first successful applicat: :ion of the ejector was in conjunction with the ordinary reciprocating pump, as a vacuum augmenter, patented by Parsons. The ejector, which operated on steam, was allowed to work at a suitable compression ratio, and it served to reduce very greatly the size of reciprocating pump necessary. This development naturally led to the adoption of the two-stage steam operated plant, which at once became popular, and remains standard practice to-day. Each unit has a compression ratio well within the limit, so that stability as well as efficiency is assured, if correctly designed.

Several methods have been devised to overcome the difficulty of varying the throat diameter. Perhaps the most general way of doing this is by enlarging it slightly to ensure successful starting, and another method is to allow atmospheric air to enter the diffuser at a suitable point so that the throat section is always filled, but attempts have also been made to vary the diameter by mechanical means.

An intercondenser, in which the main condensate is used as cooling water, is usually introduced between the first and second stages so that the steam consumption in the latter is reduced, and with an after-condenser also using the main condensate, nearly all the heat in the steam is recovered. The same applies to most other arrangements in connection with condensing plants.

Very little is understood about the working of the ejector, although it has been generally accepted that the suction fluid is entrained by friction. This led to the adoption of a cluster of very small bore steam nozzles in the form of a ring, in place of the single nozzle, in order to increase the surface area of the jet. Lately the simple friction theory has lost support, and the advantages of this type of ejector have been questioned. It is suggested that the entrainment of fluid is due to a natural tendency for the jet to over-expand on issuing from the nozzle, and the ejector is so designed that it takes advantage of this property of the jet by allowing a limited quantity of fluid to leak in at the point of lowest pressure. When the ejector is started, the over-expansion causes an influx of surrounding fluid, and the vacuum is built up according to the amount of induced fluid that has to be dealt with. In principle the ejector is thus similar to the recompressing divergent nozzle, with the difference that suitable provision is made for the entrainment and compression of an external fluid.

The question of compression losses in the diffuser has also an important bearing on design, but here again nothing definite is known, though as a result of researches on divergent nozzles by Professors Mellanby and Kerr,[1] and by the author, it is known that the compression of a high speed jet is accompanied by very considerable losses.

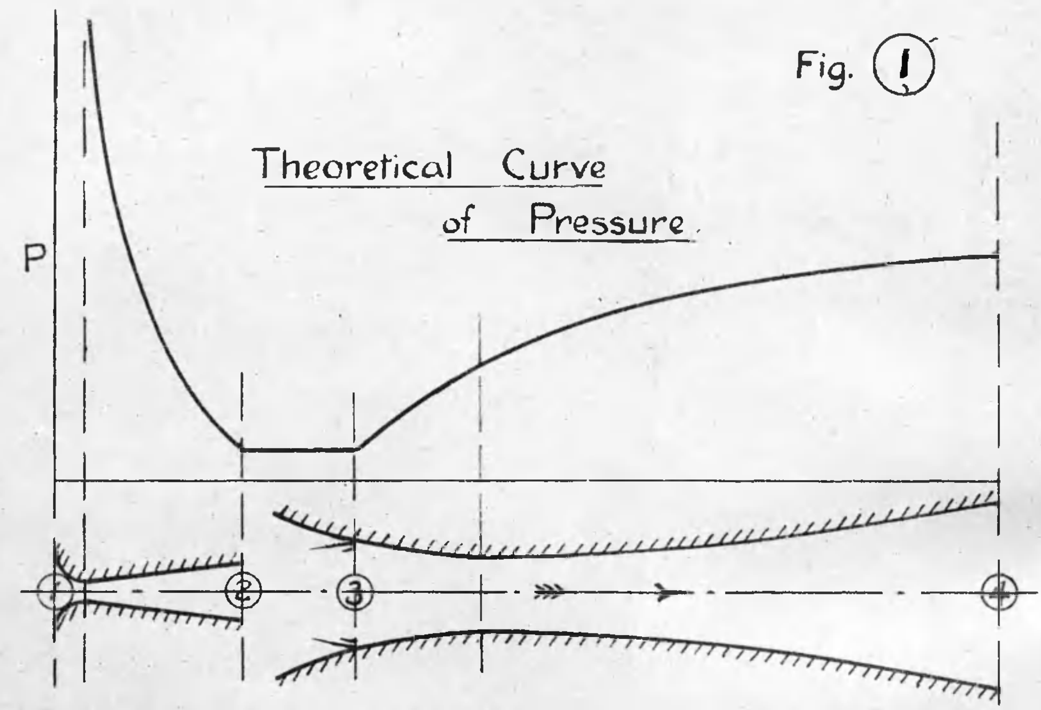
The aim of this research was first of all to make a systematic observation of the effect of the leading dimensions on air-operated ejector working, and hence to find out the most efficient arrangement for operating under given conditions. By working under these "optimum" conditions, it would be possible to eliminate, more or less, the influence of design, and so an analysis of the fluid losses could be attempted. Such an analysis would be quite out of the question with an ejector of fixed dimensions. After assessing the losses which are known to exist, such as the nozzle and diffuser friction losses, and the theoretical energy loss in the combining of the streams, it should be possible to reach some conclusions in connection with the entrainment and compression problems just referred to.

From the experience gained in experiments on high velocity jets by the author, it was also thought likely that photographs of the jet in the entrainment space would prove of some interest.

THEORY OF THE EJECTOR.- At this stage, when the knowledge of fluid flow is still very incomplete, it is only possible to apply the fundamental theory to the various phases of ejector action. This includes the usual assumption that the stream is homogeneous in section.

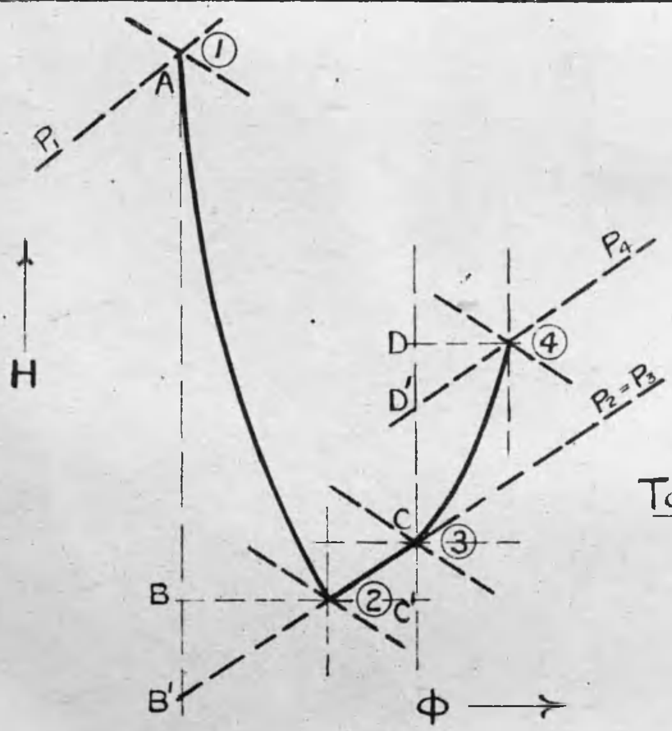
Air is used both as the operating and the suction fluid in the investigation, for the reason that with a mixture of fluids in the diffuser a close analysis would be out of the question. The theory

Fig. 1



Theoretical Curve of Pressure.

Fig. 2



Total Heat – Entropy Chart.

outlined therefore deals with air flow, though for steam as the operating fluid the conditions are much the same when the proper index of expansion is taken. The suction air is assumed to be dry, which is very nearly the case for atmospheric air, but in a condenser plant the conditions are altered, of course, by the presence of water vapour to the saturation point.

The following symbols for the properties of the air are used:-

P = absolute pressure in lb. per sq. in.

V = volume in cubic ft. per lb.

T = absolute temperature in ^{o}F.

v = velocity in ft. per sec.

M = flow of operating air in lb. per sec.

M' = flow of suction air in lb. per sec.

E = kinetic energy in ft. lb. per lb. operating air.

r = pressure ratio referred to the nozzle supply pressure.

r_a = pressure ratio referred to atmospheric pressure.

The suffix (1) denotes the nozzle supply,

(2) the nozzle jet before combining,

(3) the combined stream before compressing,

(4) the discharge stream.

The various stages can be followed out in the pressure and entropy curves sketched in Figs. 1 and 2.

Expansion Stage (1) - (2). For the suitable expansion of the fluid a nozzle of the convergent divergent type is required. If the inlet velocity to the nozzle be neglected (i.e., when the ratio of supply pipe to nozzle throat diameter is sufficiently great) the kinetic energy at the outlet for adiabatic expansion is given by

$$E_2 = \frac{144\,n}{n-1} P_1 V_1 \left\{1 - r^{\frac{n-1}{n}}\right\} \quad - - - - (1)$$

from standard nozzle theory, where n is the index of expansion. It has to be modified to allow for friction losses, which are represented by the factor "k_n". This factor was introduced by Professors Mellanby and Kerr,[2] and is equivalent to an energy loss just as $\left(1 - r^{\frac{n-1}{n}}\right)$ is equivalent to the energy in adiabatic expansion. When n is given its value of 1·40 for air - assuming constant specific heat values as being sufficient for the purposes of the investigation - and substitutions are made,

$$E_2 = 186\,T_1 \left(1 - K_n - r^{0{\cdot}286}\right) \quad - - - - - (2)$$

The loss term is obtained from the dimensional equation.

$$K_n = c\int_0^l \frac{p}{A}\left(1 - K_n - r^{\frac{n-1}{n}}\right) dx \quad - - - - - - (3)$$

where $\frac{A}{p}$ is the hydraulic mean depth at a point and dx an element of length. The constant c has a valuefor machined nozzles of about 0·005. Values of k_n, the loss up to a point, are assumed for the moment in the integrand, and the curve of $\frac{p}{A}\left(1 - K_n - r^{\frac{n-1}{n}}\right)$ is integrated for the whole length of the nozzle, as the jet is in contact with the walls throughout its length. A small factor, estimated at 0·004, is added to include entrance losses. A closer approximation to k_n is obtained from this integration, so that a repetition of the process will enable the loss to be determined with greater accuracy.

On the H/ϕ chart, (1) - (2) represents the expansion stage, AB' the energy in adiabatic expansion, and AB that liberated in the actual expansion, so that the ratio AB/AB' represents the nozzle efficiency.

Entrainment Stage, (2) - (3). From the principle of the conservation of momentum, it is known that the sum of the momenta of the jet and of the air induced by it is equal to the momentum of the combined stream, in any given direction. This assumes that there is no change in pressure during the entrainment, and no shock losses due to the uniting of the streams or to reaction from the sides of the diffuser. Since the amount of induced air is small when compared with the area of entry into the diffuser, its momentum is negligible, so that the momentum equation can be put

$$Mv_2 = (M+M')v_3 \qquad - - - - (4)$$

Since E_3 is [illegible] operating [illegible]

$$\therefore \frac{v_3^2}{2g} = \left(\frac{M}{M+M'}\right)^2 \frac{v_2^2}{2g} \qquad - - - - (5)$$

$$\left(\frac{M+M'}{M} \frac{v_3^2}{2g}\right) = \frac{M}{M+M'} \cdot \left(\frac{v_2^2}{2g}\right)$$

$$\therefore E_3 = \frac{M}{M+M'} E_2 \qquad - - - - (6)$$

This is an expression for the kinetic energy of the jet which is available for compressing the combined stream. It is evident from this equation that there is a loss of kinetic energy during entrain: :ment, the magnitude of the loss being dependent on the ratio $\frac{M}{M+M'}$, and it goes to reheat the combined stream.

From the state of the jet fluid and that of the induced fluid, before combining, can be deduced the state of a mixture of the two, with the inclusion of the reheat equivalent. This is represented on the H/ϕ chart by adding the reheat value to the total heat of a normal mixture of the two fluids to obtain the point (3). CC' is equivalent to the reheat of the operating fluid during the constant pressure stage (2) - (3).

Compression Stage, (3) - (4). At the point (3) the combined stream strikes the wall of the diffuser, and compression starts. If the velocity of the stream is above the critical, compression requires to take place in a convergent channel until the critical velocity is reached, and thereafter in a divergent channel. In the case where the velocity of the stream is below the critical at the start of compression, the convergent port: :ion can be dispensed with, as far as theoretical requirements go. The first case is, of course, just the reverse of the expansion in the convergent-divergent nozzle, and the second of that in the convergent nozzle. The areas at the various sections could be calculated in the usual manner from the equation of continuity of flow, provided that the flow quantity, the velocity and the specific volume were known.

The kinetic energy of the combined stream absorbed by adiabatic compression up to atmospheric pressure is given by the equation

$$W = \frac{M+M'}{M} \cdot \frac{144n}{n-1} P_3 V_3 \left(\frac{1 - r_a^{\frac{n-1}{n}}}{r_a^{\frac{n-1}{n}}} \right) \quad - - - (7)$$

per lb. operating air, so that when substitutions are made

$$W = \frac{M+M'}{M} \cdot 186\, T_3 \left(\frac{1 - r_a^{0.286}}{r_a^{0.286}} \right). \quad - - - (8)$$

The diffuser friction loss term k_d is calculated in the same manner as that of the nozzle, by integrating the energy curve. Estimation of the energy at a point in the compression can be made only approximately, however, as it consists of the kinetic energy necessary for compression from the point up to atmospheric pressure, together with the residual kinetic energy in the stream, and that required to overcome the friction and compression losses incurred during compression. The residual kinetic energy, which is generally very small, can be calculated from reaction measurements, but the compression loss remains unknown, so that it is not possible to determine the friction loss exactly. But the P/A term in the integrand is much less than that for nozzle flow, so that the diffuser friction loss is a comparatively small one, and an approximation is consequently quite sufficient. Hence the additional energy absorbed by friction, per lb. operating air

$$= \frac{M+M'}{M} \cdot \frac{186\, T_3}{r_a^{0.286}} \cdot k_d \quad - - - - (9)$$

The residual kinetic energy in the stream is best obtained by measurement of the reaction of the

Fig. 3 — Experimental Air Ejector.

ejector. If R is the reaction in lb., then

$$R = (M + M')\,v_4/g \qquad - - - - - (10)$$

$$\text{i.e. } E_4 = \frac{gR^2}{2M(M+M')} \qquad - - - - - - (11)$$

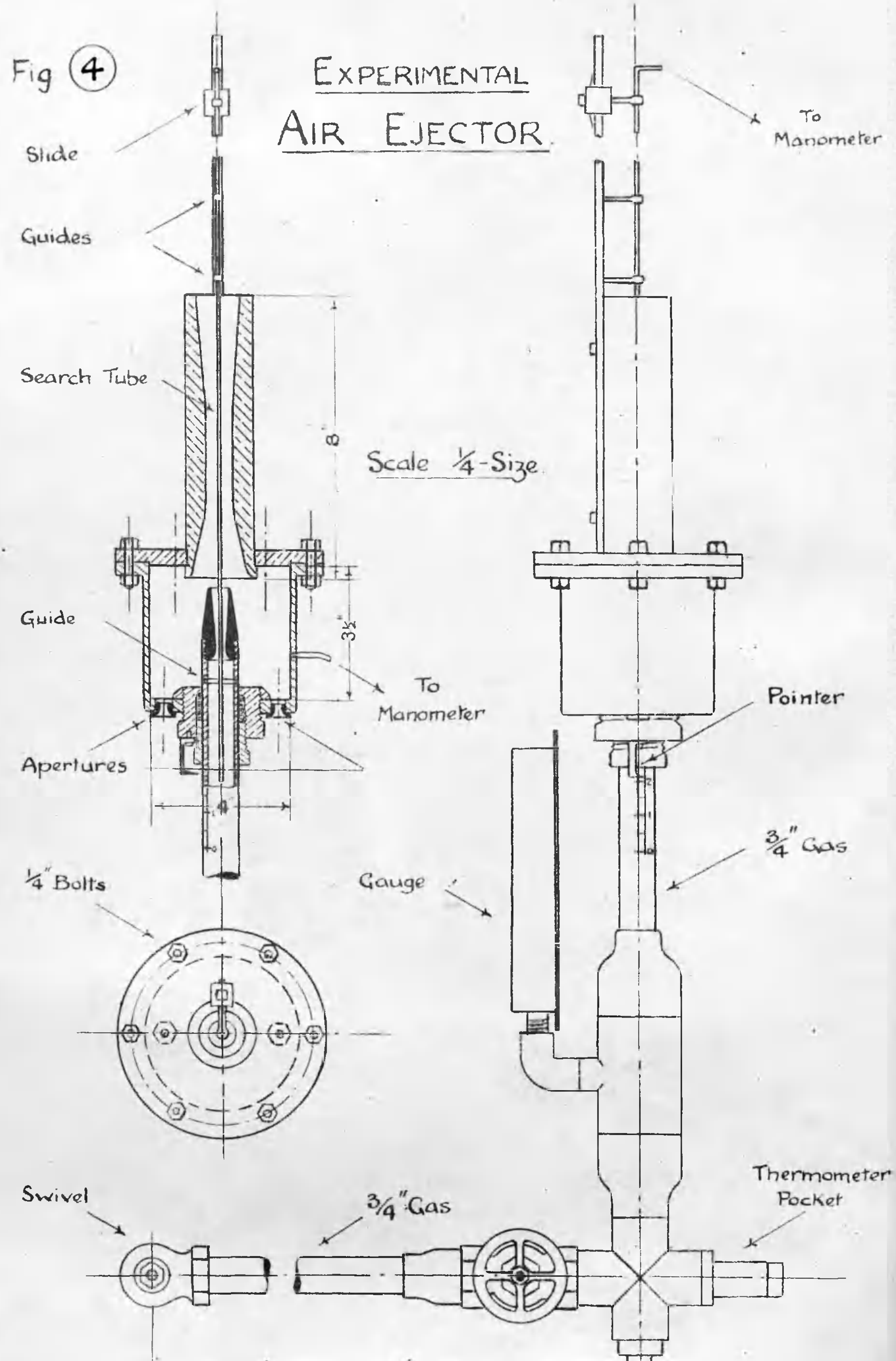
The work done in compression, together with the friction loss and the residual kinetic energy E_4 should be equal to the available kinetic energy E_3 in equation (6) if there are no other losses besides friction in the diffuser, so that any difference will be equivalent to the losses unaccounted for.

On the H/ϕ chart, (3) - (4) represents the compression stage, CD' the energy required for adiabatic compression, and CD that actually required for compression, so that the ratio CD'/CD represents the efficiency of compression.

<u>EXPERIMENTAL PLANT</u>.- The air supply for the tests was obtained from a two-stage compressor, delivery being made at 100 lb. per sq. in. gauge to a large receiving tank.

In Fig. 5 are given the details of the straight circular nozzles of machined gun-metal used in the tests. Since they were intended to cover a wide range of expansions, several were required, and each is to standard as far as entry curve, throat diameter, and length are concerned. It is well known, and will be demonstrated in the paper on nozzle losses, that a nozzle which over-expands loses efficiency, so that to accommodate the range of expansions, the nozzles, three in number, have a different divergent taper. If, Fig 4

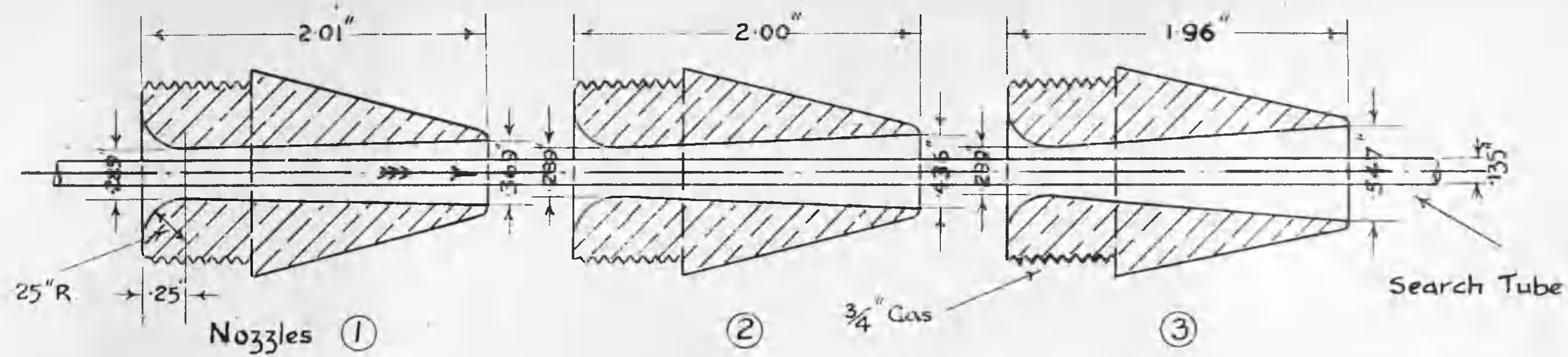
EXPERIMENTAL AIR EJECTOR

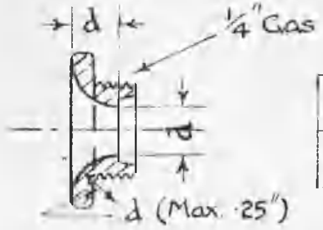


then, for each test the nearest under-expanding nozzle is chosen, the losses would be negligible, for under-expansion must be considerable before there is an appreciable loss. Tests made between an under-expanding and a correctly designed nozzle verified this statement. The outlet area corresponding to a given ratio was calculated in the usual way from theory, with the necessary allowance for an $\frac{1}{8}$" search tube which passes through the nozzle.

The question of nozzle outlet face, and its possible effect on the induction of the air is one of considerable interest, and will be referred to later. It is obvious that the tapering of the nozzle on the outside to the minimum thickness of metal would provide the most favourable flow passage for the induced air. The nozzles were screwed to fit a $\frac{3}{4}$" pipe about a foot in length, which was machined and ground to fit a gland in the combining chamber - a circular box flanged to receive the diffuser plate (see Fig.4). The diffuser was screwed into this plate, and the whole could slide along the nozzle supply pipe, on which a scale of inches was marked off. The apparatus was mounted on a long horizontal arm pivoted at a swivel joint, and through this passed the air supply from the compressor receiving tank. This enabled the ejector to move freely in a vertical direction, and so the reaction of the air leaving the diffuser could be measured on a balance, just as in the nozzle reaction tests described in the paper on divergent nozzle losses. A pressure gauge and thermometer to determine the initial state of the operating air were included.

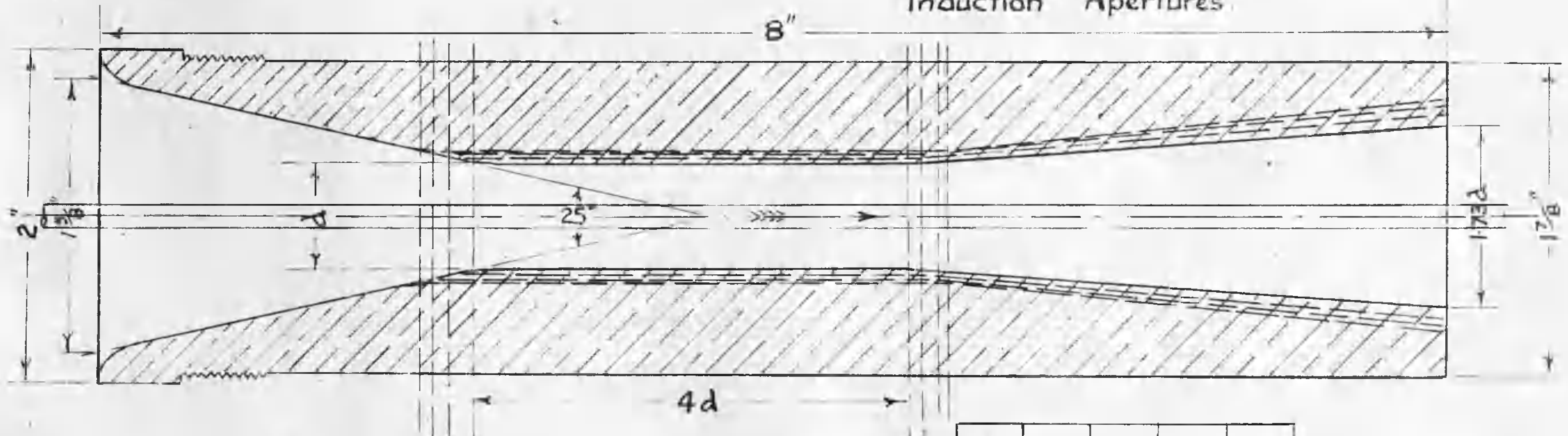
Fig. 5 Details of Experimental Air Ejector





Ap.	1	2	3	4	5	6
d (in)	·0693	·1245	·1957	·2505	·299	·352

Induction Apertures



Diffusers A – D

Diff	A	B	C	D
d (in.)	·629	·694	·750	·788

Measurement of the induced or suction air was effected by means of a series of apertures leading into the combining chamber. The apertures took the form of convergent nozzles, and their proport: :ions are given in Fig. 5. The possibility of their position in the chamber having an effect on the working of the ejector was considered, and led to an investigat: :ion being carried out under fixed conditions with the apertures in the various positions illustrated in Fig. 6. In (b) the positions are reversed, in (c) the air is drawn in through a single aperture which gives the same flow quantity as the other two combined, and in (d) the incoming jets are broken up by a baffle plate placed as shown. The tests showed that the capacity of the jet for entraining the surrounding fluid is quite independent of the aperture positions, with the except: :ion of (c), the one-sided effect of which slightly impairs the performance of the ejector. This goes to prove that the ejector action is practically independent of the design of the chamber or the position of the induction pipe.

As has been pointed out when dealing with the theory of the ejector, the diffuser design should be followed out on the same lines as that of the nozzle, but a design based on theory alone does not take into account the special requirements of the entrainment action and the indeterminate nature of the characteristics of the combined jet. It has been said that theory requires a convergent channel, a throat or minimum area section, followed by a divergent channel. The first problem, therefore, is to choose the most suitable throat

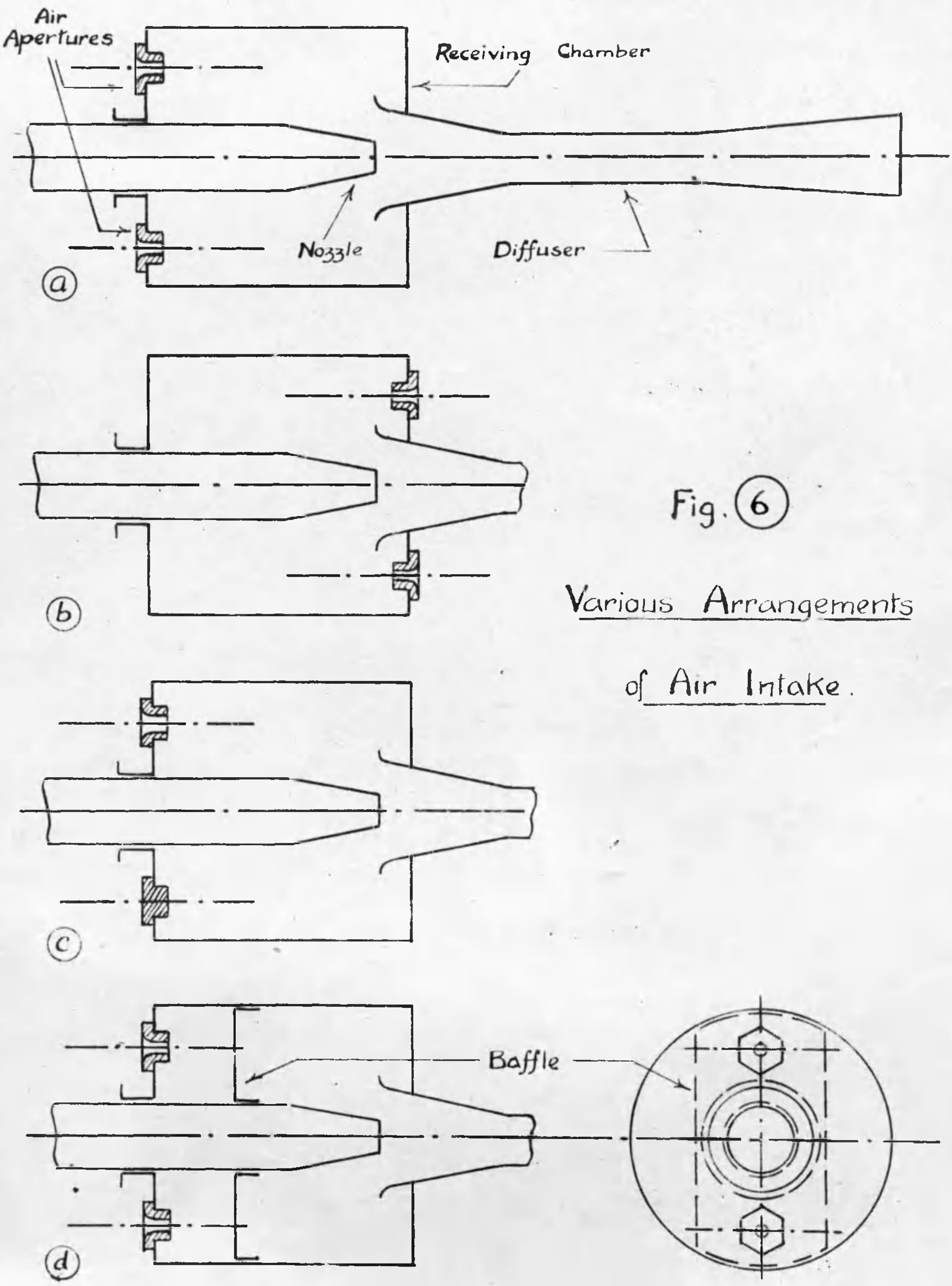


Fig. 6

Various Arrangements of Air Intake.

area. Very often it is erroneously calculated by applying the equation of continuity of flow to the fluid conditions just before compression commences, i.e.

$$A_3 = \frac{144\,(M+M')\,V_3}{v_3} \qquad \text{(in sq. in.)}$$

It is obvious that unless the velocity at (3) in Fig. 1 is below the critical this cannot apply to the section of minimum area.

Owing to the diffuser losses and other unknown factors a direct calculation of the true throat diameter is out of the question, so that a series of tests with diffusers of different diameters is probably the best solution of the problem. Before these tests were carried out, it was necessary to choose the most suitable form of diffuser, i.e., the angle of convergence, the extension of the minimum area section, if any, and the subsequent divergence. Experience seems to have demonstrated that 25° is the best entrance angle, and it is general practice to extend the throat section a few diameters or else make the divergence start very gradually.

The critical importance of the throat area was recognised immediately, so that, as there is no practical means of making it variable, the whole series of tests run on the ejector depended on the choice of a set of diffusers of suitable diameters. Later experience showed that four diffuser diameters provided sufficient optimum values for the series of tests.

Fig. 5 gives the various diffuser sizes. The diffusers were bored in turn out of a 2" gun metal cylinder, so that in each case the throat section was extended to four diameters in length, and the outlet area was three times that at the throat.

Pressures along the axis of the nozzle and diffuser were determined by means of a simple search-tube apparatus. A slide attached to the end of the tube moved up and down a graduated guide rod, which was screwed to the side of the diffuser. Comparative rigidity was obtained by fixing guides in such positions that no interference with the air stream resulted. A 1/32" hole was drilled in the side of the tube, and connection was made to a mercury manometer. So that the nozzle area might remain unchanged by the movement of the tube, the latter required to be of exactly uniform cross section, and so long, that when the hole was opposite the diffuser outlet the tube remained in the lower guide.

<u>METHOD OF TESTING</u>.- Before proceeding with the tests, it was ascertained that for a given nozzle inlet pressure the temperature became steady as soon as the compressor had heated up to normal running conditions. This enabled a curve of corresponding inlet pressures and temperatures to be made, and the values were substituted in the flow formula for expansion below the critical ratio:-

$$M = \frac{\cdot 534\, P_1 A_t}{\sqrt{T_1}}$$

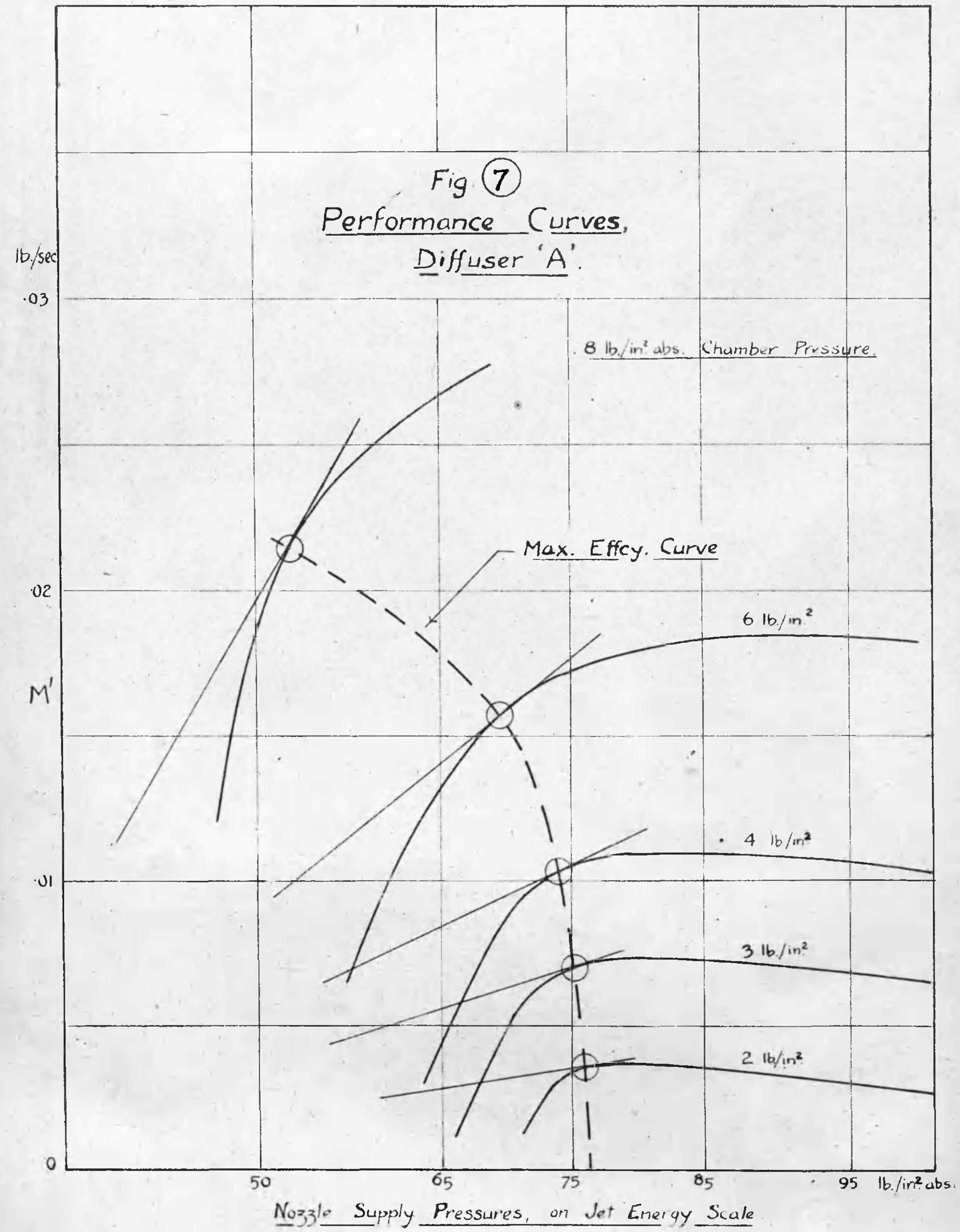


Fig 7
Performance Curves,
Diffuser 'A'.
lb./sec
·03
·02
·01
0
M'
8 lb./in.² abs. Chamber Pressure.
Max. Effcy. Curve
6 lb./in.²
4 lb/in²
3 lb./in²
2 lb/in²
50
65
75
85
95
lb./in.² abs.
Nozzle Supply Pressures, on Jet Energy Scale

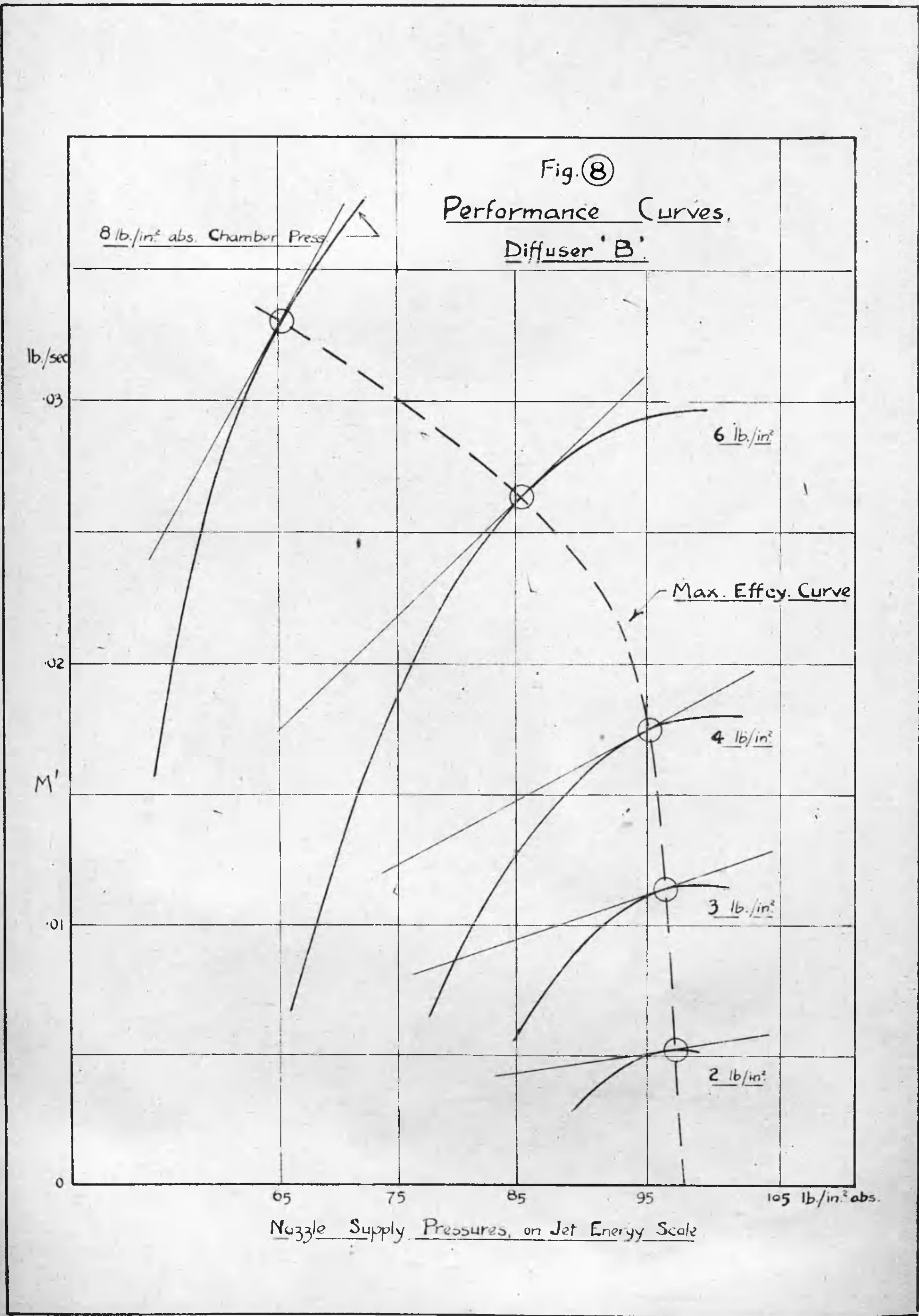


Fig. 8
Performance Curves.
Diffuser 'B'.
8 lb./in.² abs. Chamber Press.
6 lb./in²
Max. Effcy. Curve
4 lb/in²
3 lb./in²
2 lb/in²
lb./sec
·03
·02
M'
·01
0
65
75
85
95
105 lb./in.² abs.
Nozzle Supply Pressures, on Jet Energy Scale

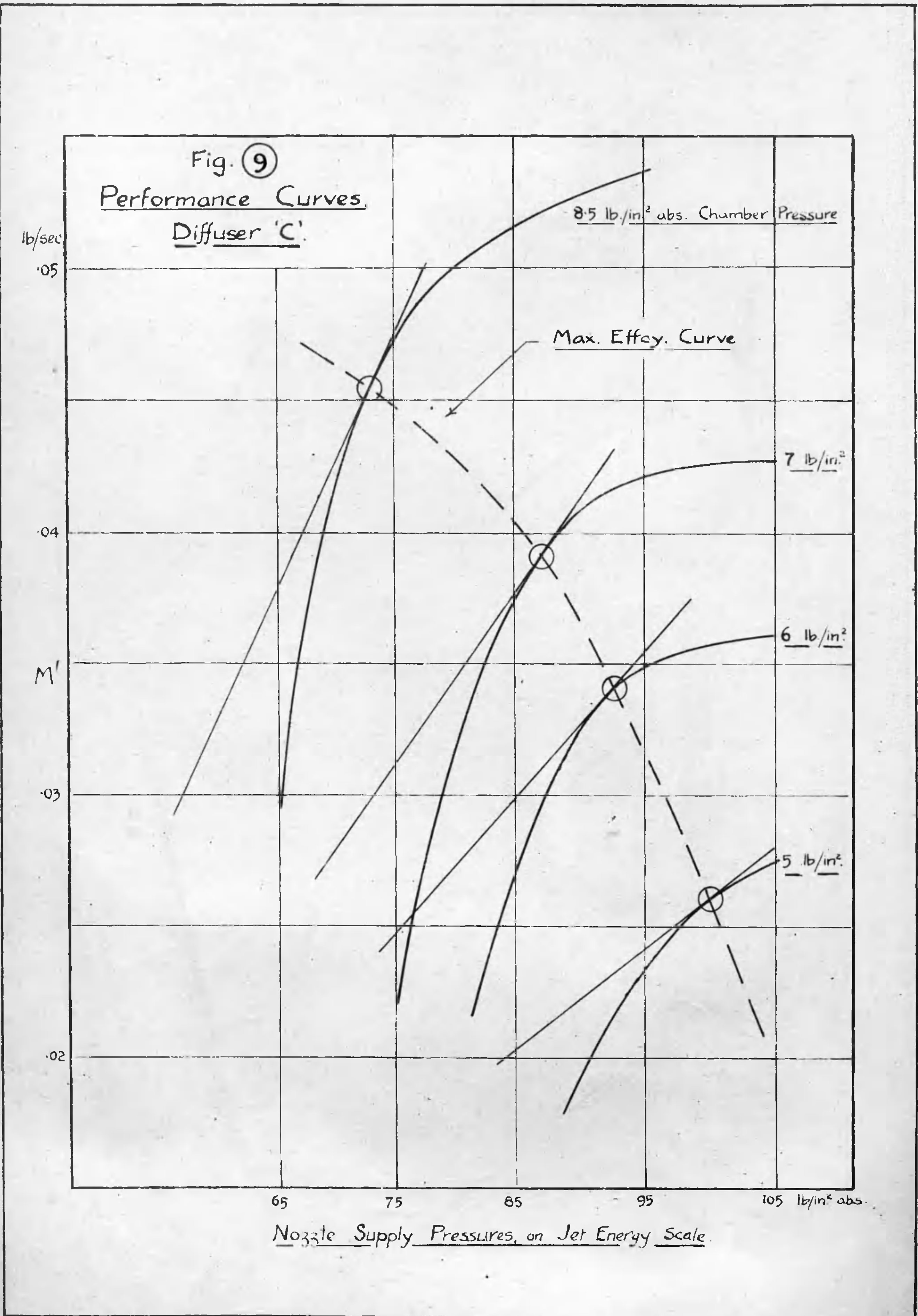


Fig. 9
Performance Curves, Diffuser 'C'.
lb/sec
·05
·04
M'
·03
·02
8·5 lb./in² abs. Chamber Pressure
7 lb/in²
6 lb/in²
5 lb/in²
Max. Effcy. Curve
65
75
85
95
105
lb/in² abs.
Nozzle Supply Pressures, on Jet Energy Scale

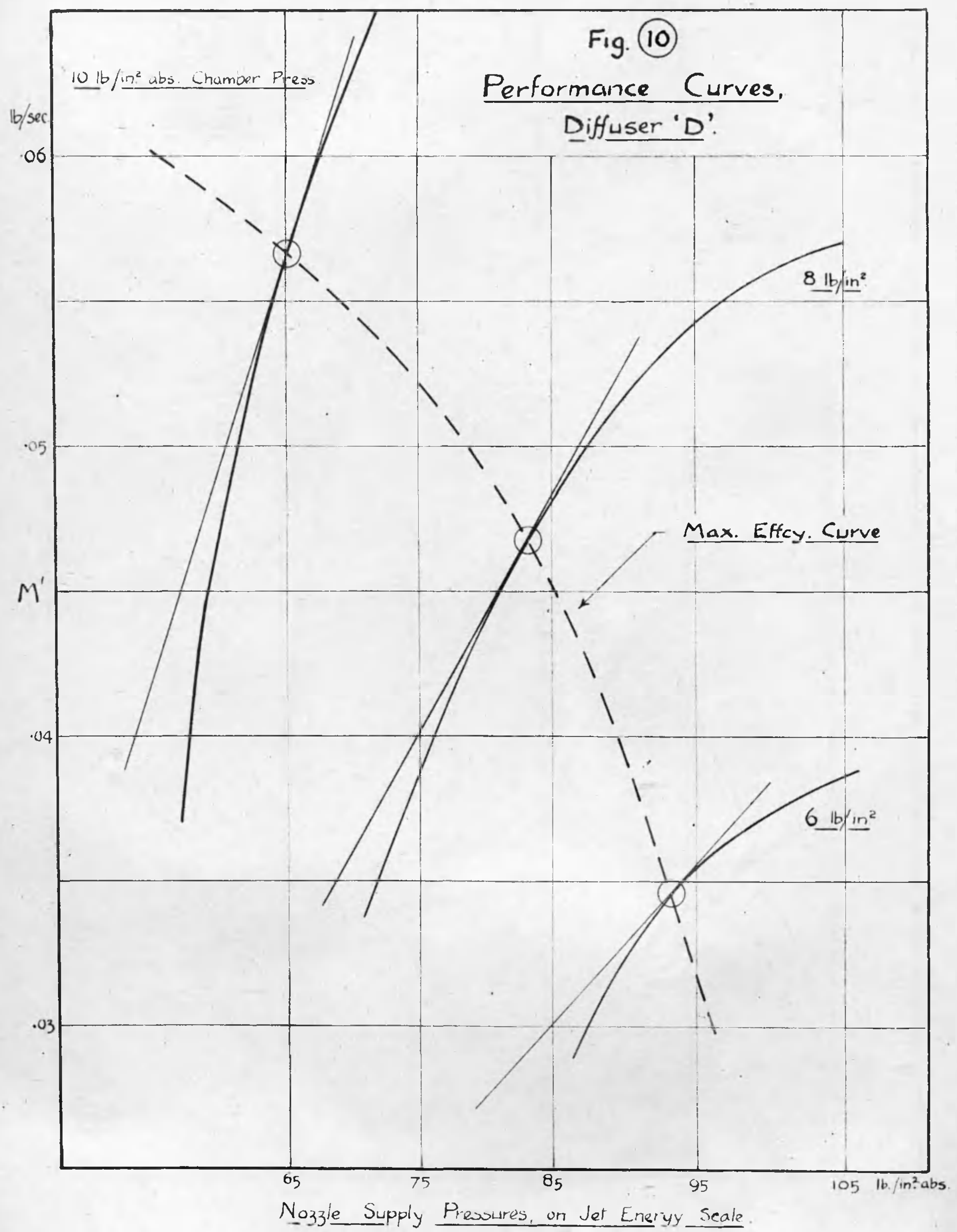


Fig. 10
Performance Curves,
Diffuser 'D'.
10 lb/in² abs. Chamber Press
lb/sec
·06
·05
·04
·03
M'
8 lb/in²
Max. Effcy. Curve
6 lb/in²
65
75
85
95
105 lb./in² abs.
Nozzle Supply Pressures, on Jet Energy Scale

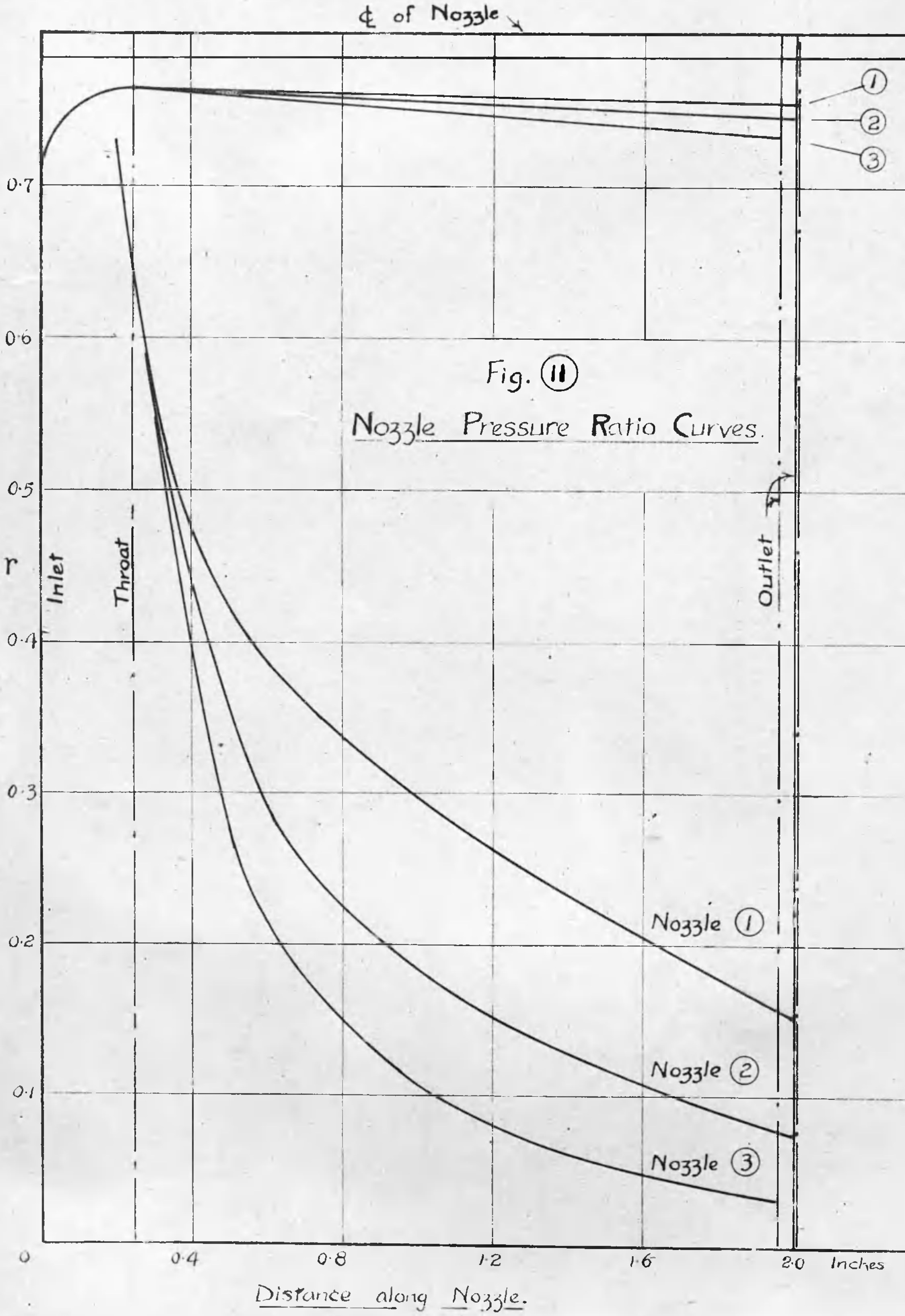
where A_t is the throat area of the nozzle in sq.in. With a discharge coefficient of 0·97, a curve of nozzle discharge on a pressure base was obtained. The coefficient was deduced from the results of tests on nozzles of different length and taper (which will be included in the author's paper on divergent nozzle losses.) Since the greatest taper is only 10^{0}, the one coefficient is suitable for the three nozzles. The coefficient should be constant for all outlet ratios below the critical, and this is found to be very nearly the case, from tests on steam nozzles, down to very low ratios.

Also, by substituting in the formula for the energy in adiabatic expansion to atmospheric pressure,

$$E = 186\,T_1 \left(1 - r^{0\cdot 286}\right),$$

a curve of the jet kinetic energy theoretically available in expansion from the nozzle inlet to the diffuser outlet pressure was drawn.

The first series of tests had as its object the determination of the most suitable operating conditions for each diffuser diameter. With the first diffuser size, readings of the chamber pressure were taken, at a diffuser distance from the nozzle which gave the best results, for a range of nozzle supply pressures and suction apertures. The nozzle corresponding to the particular expansion ratio in each test was employed. The chamber pressures were first of all plotted on a base of suction quantity, and curves of constant nozzle pressure drawn through the points. From



℄ of Nozzle
Fig. 11
Nozzle Pressure Ratio Curves.
0.7
0.6
0.5
r
0.4
0.3
0.2
0.1
0
0.4
0.8
1.2
1.6
2.0
Inches
Distance along Nozzle.
Inlet
Throat
Outlet
Nozzle 1
Nozzle 2
Nozzle 3
1
2
3

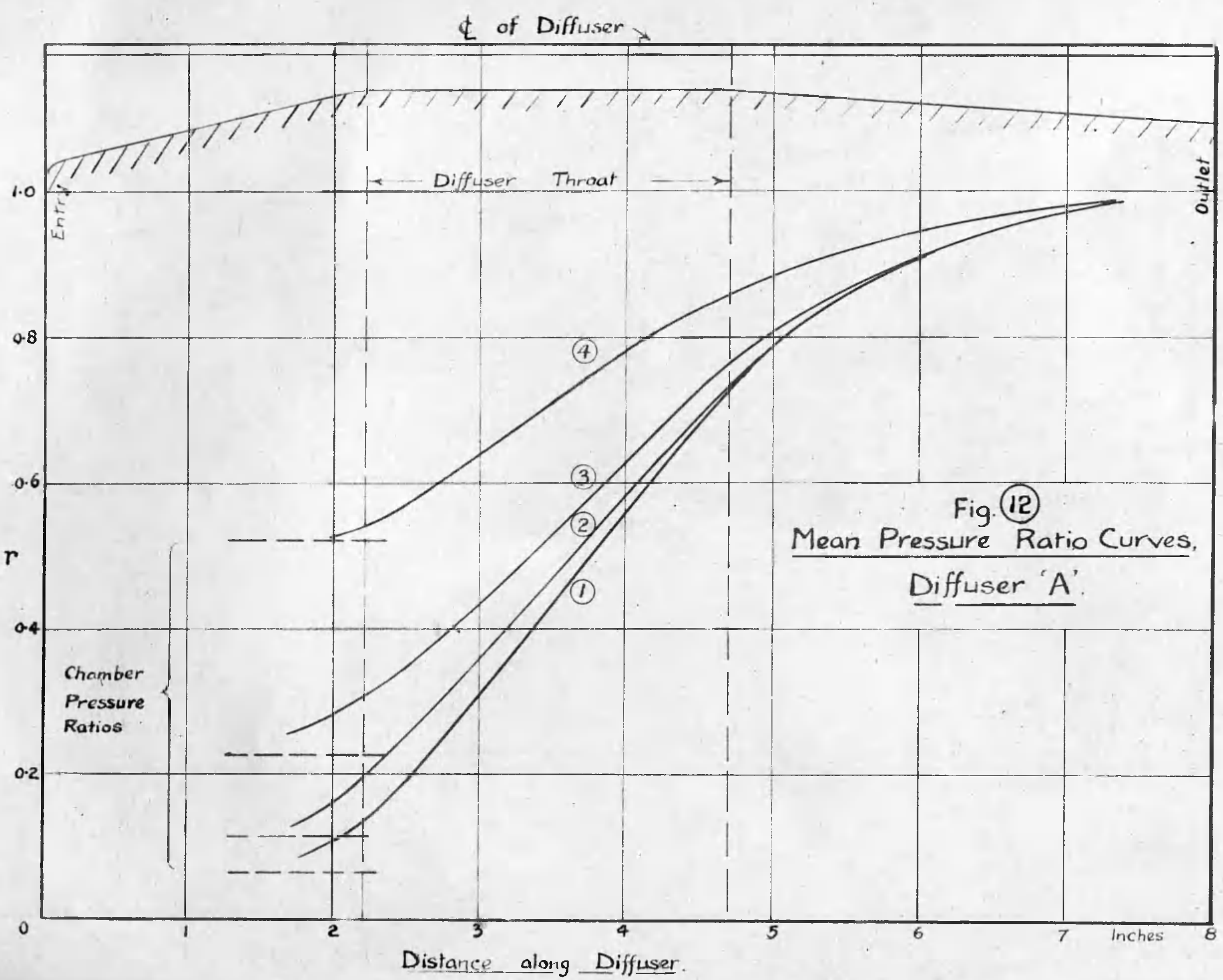
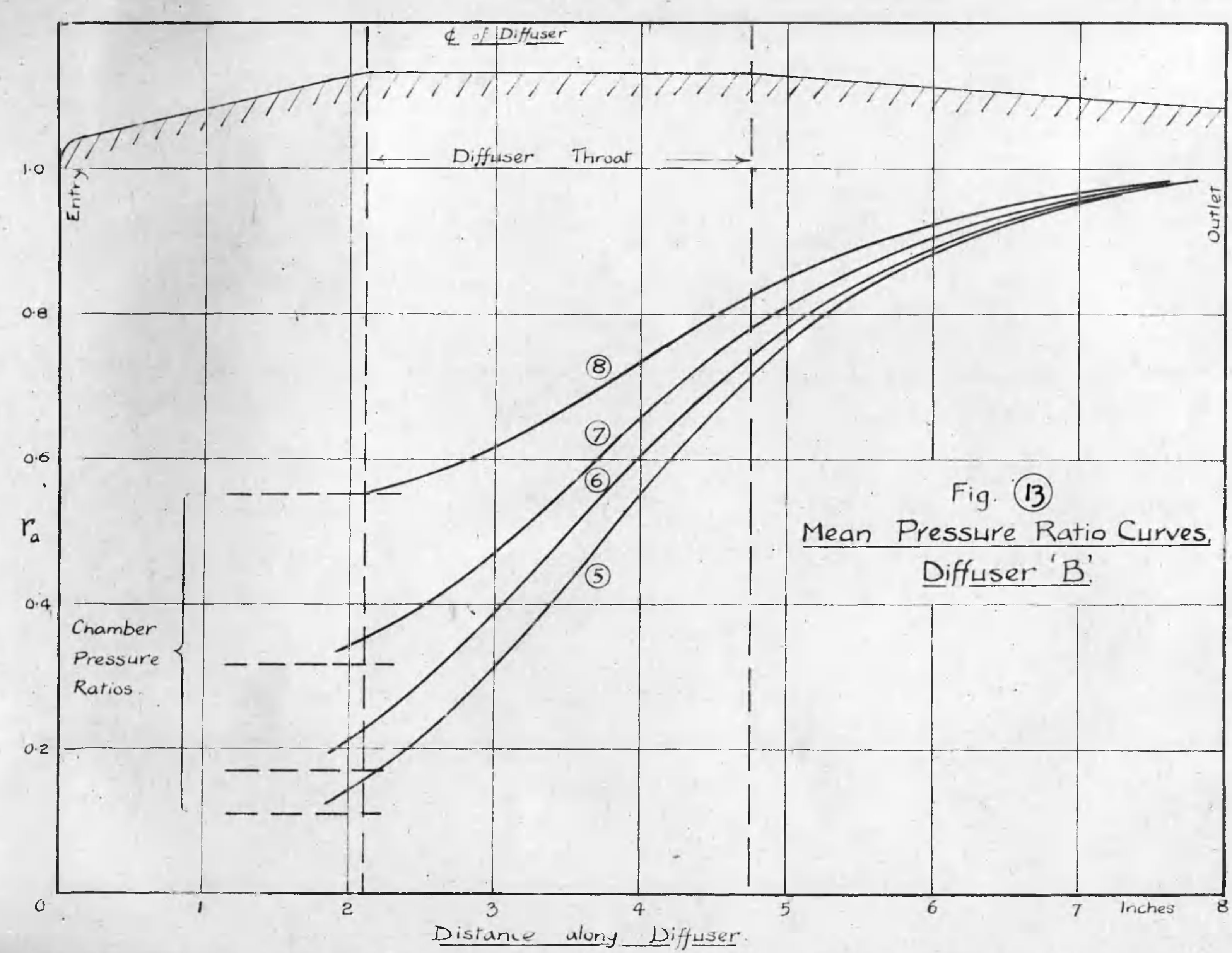


Fig. 12
Mean Pressure Ratio Curves,
Diffuser 'A'.



℄ of Diffuser
Diffuser Throat
Entry
Outlet
1·0
0·8
0·6
r_a
0·4
0·2
0
Chamber Pressure Ratios.
8
7
6
5
1
2
3
4
5
6
7
8
Inches
Distance along Diffuser.
Fig. 13
Mean Pressure Ratio Curves,
Diffuser 'B'.

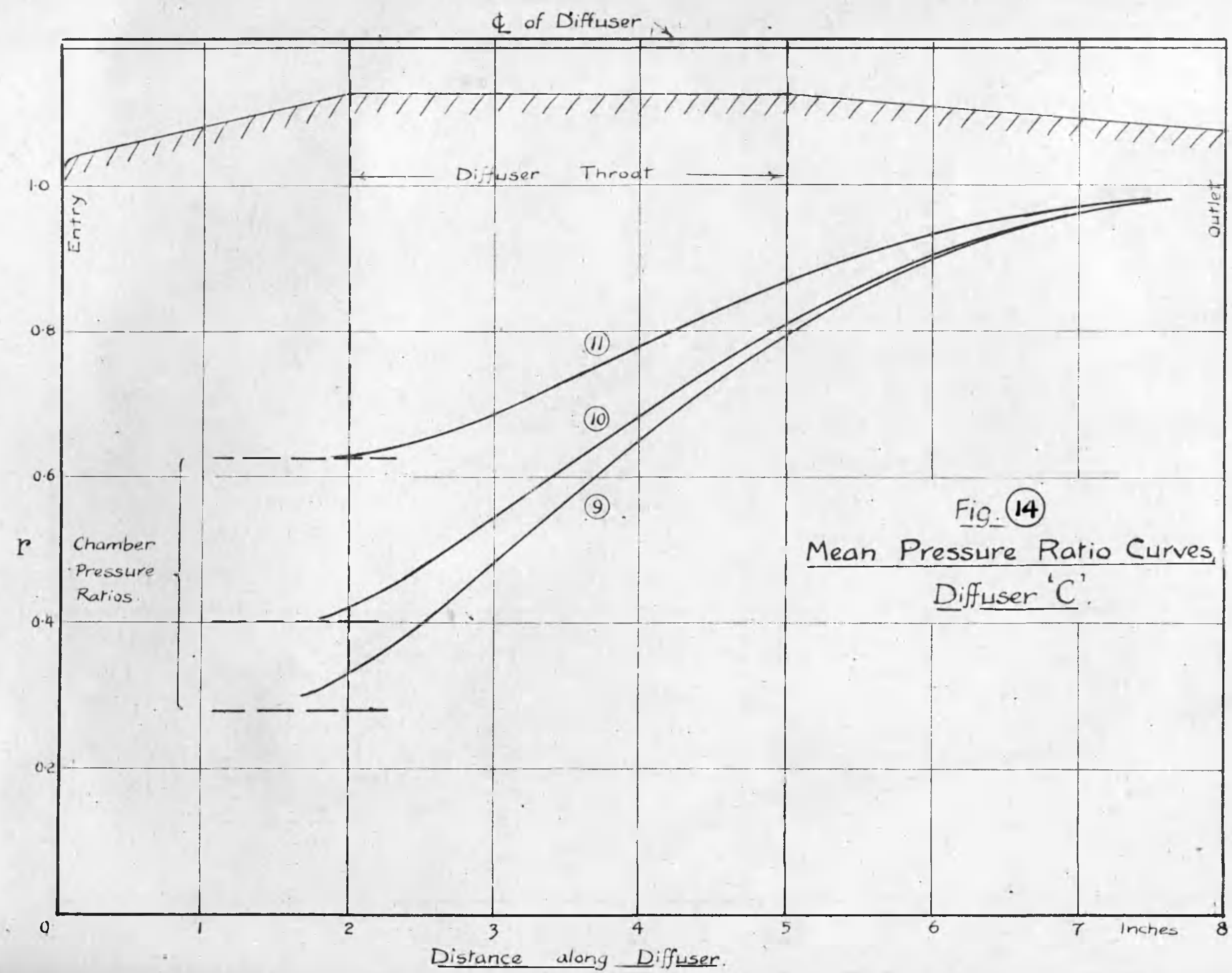


Fig. 14
Mean Pressure Ratio Curves.
Diffuser 'C'.

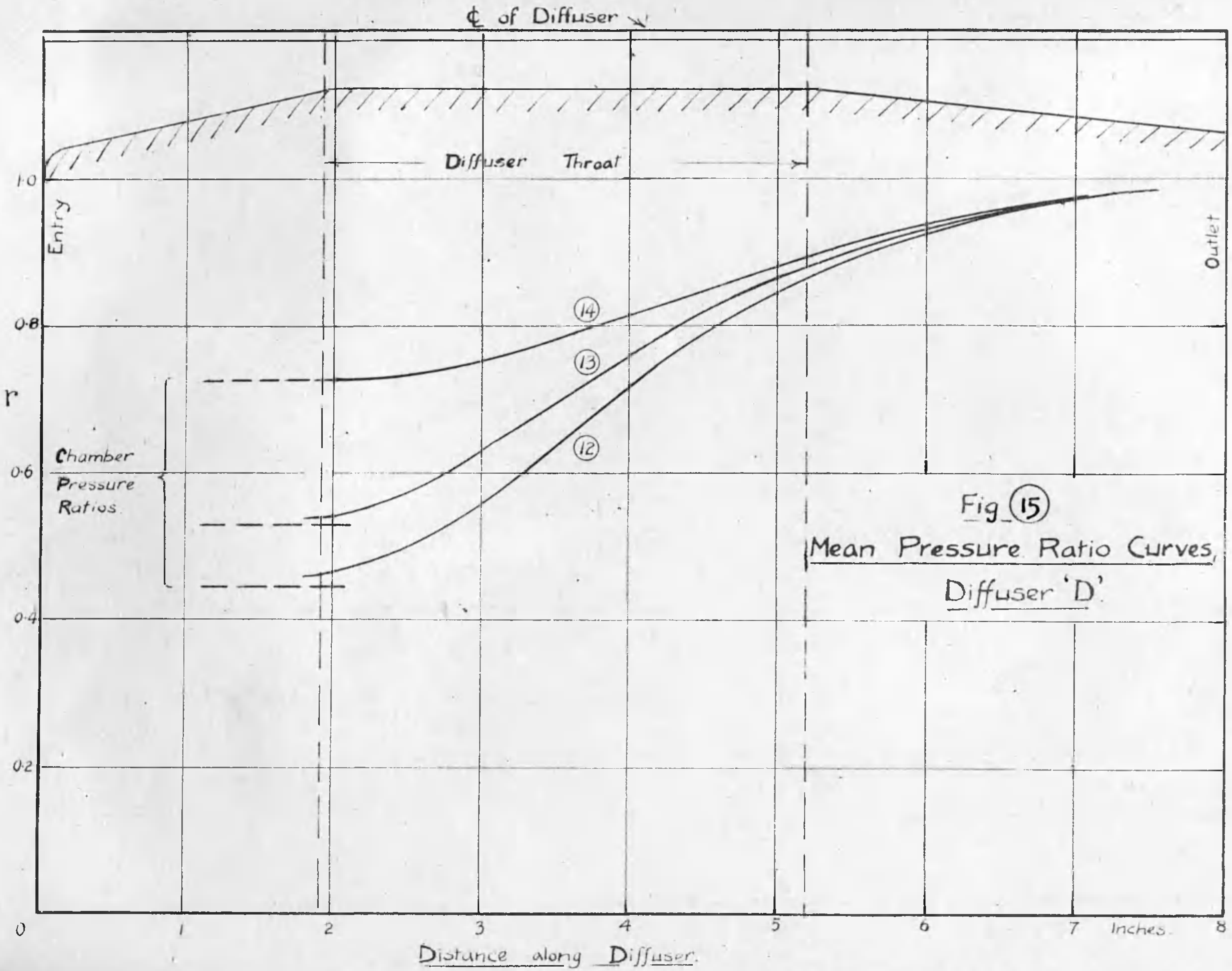


Fig 15
Mean Pressure Ratio Curves,
Diffuser 'D'

these were derived curves (Fig. 7) of suction quantity at constant chamber pressure, on a base of the energy expended by the operating air in expansion from the supply pressure to atmospheric pressure. The energy scale is obtained from the curve of jet energy against nozzle pressure referred to previously. It is obvious that the points of contact of tangents to the curves drawn from the origin give readings of the suction quantity and nozzle pressure which represent the most efficient conditions of working for each chamber pressure, i.e., the maximum ratio of work done to energy supplied. This corresponds to the conditions aimed at in practice, for the operating vacuum (or chamber pressure) is always specified first, and the best working condition determined after.

A maximum efficiency curve drawn through the points of contact enabled the nozzle supply pressures corresponding to the suction quantities given by the various apertures to be read off. Under these conditions of maximum efficiency the ejector was then operated, and the data required for an analysis obtained. This involved the recording of the nozzle and apertures used, the nozzle supply pressure, the chamber pressure, the atmospheric pressure, search-tube pressures at various intervals along the axis of the nozzle and diffuser, and, in addition, the reaction of the outlet stream.

The tests were repeated with the other diffuser sizes, and the results tabulated as shown. Thereafter the calculations indicated by theory were followed out to determine the balance of energy.

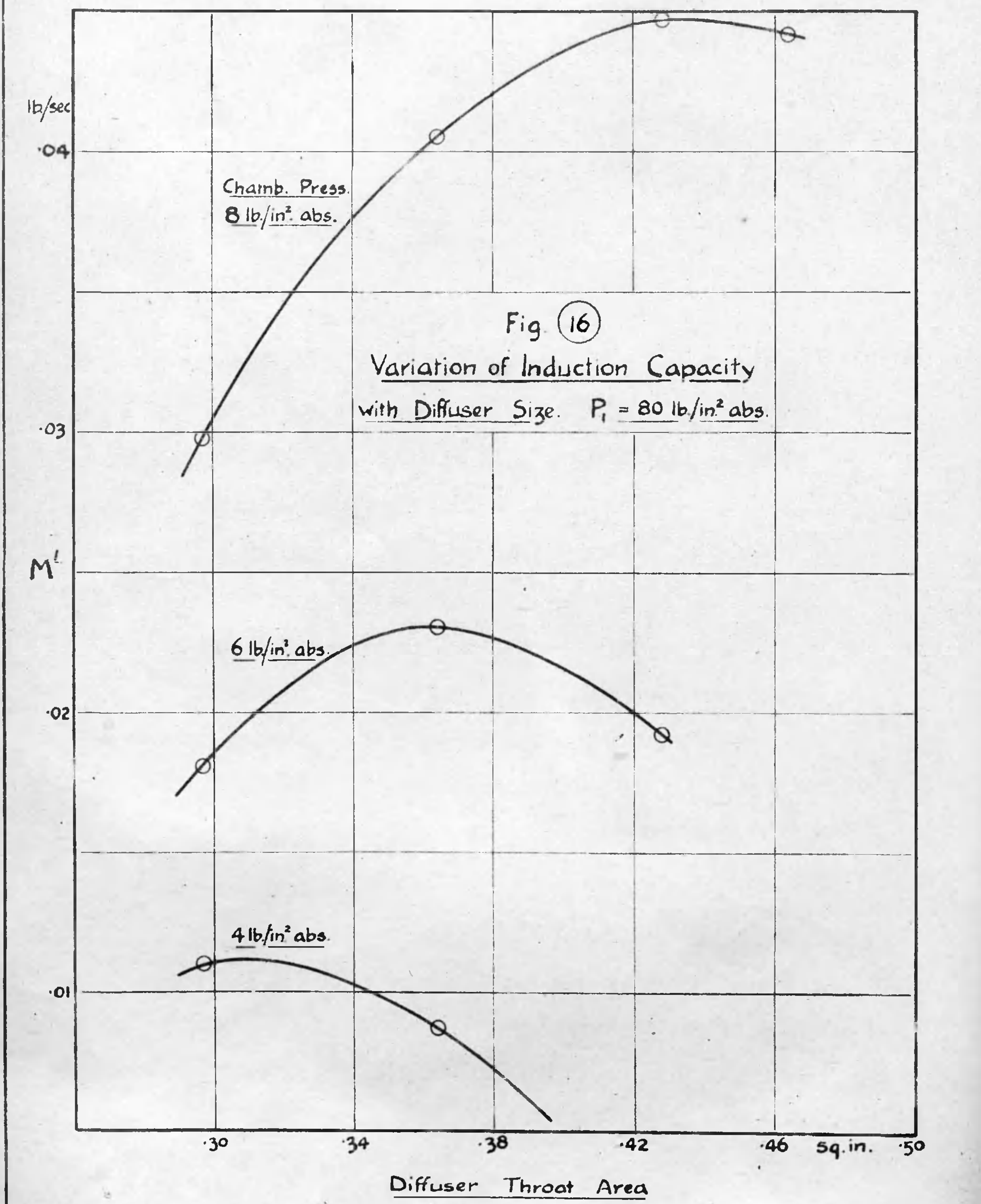


Fig. (16)
Variation of Induction Capacity
with Diffuser Size. P_1 = 80 lb/in² abs.

Figs. 7 - 10 represent the maximum efficiency curves, Fig. 11 the nozzle pressure ratio curves, and Figs. 12 - 15 the diffuser pressure ratio curves.

<u>EFFECT OF DIMENSIONS</u>.- Before considering the problem of entrainment and compression losses, the effect of dimensions on the working of the ejector, as deduced from an analysis of the tests, can be discussed. There are three important variables which come under this heading, viz., the diffuser throat diameter, the distance of the nozzle from the diffuser, and the form of the diffuser. It has already been emphasised that neither does any accepted rule nor accurate theoretical relationship exist which would assign definite values to these variables. In addition, there seems to be a complete absence of data available for their estimation. An attempt will be made, therefore, to analyse the effect of each on the performance of the ejector, and summarise the results. Since the latter were obtained from an air operated ejector, the actual dimensions will not apply to steam flow, but the effect of dimensions on the efficiency and stability of working will be generally the same.

The diffuser throat diameter is seen at once to be the dominating factor in the design of an ejector. It is the critical and ruling dimension. Proof of this is readily supplied from a glance at the curves in Fig. 16, which are deduced directly from the performance curves, Figs. 7 - 10. For a supply pressure of 80 lb. per sq. in. abs., the variation of induction quantity with the diffuser throat area at various chamber pressures

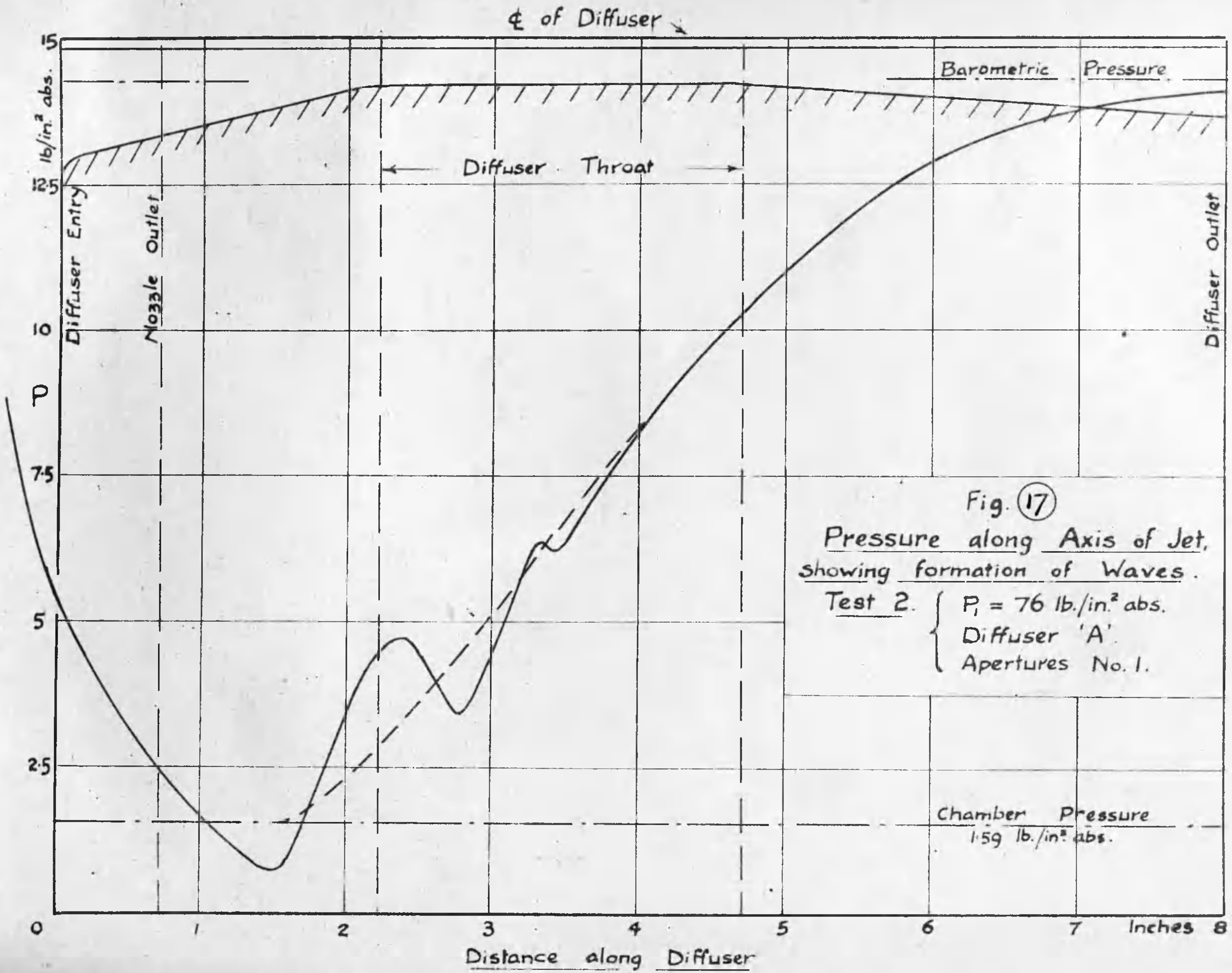


Fig. 17

Pressure along Axis of Jet, showing formation of Waves.

Test 2. P_1 = 76 lb./in.² abs.
Diffuser 'A'.
Apertures No. 1.

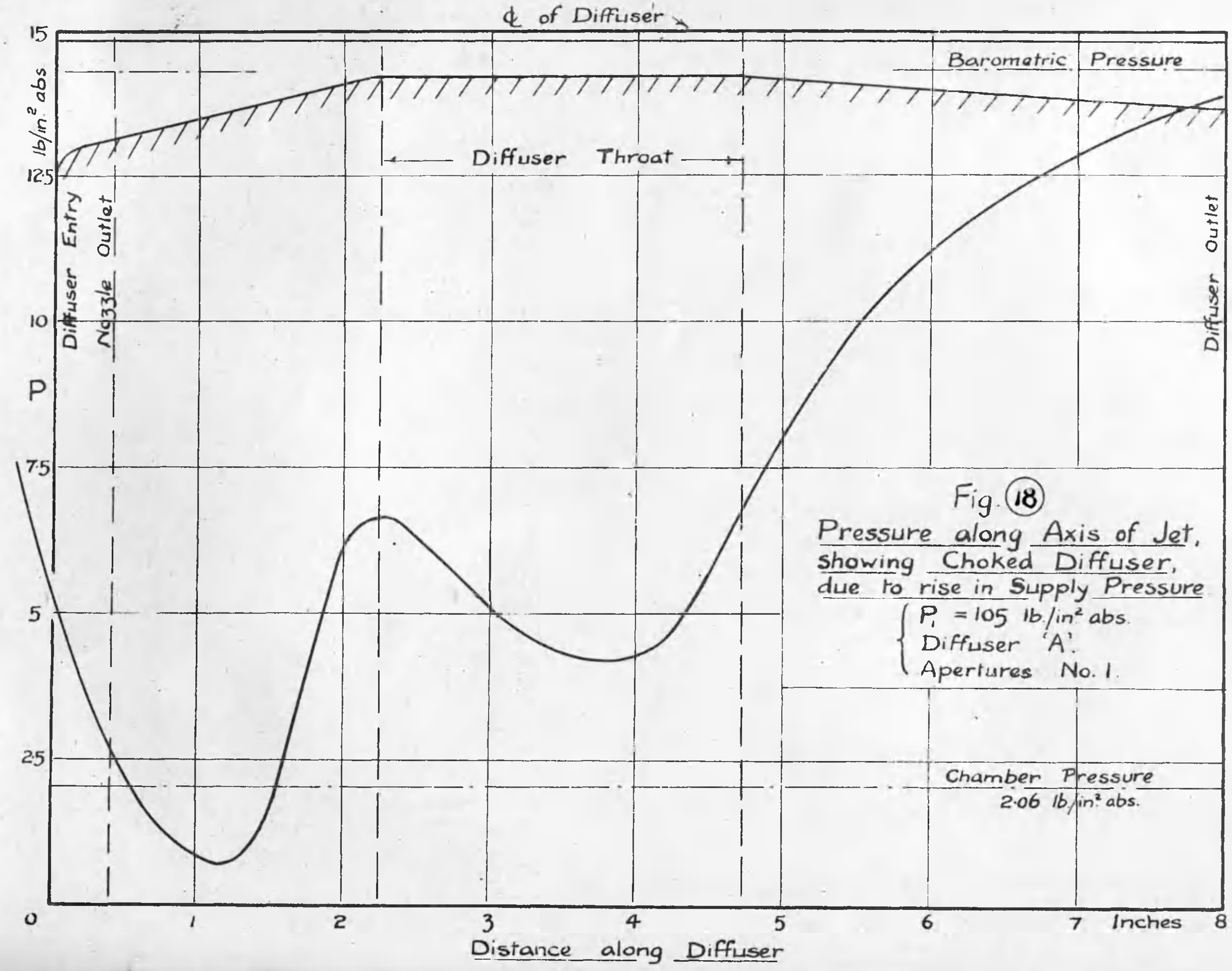


Fig. 18
Pressure along Axis of Jet, showing Choked Diffuser, due to rise in Supply Pressure
P_1 = 105 lb./in² abs.
Diffuser 'A'.
Apertures No. 1.

is shown. A considerable difference in the amount of air entrained may be noted for a comparatively small change in throat area, and if the curves are extended,the limiting area, beyond which no air can be extrained at the specified chamber pressure, is obtained. This is due to the fact that the theoretical sectional area of the stream in the diffuser is dependent on the quantity of induced air and the chamber pressure, as well as the nozzle throat size and the supply pressure, and so choking occurs when the diffuser is too small for the stream, and a leak back of air into the chamber when it is too large.

The choking effect is observed on comparing Figs. 17 and 18. The first represents the compression under the normal working conditions of Test 2, and features the gradual damping out of the waves formed at the nozzle outlet. In the second figure the increase in the nozzle supply pressure, and consequent enlargement of the jet, chokes the diffuser and causes the large wave of compression in front of the diffuser throat. An increase in chamber pressure due to the unfavourable conditions is also noted.

From the performance curves, Figs. 7 - 10, can be deduced what may be termed the ideal, or "optimum dimension", performance curves of the ejector. These are curves of chamber pressure against induced air quantity for constant nozzle supply pressures, of which an example at 80 lb. per sq. in. abs. is given in Fig. 19. Each point on the curves is obtained when working with the dimensions which will give the maximum efficiency.

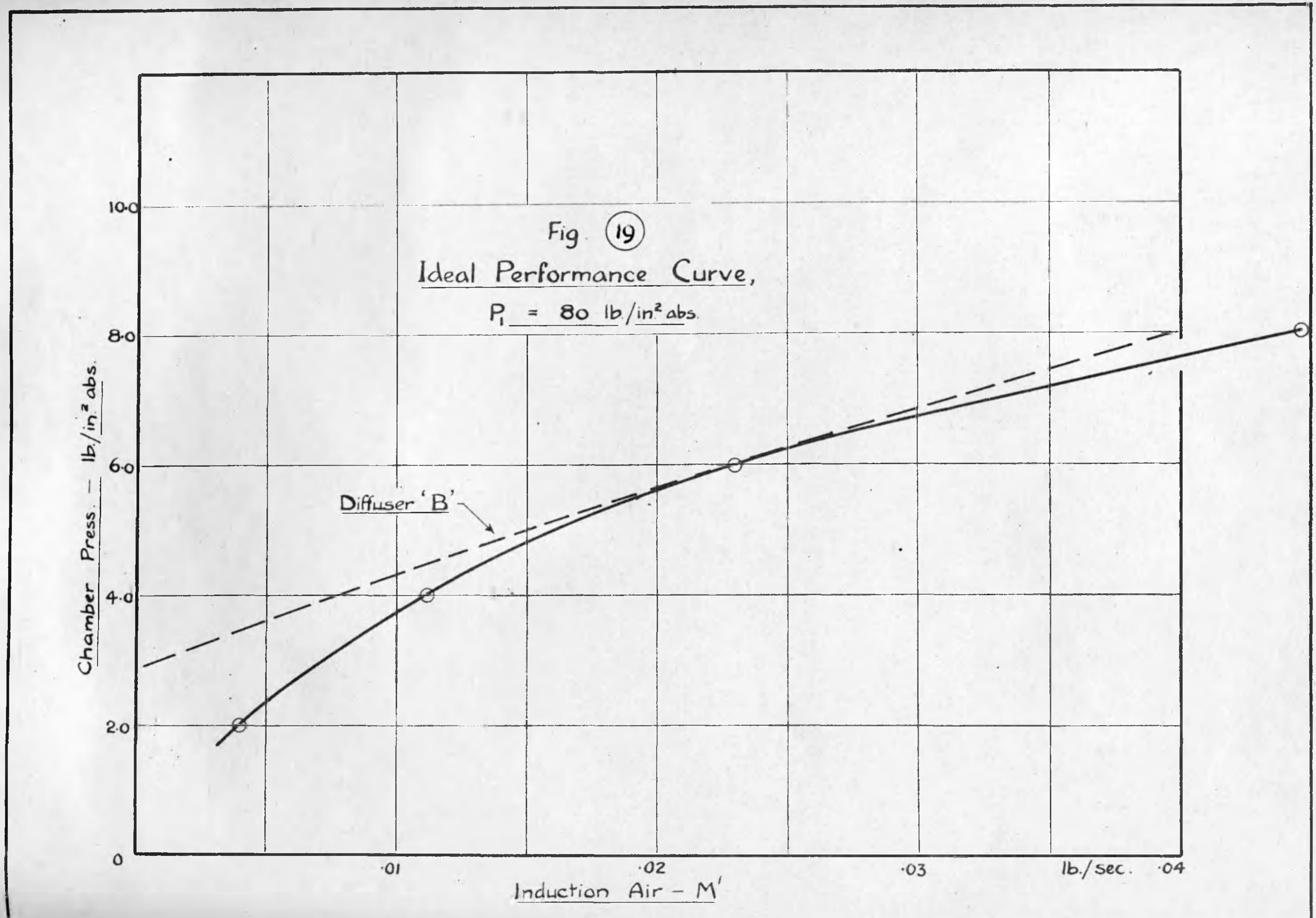


Fig. 19
Ideal Performance Curve,
P1 = 80 lb./in.² abs.
Diffuser 'B'
Chamber Press. – lb./in.² abs.
10·0
8·0
6·0
4·0
2·0
0
·01
·02
·03
lb./sec. ·04
Induction Air – M'

An interesting comparison may be made between this varying dimension curve and the more or less familiar performance curve of an ejector of fixed diffuser size. The dotted curve represents the perfor:mance with the 0·694" throat diameter, and the optimum distance between nozzle and diffuser. There is a striking contrast between the first curve in its tendency towards the origin in a continuous curve, and the dotted curve in its straightness and the definite value of the chamber pressure it gives with no induction. It gives a fair idea of the difficulties to be met with in comparing ejector performances, and the relative merits of single and two-stage working, for instance. The ideal curve shows that there is no definite limit to the vacuum obtainable from the single stage, with dry air, and that the limiting conditions are to be fixed by considering the efficiency and stability.

It has been stated previously that the diffuser diameter involves the question of stability as well as efficiency, for it is found that a much larger area is required, both by theory and experiment, for starting the ejector than for normal running.[3] In addition, the nozzle operates very inefficiently on starting, owing to the very great recompression losses. By enlarging the diameter to ensure a successful start, it is impossible to maintain the vacuum above a certain point, but this difficulty is largely overcome by dividing the compression into two or more stages, while increased efficiency is also gained. It is impossible to study the question of starting conditions in the experimental plant, owing to the small size of the

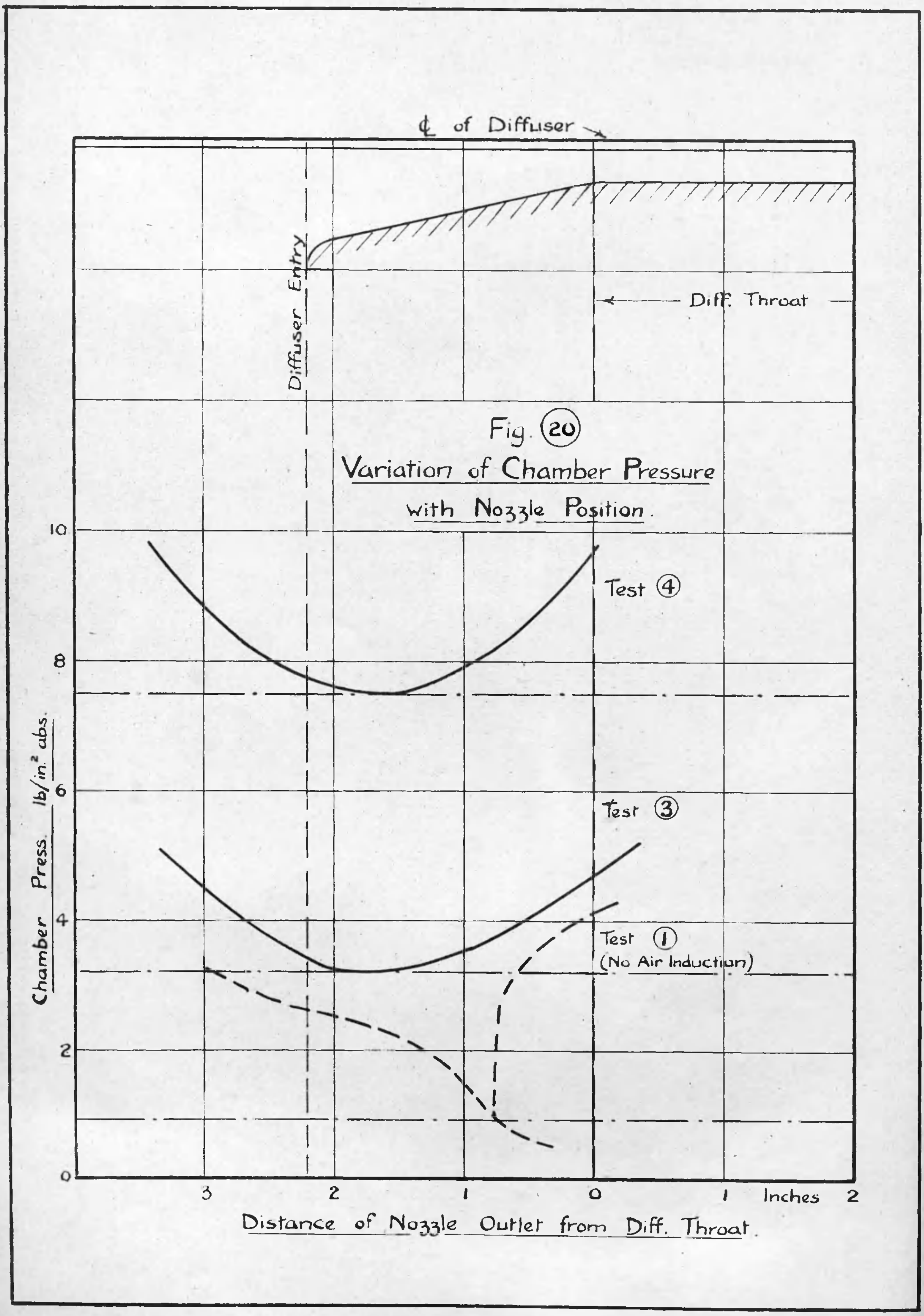


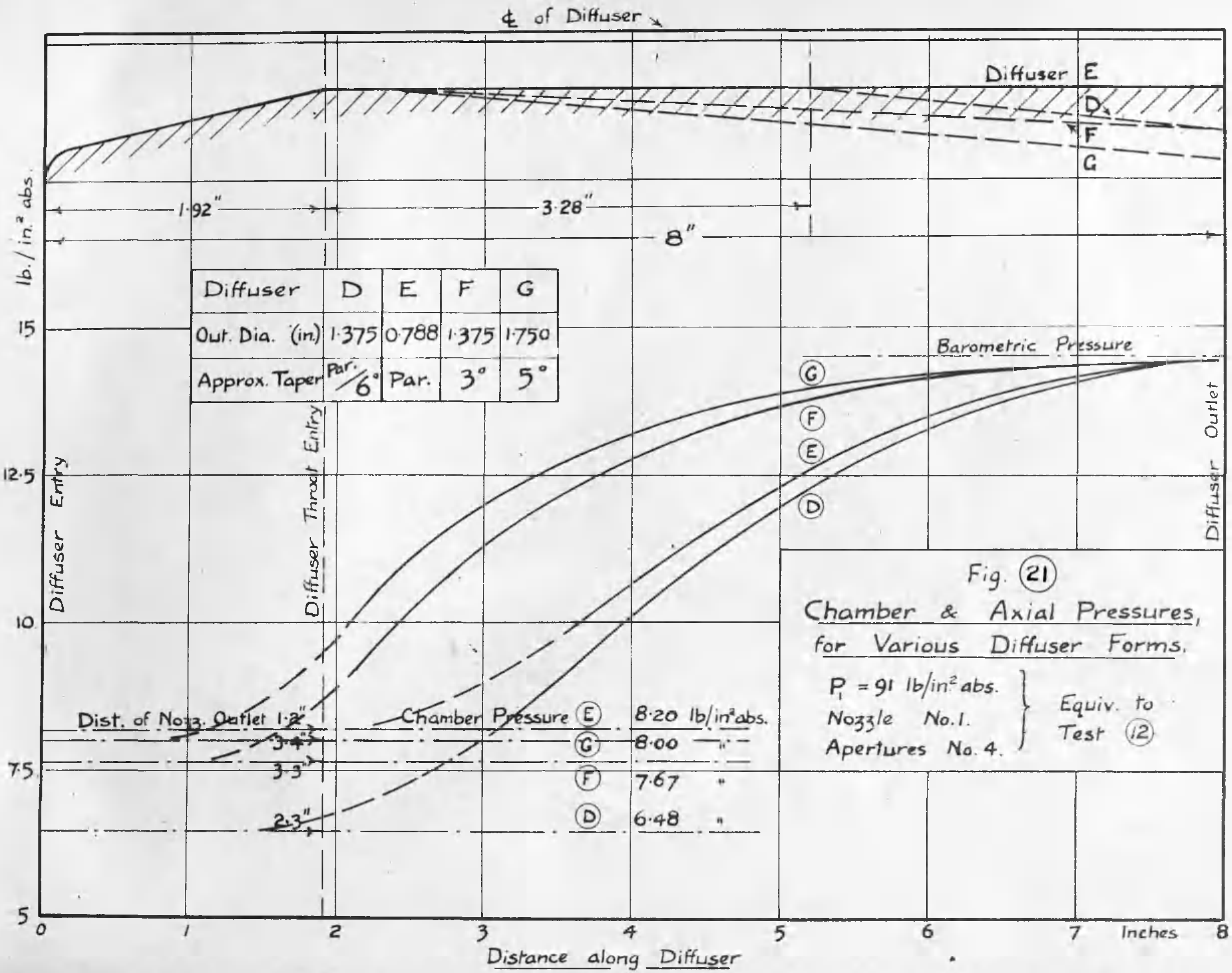
Fig. 20
Variation of Chamber Pressure with Nozzle Position.

chamber, which allows the ejector to adjust itself to its own vacuum almost immediately.

The problem of diffuser throat diameter naturally leads on to that of diffuser form, and the distance of the nozzle from the diffuser. In all the tests the nozzle distance has been such as to give the reading of minimum chamber pressure. There are several factors upon which this distance depends, namely, the throat and outlet sizes of the nozzle, the nozzle supply pressure and the chamber pressure, as well as the form of the diffuser. An analysis of the test results shows that the distance is least when the ejector is working at a high vacuum with little air suction, while an increase of nozzle pressure, by increasing the dimensions of the jet, requires an increase in the nozzle distance. (See Fig. 22, later.)

The effect of nozzle distance on the chamber pressure is recorded for a few tests in Fig. 20. Under ordinary conditions, no critical effects are noticed, but when no air, or a very small amount, is being induced, a region of instability is set up, as shown by the dotted curves. The upper one is obtained when the nozzle is drawn away from the diffuser, and the lower one when it is advanced towards it. It is difficult to state the exact cause of this instability, but there can be no doubt that the very great change in jet size resulting from a small variation in the chamber pressure at such high vacua has a great deal to do with it.

Considering now the form of the diffuser, the most important problem is the rate of divergent taper immediately after the throat section is reached. The



Diffuser	D	E	F	G
Out. Dia. (in.)	1·375	0·788	1·375	1·750
Approx. Taper	Par./6°	Par.	3°	5°

Fig. 21
Chamber & Axial Pressures, for Various Diffuser Forms.

P_1 = 91 lb/in² abs.
Nozzle No. 1.
Apertures No. 4.
Equiv. to Test 12

experimental diffusers were designed on the lines indicated because there was good reason to believe that an extension of the throat section, or a very gradual taper at first, would be the ideal form. Since the form of channel in which the stream would be most suitably compressed is a most interesting question in fluid flow, quite apart from its application to the ejector, it was decided to investigate different types, and at the same time verify the assumption in the case of the experimental diffusers.

The tests were carried out on a set of four distinct diffuser types, and the results are given in Fig. 21. The ejector was operated in each case under conditions similar to those in Test 12, with a diffuser throat diameter of 0·788", and the chamber pressure and search-tube pressure readings were taken at the optimum distance of the nozzle from the diffuser. Diffuser D is of the type used in Test 12 , in E the parallel section is extended to the outlet, F has an outlet area the same as D, and in G the rate of divergence is increased. In both F and G the parallel section has been reduced to a minimum.

The chamber pressure readings show that, as expected, diffuser D is easily the best of the four types. Comparing D and F, it is evident that the unsuitability of the latter is due to a too rapid divergence immediately after the throat section has been reached. In G this effect is accentuated, and the slope of the pressure curves just beyond the throat in the case of F and G seem to show that compression is taking place too rapidly. Diffuser E provides an illustration of the other extreme, where the compression is drawn out too long. From a first glance at the curves it seems that D and E are very much the same, but the higher chamber pressure of the latter gives the appearance of an earlier compression. Possibly the most surprising feature of the tests is that diffuser E gives almost as good a performance as G.

The possibility of the chosen conditions of test being suitable for one diffuser form but not for another, was given consideration. For instance, it was thought possible that an extension of the throat might have the same influence as a reduction in the diameter, and so result in a different optimum value of the latter. Tests under varying nozzle supply pressures for each diffuser did not give any evidence of this, however, and showed that the results in Fig. 21 are representative of the performances of the various diffusers.

There remains no doubt as to the ideal diffuser form; it is necessary that divergence should not take place rapidly after convergence. This condition is fulfilled by having a parallel extension to the throat or a very gradually increasing divergence after the throat section has been reached. By cutting down the diffusers, it was also observed that the minimum length required for efficient working is considerably less than that chosen for the experimental diffusers.

The results of the investigation on diffuser design can be combined to form the instructive calculation chart shown in Fig. 22. It is intended to give some idea of how an actual plant could be designed from specifications, although for direct practical use

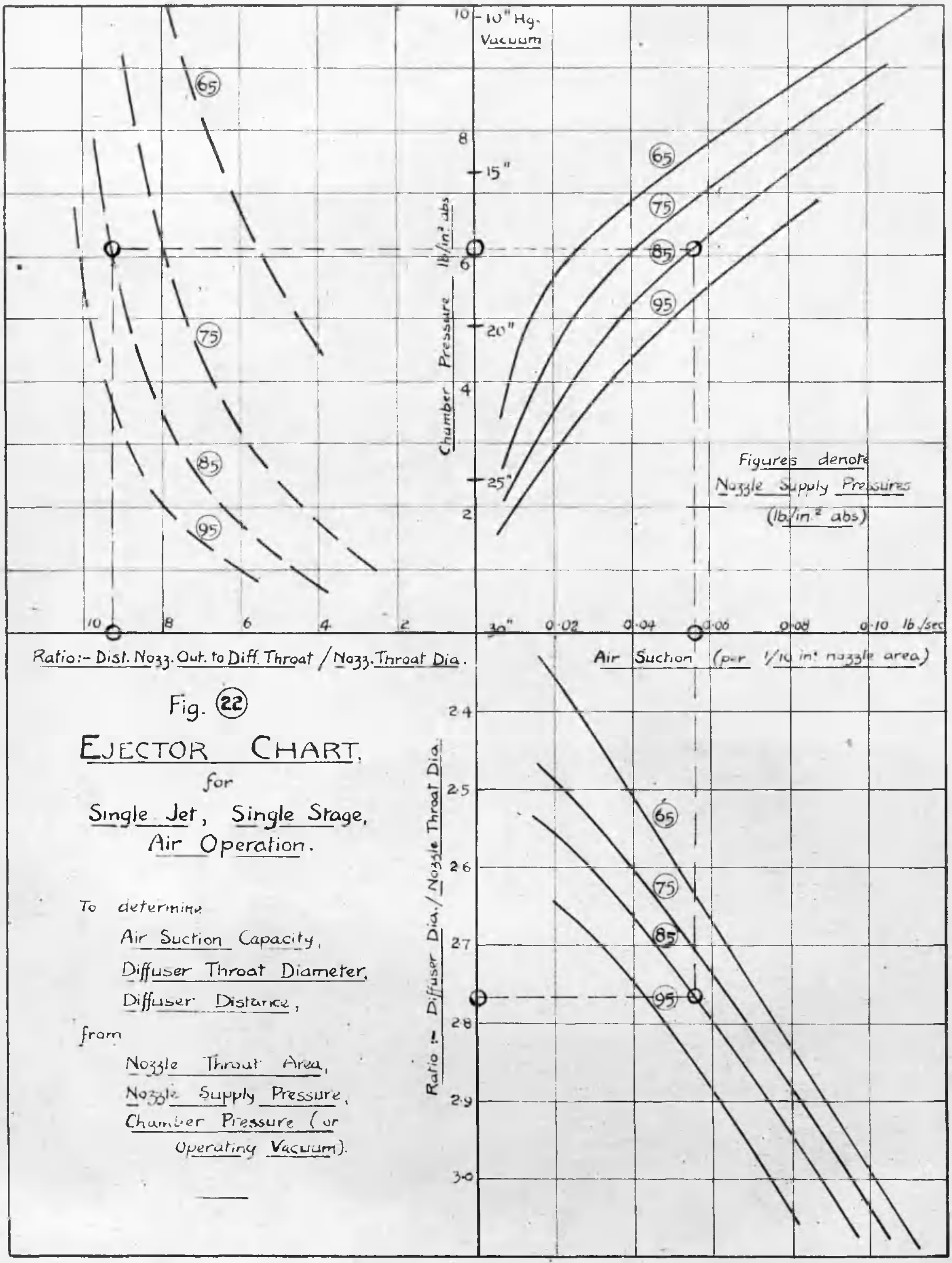


Fig. 22

EJECTOR CHART

for

Single Jet, Single Stage, Air Operation.

To determine

Air Suction Capacity,

Diffuser Throat Diameter,

Diffuser Distance,

from

Nozzle Throat Area,

Nozzle Supply Pressure,

Chamber Pressure (or Operating Vacuum).

a chart would require, of course, to be based on steam as the working fluid, and on a more comprehensive range of supply pressures. Such a chart, the author believes, would be more useful for design purposes than a theoretical calculation or empirical formulae.

The usual specifications include the nozzle throat area of each ejector unit in the plant, the supply pressure of the working fluid, the amount of air to be dealt with by the plant, and the vacuum from which it has to be pumped. The chart is based on unit nozzle throat size, which, along with the supply pressure, governs the ejector consumption. Constant nozzle pressure curves of chamber pressure (or operating vacuum) against air induction quantity per unit nozzle throat area were deduced from the tests in the same manner as the ideal performance curve in Fig. 19. From these the air handling capacity of each unit in the plant can be calculated from the given nozzle size, and hence the number of ejector units required to deal with the specified air quantity. The diffuser throat diameter, and the distance of the nozzle outlet from the commencement of the diffuser throat section, for unit nozzle throat diameter, can then be read off on curves which also have been derived from the results of the tests.

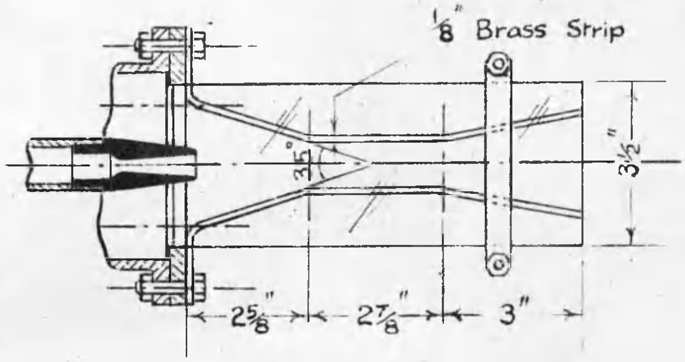
For example, if the stage vacuum is to be 17·5 inches of mercury, the nozzle supply pressure 85 lb. per sq. in. abs., and its throat diameter 0·25", then the induction capacity of the ejector will be 0·027 lb. per sec., the diffuser throat diameter 0·69", and the nozzle distance round about 2·3".

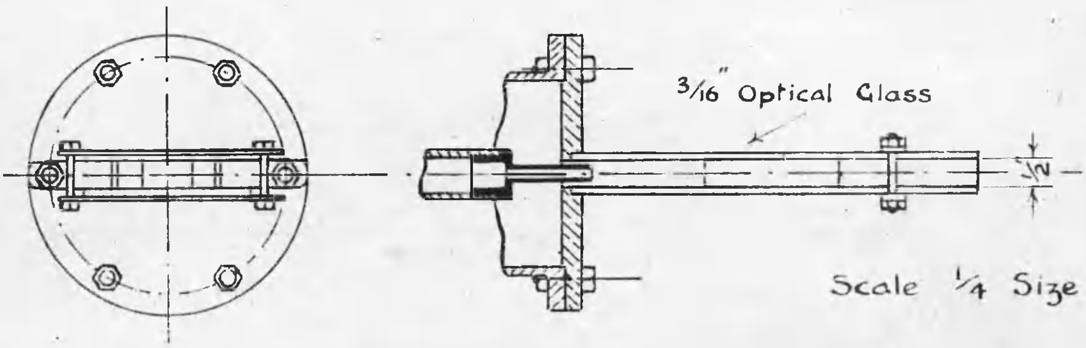
Further evidence of the effect of dimensions is provided by photographs of the jet, which are made possible

Fig. 23

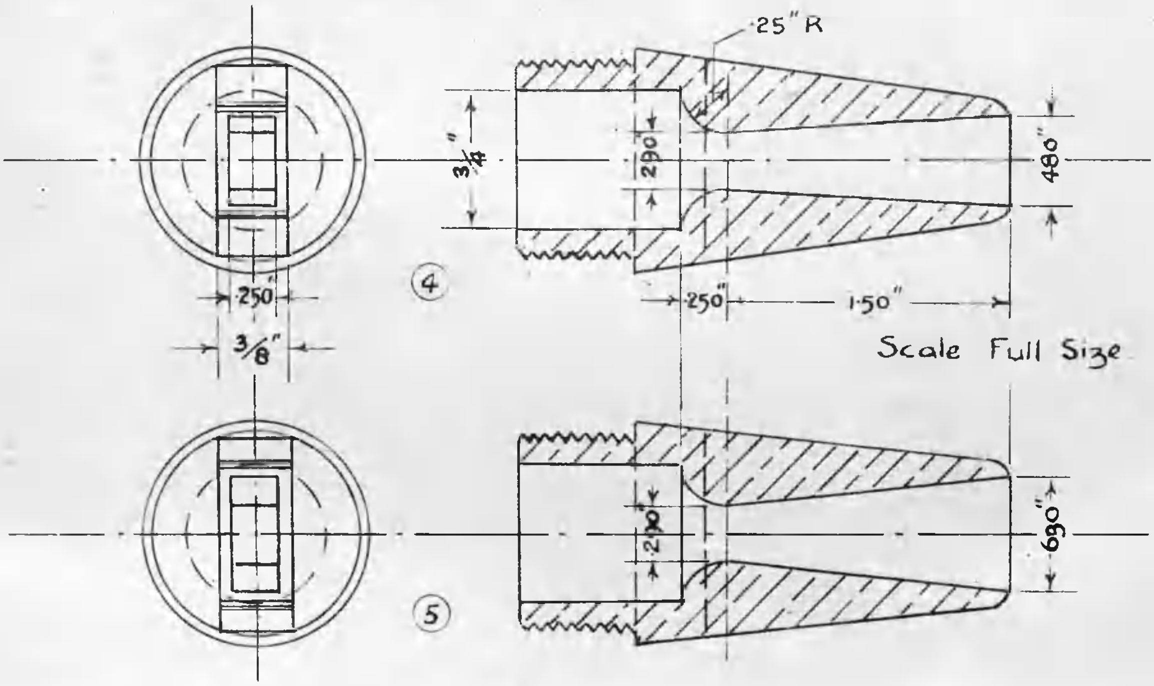
Experimental

Air Ejector.





Sectional Elevations & Plan
of Rectangular Glass-Sided Diffuser.



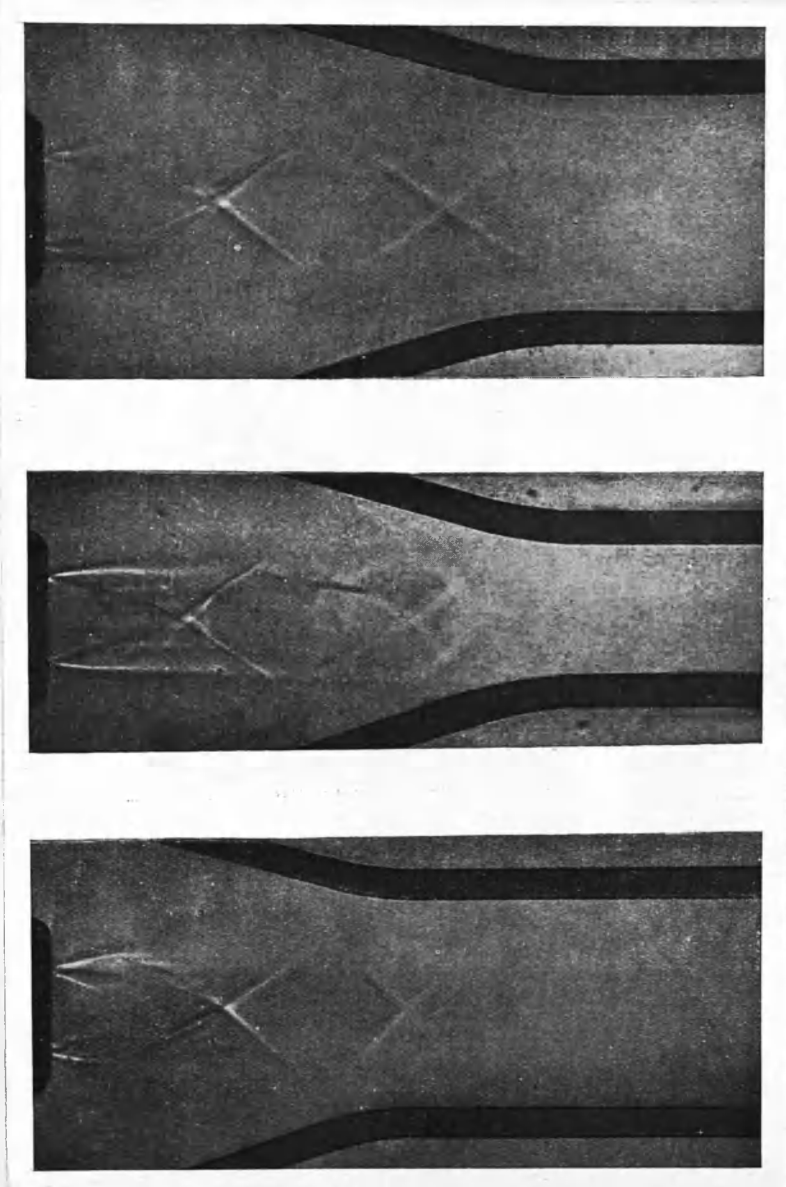
Rectangular Nozzles.

by the refraction of light passing through a high velocity stream.[5] To obtain the photographs a glass-sided diffuser was required so that the rays of light from an arc spot lamp could pass through the combining space on to a screen beyond. It would be out of the question to use this method with a glass diffuser of circular section, owing to the refraction caused by the glass, but a rectangular form, both of nozzle and diffuser, was found to give a reasonable efficiency and to conform closely enough with the actual diffuser form for the purposes of observation.

The details of the apparatus are shown in Fig. 23, where it is seen that the diffuser and nozzle were adapted to fit the existing apparatus. To give the expansion required by the experiments, a nozzle (No.5) of the dimensions given in the figure was necessary. Two of the sides of this nozzle are parallel and ¼" apart. The diffuser was framed by brass strips, ½" in width, which were bolted on to the combining chamber in such a way that their distance apart, and hence the diffuser throat width, is adjustable. These strips act as distance pieces for two plates of 3/16" optical glass, held in position by a slot in the chamber cover plate and by a clamp.

With the chosen nozzle pressure and air aperture, it was necessary first of all to find the diffuser diameter and distance to give the best results, and as these dimensions were easily varied on the apparatus, they could be determined directly by trial. An ordinary camera was used to take photographs of the shadows thrown on the screen by the refraction of the

Fig. 24 – Shadow Photographs of waves in the combining chamber of a glass-sided ejector, to show the effect of dimensions on the jet form. Nozzle pressure 90 lb./in.^{2} abs.; induction 0·022 lb./sec.



(a)

(b)

(c)

(a) Optimum diffuser distance and throat area. Chamber press 7·0 lb/in.^{2} abs.

(b) Diffuser throat area reduced.

(c) Diffuser distance reduced.

light passing through the jet. During exposure, the compressor supplied air to the nozzle at steady pressure, and great care was taken to prevent condensation on the glass sides of the diffuser.

The first exposure represents the ejector working under the optimum conditions as in Fig. 24 (a); in (b) the throat area of the diffuser was reduced by bringing the sides closer together; and in (c) the distance of the nozzle from the diffuser was reduced.

In all three photographs the form of the jet can be followed, more or less, by the refraction effects of the waves formed naturally at the nozzle outlet. These compressions and rarefactions are always present at the outlet of a nozzle which expands beyond the critical ratio, which is 0·528 for air. When the nozzle under expands (i.e. the back pressure is below the nozzle outlet pressure) the waves have their origin at the outlet edge of the nozzle, and they are of the same type as the compression wave formed at the nose of a bullet,[4] referred to more fully in the paper on nozzle losses. If v_c is the velocity of sound in air at a point in the stream, then the angle α between the nozzle axis and the wave cone is such that $\sin \alpha = v_c / v$.

The greater the difference between the outlet and the back pressures, the more pronounced do the waves become, but even when these pressures are the same, minor waves still appear. The nozzle outlet section was designed to under expand slightly so that the waves might be fairly well defined in the photographs, while at the same time the nozzle would be working under normal conditions.

No outward effect of the entrainment of the air can be observed in the photographs; in (a) the jet is not visibly disturbed almost until the throat section is reached, when entrainment takes place. It is interesting to note the clearance between the visible boundary of the jet - that at which the waves are reflected - and the diffuser walls. In (b) the sectional area of the diffuser is considerably reduced, and its effect of choking the ejector is plainly seen in the setting up of a disturbance surrounding the stream, in front of the throat. This disturbance is of the same type as that illustrated by the pressure curves in Fig. 18, where the diffuser is too small for the jet. The considerable reduction in wave length is due to an increase in chamber pressure with less efficient working, and hence a reduction in the jet velocity and the amplitude of the waves.

In (c) the effect of bringing the nozzle closer to the diffuser is not nearly so marked. Compression takes place sooner, of course, and the amplitude of the waves is seen to decrease rapidly as compression starts. The chamber pressure is only slightly higher than that in (a) as a result of moving the nozzle, and this is reflected in the very similar wave form at the outlet in each case. It is significant to note that varying the nozzle distance gives a chamber pressure curve, as in Fig. 20, which shows no effect of the varying wave front at the point where the stream reaches the diffuser. This seems to prove that the waves have no direct influence on the entrainment of the air.

Loss Tests.

Table I.

No. of Test	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Diffuser	A				B				C			D		
Nozzle	3	3	2	1	3	3	2	1	2	2	1	1	1	1
Dist. of Nozzle Outlet from Diff Throat Entry in.	78	1·51	2·02	1·47	1·24	2·56	2·39	1·56	2·27	2·34	2·30	2·31	2·30	1·89
P_1 lb/in² abs	77	76	75	57	97	96	95	66	104	93	70	91	84	60
T_1 °F	576	576	576	565	585	584	584	572	588	583	574	582	579	569
M lb./sec	·0867	·0856	·0845	·0642	·1080	·1070	·1060	·0744	·1156	·1040	·0789	·1048	·0969	·0694
P_2 lb/in² abs	·91	1·59	3·19	7·47	1·60	2·50	4·60	8·00	3·92	5·85	9·08	6·46	7·71	10·6
P_A & P_4 lb/in² abs	14·30	14·30	14·30	14·30	14·50	14·60	14·60	14·60	14·60	14·60	14·60	14·50	14·50	14·50
Suction Apertures	0	1	2	3	1	2	3	4	3	4	5	4	5	6
M′ lb./sec	0	·00254	·00821	·0203	·00254	·00821	·0203	·0332	·0203	·0332	·0461	·0381	·0464	·0584
r_2	·0118	·0209	·0425	·131	·0165	·0261	·0484	·121	·0377	·0629	·130	·0710	·0918	·177
E_2 ft.lb/lb oper. air	70,800	65500	57400	40700	68900	64200	56000	42300	60,000	52,800	41,100	51,600	47,400	35700
(M + M′)/M	1·00	1·029	1·096	1·320	1·023	1·068	1·192	1·447	1·174	1·319	1·584	1·363	1·480	1·840
E_3 ft.lb/lb. oper. air	70,800	63,700	52,300	30,900	67,500	59,700	47,200	29,300	51,100	40,200	25,700	37,900	32,100	19,400
E_4 ft.lb/lb oper. air	840	1,010	1,590	1,400	1,480	1,650	1,920	2,090	1,950	1,680	1,620	1,520	1,480	1,010
T_4 °F	429	440	458	484	418	443	467	499	467	488	505	489	497	514
r_3	·0636	·111	·223	·520	·110	·171	·315	·556	·268	·401	·622	·446	·532	·732
W ft.lb/lb. oper. air	43,400	39,200	32,800	20,600	38,000	35,300	29,000	20,800	31,900	27,500	19,000	25,800	22,600	15,100
Diff. Friction Loss { ft.lb/lb. oper. air	2,600	2,300	2,200	1,700	2,300	2,100	2,000	1,800	2,100	2,000	1,600	1,900	1,800	1,400
Other Losses { ft.lb./lb. oper. air	24,000	21,200	15,700	7,200	25,700	20,600	14,100	4,600	15,100	8,900	3,500	8,700	6,200	1,900
" " { ft.lb./lb. comb. air	24,000	20,600	14,300	5,450	25,100	19,300	11,800	3,200	12,800	6,750	2,400	6,400	4,200	1,030

LOSSES AND ENTRAINMENT ACTION.- By following fundamental theory in the analysis of the test results, several assumptions already alluded to have had to be made, and, of course, they do not represent exactly the existing conditions. However, the object of the analysis is primarily to suggest the nature of the losses, and not to assign definite values to them. Allowance has been made for the known losses in the various stages, i.e., the nozzle and diffuser friction losses and the theoretical loss in energy due to the combining of the streams. The energy balance is left to account for the unknown losses which occur between the nozzle outlet and the diffuser outlet.

It is probable that the theoretical loss of kinetic energy by following the law of the conservation of momentum does not represent the actual loss in the combining of the jets, so that the unaccounted loss must be divided between this and a loss in the compression of the stream in the diffuser. In connection with divergent nozzles it is well known that the loss in compression following the over-expansion of a jet is a very important one, and as the fluid actions in the divergent nozzle and the ejector are similar in principle, it is most likely that the greater part of the loss is incurred in this stage.

It is found that, by selecting the compression ratio r_a as base, the losses in each test fall on a common curve (Fig. 25), the trend of which seems to support the theory of a compression loss. For, while it is impossible at this stage to estimate the magnitude of the compression loss, it is certain to increase as the

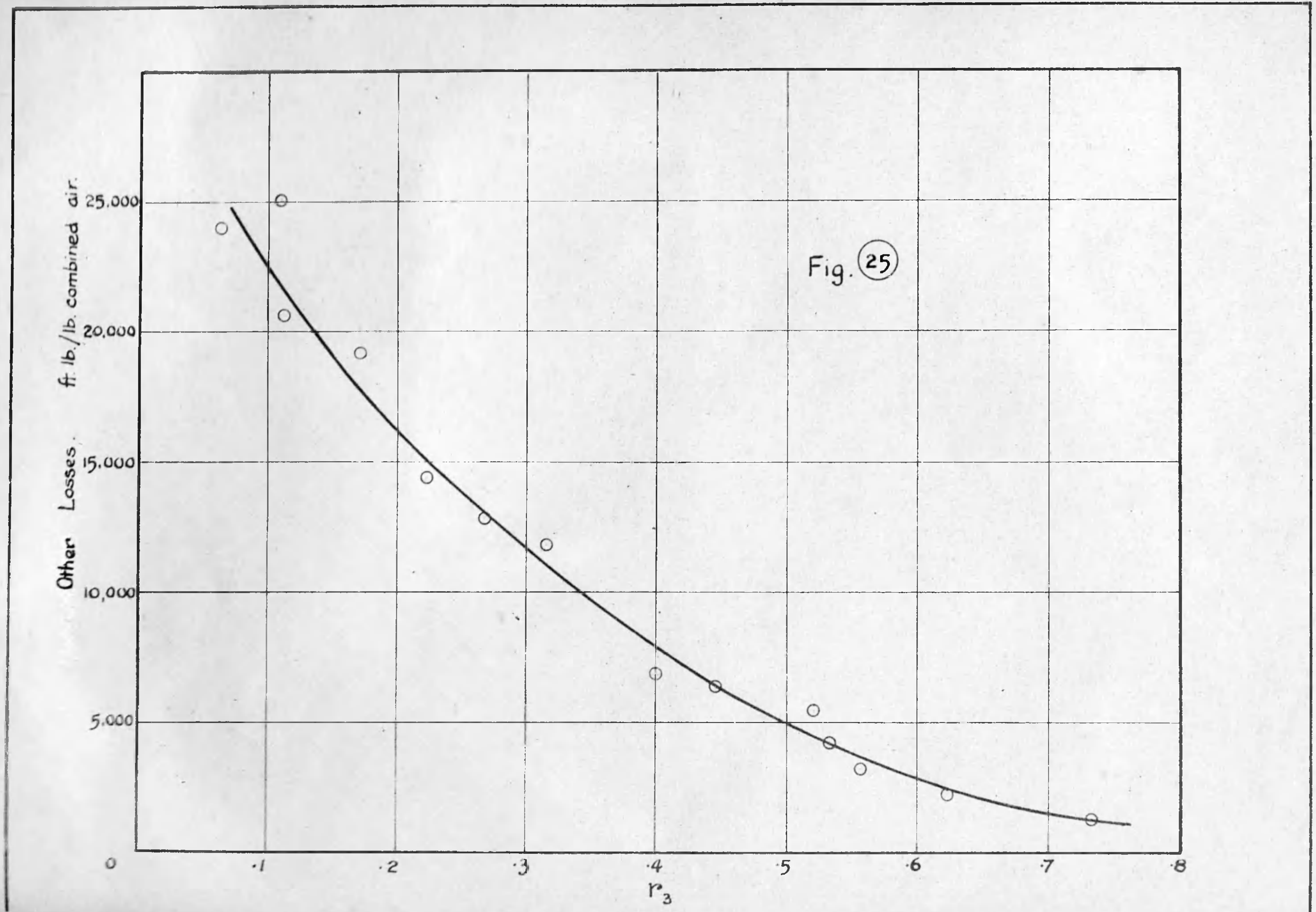


Fig. 25
Other Losses. ft. lb./lb. combined air.
25,000
20,000
15,000
10,000
5,000
0
·1
·2
·3
·4
·5
·6
·7
·8
r_3

compression ratio r_a decreases. Such a loss certainly could not be ascribed to the difference between the theoretical and the actual kinetic energy loss in entrainment. The irregularity noticed in the tests at the extreme pressure ratios is due to the instability as observed in the nozzle distance curves for Test 1 in Fig. 20.

The efficiency of the ejector may be taken as the ratio of the work done on the entrained air to the energy expended overall by the operating fluid. Since the amount of air entrained may vary between very wide limits, and since in practice nearly all the heat units in the operating steam are regained in the main condensate used to condense the steam, such a definition of the efficiency is of little or no practical value. For purposes of design, etc., the efficiencies of the separate stages are considered. These are the nozzle efficiency, the ratio of combined to nozzle jet kinetic energies, and the diffuser efficiency.

The nozzle efficiency is usually calculated from empirical formulae, and has a value round about 90%. The ratio of combined to nozzle jet kinetic energies is often passed over as being unity, but this is not permissible unless the amount of induced air is very small, for it has been seen that the energy ratio is given by $\frac{M}{M+M'}$ where M and M' are the operating and induced fluid quantities respectively.

The diffuser efficiency may be defined as the ratio of work done in compressing the combined jet to the kinetic energy of the combined stream, or, more exactly, the difference between the kinetic energies of the combined and discharge streams. There seems to be no empirical rule for its estimation, so that the assumed values vary over a wide range. Some assert that, under normal conditions of working in a condenser plant, it cannot be more than 50%, and others give such high values as 70% or even greater. Since the efficiency varies so much with the conditions, it would be quite useless to make comparisons from the tests, but in a typical example where the compression ratio is 1 : 4·5, corresponding to a two-stage ejector vacuum of $28\frac{1}{2}$", the diffuser efficiency works out at 65%, with a nozzle supply pressure of 75 lb. per sq. in. abs., and 10·3 lb. of operating air per lb. induction air.

The practice of passing the induced air through a restricted opening at the mouth of the diffuser, so that it may meet the operating jet with a considerable velocity, and thereby reduce the energy loss in combining, is employed in some cases, especially where there is a small range of compression, but it does not seem to offer any advantage in an ejector of the type used in connection with steam condensing plant.

Another modification, that of cooling the stream before and during compression, by water jacketing, is claimed to have several advantages in this class of ejector. By reducing the temperature in the combining space the water vapour pressure is reduced, so that more air can be dealt with, and, in addition, less work has to be done in compression. The effect of water cooling could not be observed in the experimental plant, for, under the particular conditions of working, the chamber pressures were considerably below atmospheric.

The choice of operating fluid supply pressure to give satisfactory results does not appear from the tests to require careful consideration. An analysis of the performance curves (Figs. 7 - 10) shows that the efficiency of the ejector when working under optimum conditions is almost unchanged over the fairly wide range of pressure in the tests.

Curves of performance at various supply pressures, using the same fixed ejector plant, are often cited to show that it would be uneconomical to run an ejector outside a certain range of supply pressure. It is obvious that this is more a question of the suitability of the particular dimensions of the ejector in question than anything else, for the supply pressure is one of the variables which influence the theoretical dimensions of the stream. Figs. 7 - 10 show how critical is the effect of supply pressure on the capacity of an ejector of fixed diffuser diameter.

The problem of how the entrainment of the fluid surrounding a high velocity jet actually takes place has been the subject of considerable interest and speculation. It has been stated that the idea of friction alone being responsible for the entrainment has lost favour; it is now generally recognised that the jet of fluid leaving the nozzle expands to a pressure below that of the chamber, and so causes an inrush of the surrounding fluid, on the same principle as over-expansion occurs inside the divergent type of nozzle.

This over-expansion effect at the nozzle outlet must not be confused with the familiar stationary waves of compression outside the nozzle, which are usually due to the interference of the outlet edge. If the back pressure is much below the outlet pressure of the nozzle, the waves caused by expansion round the edge of the nozzle have a very great amplitude, and consequently the pressure in the jet is very much below the back pressure in places.

When the outlet edge effect is eliminated by equalising the outlet and back pressures, but only in this particular case, is the wave formation due to an over-expansion tendency. The jet continues its expansion beyond the nozzle, and the streamlines diverge until the surrounding pressure forces them to reconverge, according to the observation of Prandtl.

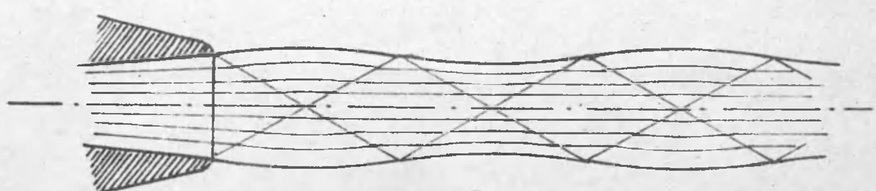


Fig. 26 (after Prandtl).

The process repeats itself as in the sketch, Fig. 26, with the result that along the axis of the jet there appears a wave formation, which is of very small amplitude, however.

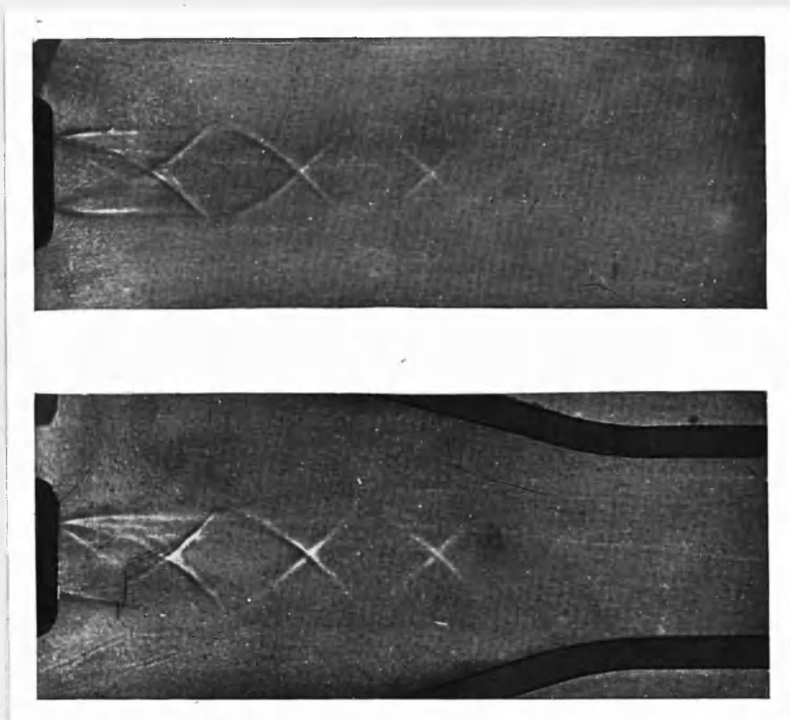
When a nozzle is discharging into an open space, the tendency of the jet to over-expand is not directly evident, but if suitably controlled in an appliance such as the ejector, a pressure lower than the back pressure is established round the nozzle exit. This property of the jet is, of course, taken advantage of in the ejector to induce an outside fluid and compress it in the diffuser. The rate at which the pressure is lowered, and the pressure finally reached, depend on the volume of the space surrounding the nozzle outlet, and the amount of fluid which is allowed to leak into the space, respectively.

The exterior form of the nozzle at the outlet might be expected to have an important effect on the entrainment of air, i.e., with the nozzle outlet in a large flat surface it was thought that full advantage of the over-expansion effect could not be taken by the entrained air. Comparative tests on the ejector with a nozzle fully tapered on the exterior, and one with a flat outlet face of 1" diameter, but otherwise similar, were made to investigate this.

Under conditions of test which required that the nozzle should be some distance away from the diffuser, it was found that the entrainment is quite unaffected by the exterior form. It was impossible to compare the nozzles in a test where they are close to the diffuser throat, owing to interference between the second nozzle and the diffuser. If allowance could be made for this, a comparison would likely show an advantage in favour of the tapered nozzle, since entrainment must take place very soon after the nozzle outlet in this case.

With the object of investigating the effect of the entrainment on the visible form of the jet, photographs of jets from the same nozzle working at the same expansion ratio, one jet expanding freely to atmosphere and the other entraining air in the glass-sided

Fig. 27 — Shadow Photographs of waves in the combining chamber of a glass-sided ejector, to compare the forms of free and combining jets of the same expansion ratio.



(a.)

(b)

(a) Expansion to atmosphere (with brass strips of diffuser removed). Nozzle pressure 95 lb./in.^{2} abs.

(b) Nozzle pressure 65 lb/in.^{2} abs.; induction 0·046 lb./sec.; chamber pressure 10·0 lb./in.^{2} abs.

ejector, were taken. In the first case (a) the frame of the diffuser was removed, and the glass faces left in position. The rectangular nozzle No. 4 of dimension given in Fig. 23, was operated at the highest pressure obtainable, and an exposure made in the same way as in previous tests. To find the supply pressure in the second case (b) which gives the same expansion ratio as in (a), trial tests had to be carried out with the diffuser and the required air apertures in position. The diffuser was finally set to the correct width and distance from the nozzle, and an exposure made when operating under the chosen supply pressure. The photographs are reproduced in Fig. 27.

From the results it is concluded that the jet is quite unaffected right up to the point of disappearance of the waves - very nearly to the diffuser throat - by the entrained air. This means that entrainment does not take place within the visible form of the jet, otherwise there would be a reduction in the wave length due to the decreased velocity of the stream. Theoretically, the jet does not reach the diffuser walls and start compressing until the throat section is reached, as the ratio of compression in the diffuser is considerably above the critical. It is therefore most likely that the entrained air joins the nozzle jet at the point where the stream fills the diffuser section, in the vicinity of the throat in this particular case. This explains how the exterior form of the nozzle has no effect on the entrainment unless the nozzle outlet is near the diffuser throat.

WATER AS THE OPERATING FLUID.- The capacity of a water jet to entrain air is known to be inferior to that of steam (or air), and, in addition, the combined stream does not form an ideal mixture for compression in the diffuser. When the amount of air to be entrained is small, however, the vacuum produced compares favourably with that obtained in the ejector operated by gaseous fluid.

This has resulted in a variety of attempts to increase the entrainment capacity of the simple jet by mechanical means, for the water jet ejector is often used for second stage air extraction in condenser plants. One method is to lead in the induction air through a series of channels formed by cones inserted between the nozzle and the diffuser; another is to give the jet a screw motion which is understood to trap an increased quantity of air. A still more unorthodox method is to replace the nozzle by a centrifugal pump, which projects an inter:
:mittent stream of water along the induction pipe directly towards the diffuser, and so air is entrained between the sheets of water.

An investigation into the respective merits of gaseous fluid and water ejectors of straightforward design would therefore lead to no useful conclusions, except possibly when the amount of entrained air is comparatively small to suit the simple form of water jet.

In conclusion, the author wishes to express his indebtedness to Professor A.L. Mellanby, D.Sc., M.I.Mech.E., for his guidance, and for granting the facilities to instal the experimental plant. The author's thanks are due also to Professor W. Kerr, Ph.D., M.I.Mech.E., for the interest he took in the work.

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THE AIR EJECTOR.

By A. D. Third.

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The Air Ejector.

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ABSTRACT.

This paper deals with the general characteristics of the air ejector. Except for a few published results on the performance of standard units, information on the subject is scarce, and the experimental lines followed in this investigation have been arranged to provide a systematic examination of the effect of the various dimensions on the stability and efficiency of operation.

Diffuser losses are deduced by analysis from the tests, and the main questions arising in the combination of the operating and suction fluids are discussed. Photographs of several aspects of the fluid action are shown.

Introduction.—The air ejector is now an indispensable adjunct to the high-efficiency steam turbine unit. With it the very high condenser vacuum necessary is made possible. It is very simple and reliable, can be placed in any convenient position, and does not require special foundations. It involves no moving machinery, requires little attention, and can be operated to its fullest capacity without stressing. According to most authorities, it also compares favourably with the reciprocating pump in the matter of steam consumption, and the only reason why it was not adopted earlier was the lack of interest in the production of a very high vacuum, the special benefits of which were not then capable of exploitation.

The ejector may be defined as a pump in which the air to be compressed is entrained by one or more jets of fluid which pass through the entrainment space at a very high velocity, the combined stream then passing into a tube called the diffuser, in which it is compressed at the expense of its kinetic energy. Steam is generally the operating fluid since it is easily available, but some manufacturers abroad claim that the water jet has its advantages.

At first, the ejector had several disadvantages. When only the single stage was employed for compression ratios above 1:8, it was found that the diffuser throat diameter required for the ejector to start was greater than that necessary for working, so that a compromise had to be made, which was not always satisfactory. Greater success was obtained with the ejector in conjunction with the ordinary reciprocating pump as a "vacuum-augmenter," patented by Parsons, but it was not until the adoption of the two-stage plant that its use became widespread. Each unit has a compression ratio under the maximum 1:8, and an intercondenser is often introduced to deal with the operating steam of the first stage.

Very little is understood about the working of the ejector, though it has been generally accepted that the suction fluid is entrained by friction. This led to the adoption of a cluster of very small bore steam nozzles in the form of a ring, in place of the single nozzle, in order to increase the surface area of the jet. Lately the advantages of this type have been questioned, and the simple *friction* theory has consequently lost support. It is evident that with the theory in such an elementary state of development, the design of a plant to allow the entrainment process to be carried out in the most efficient manner is largely a matter for experiment. Naturally this has resulted in a complexity of designs. Water-cooling the throat of the diffuser is claimed to increase the efficiency and ensure stability, while the latter can also be obtained by the rather wasteful method of allowing atmospheric air to enter the diffuser at the throat and so ensure that the section is always filled.

The aim of this research has been first of all to obtain experimentally the most efficient arrangement, and by varying the leading dimensions in a series of tests to ascertain their effect on the operation of the ejector. With the dimensions which will give the best possible results for the particular conditions of working, the performances can be analysed to obtain the balance of energy, after assessing the losses which are known to exist. By this means it is hoped to reach definite conclusions in connection with the combining of the streams and their compression in the diffuser. The latter process is of particular interest, as the results of research on divergent nozzles by Professors A. L. Mellanby and W. Kerr,[1] and by the author,[2] show that the compression of a high-speed jet is accompanied by considerable losses.

Experimental Plant.—The air ejector generally operates on steam, which is usually available at a pressure suitable for working, but for

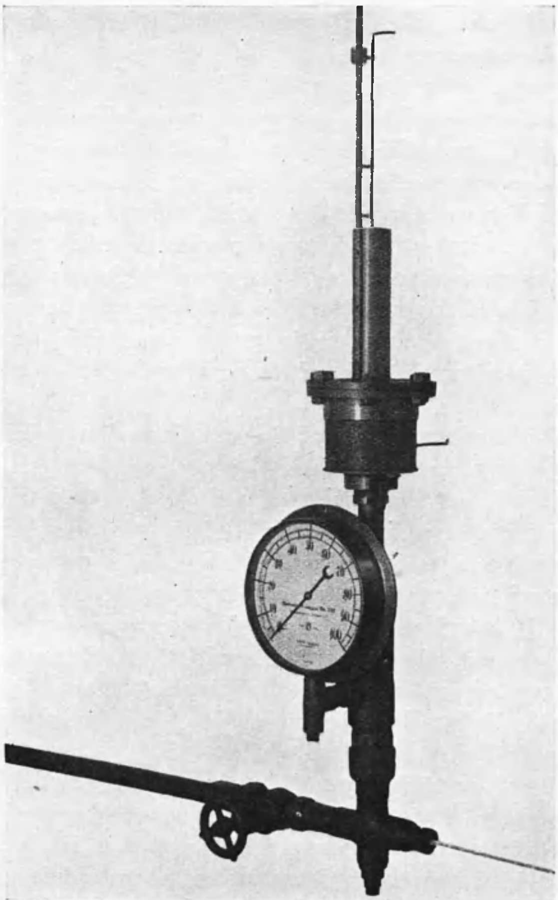


FIG. 1.—Experimental Air Ejector.

an investigation into the fundamental principles of the ejector the use of a single fluid both for operating and for compression would appear to be the most satisfactory arrangement, with a single stage of operations.

[1] This Journal, 1925.
[2] This Journal, 1926.

Air is supplied up to a maximum pressure of 100 lb. per sq. in. gauge in the laboratory by a two-stage compressor, which is quite suitable for the tests, and suction air can be drawn direct from the atmosphere.

Since the tests are intended to cover a wide range, more than one nozzle is required. Each nozzle is made of machined gun-metal, and is to standard so far as entry curve, throat diameter, and length are concerned. It is well known that a nozzle which over-expands, or one which under-expands considerably, loses efficiency, so that to accommodate the range in pressure ratios, the nozzles, three in number, have a different divergent taper. If, then, for each test the nearest under-expanding nozzle is chosen, the loss due to this cause will be negligible. The outlet area corresponding to a given ratio is calculated from theory,

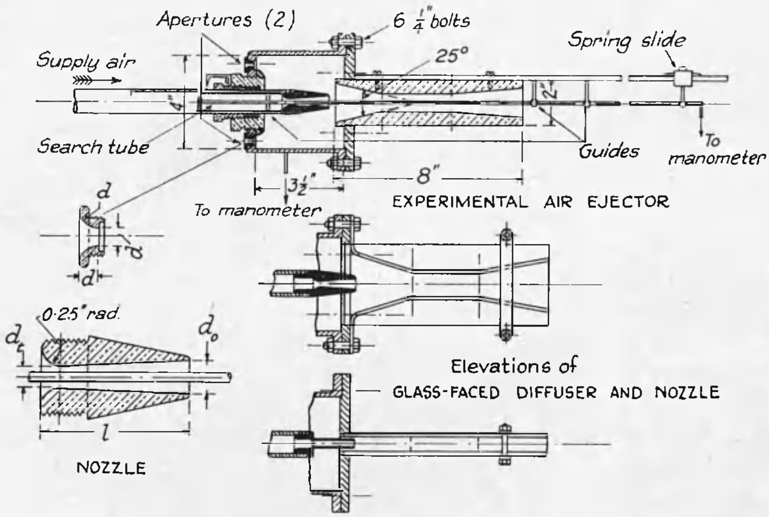


FIG. 2.

and allowance has to be made for a $\frac{1}{8}$-in. search tube which passes through the nozzle. Details are given in Fig. 2, where in order that the passage of the suction air may not be impeded, the nozzles are tapered down on the outside.

The nozzles are screwed to fit a $\frac{3}{4}$-in. pipe about a foot in length, which is machined and ground to fit a gland in the combining chamber—a circular box flanged to receive the diffuser plate. The diffuser is screwed into this plate, and the whole can slide along the nozzle supply pipe, on which a scale of inches is marked off. The apparatus is mounted on a long horizontal arm pivoted at a swivel joint, and through this passes the air supply from the compressor receiving-tank. This enables the ejector to move freely in a vertical direction, and so the reaction of the air leaving the diffuser can be measured on a balance, just as in the nozzle reaction tests described in the paper on nozzle compression losses previously referred to. A pressure gauge and a thermometer to determine the initial condition of the operating air are included.

Measurement of the suction air, or leakage air when referred to a condenser system, is effected by means of a series of apertures leading into the combining chamber. The arrangement of these apertures is found later to have no effect on the working of the diffuser, as demonstrated by setting them at the opposite end of the combining chamber, or by placing a baffle plate in the path of the incoming air.

Theoretically, the design of a diffuser should be followed out on the same lines as that of a nozzle, but the characteristics of a combined jet are insufficiently determinate to enable more than a rough estimate to be made. Thus, when the velocity of the combined jet is above the critical, the diffuser should be convergent at first and then divergent, for the conditions are just the reverse of those in the divergent type of nozzle; actually, this is modified in most types by a parallel or very gradually tapering extension of the throat section. The diffuser throat diameter can be only roughly approximated to by calculation, and no fixed rule can be given for the parallel length of the diffuser and its distance from the nozzle. For the particular requirements of the tests, it is, therefore, necessary to find out the "optimum" values of these dimensions for each case.

The overall length of the diffuser cannot be varied directly in a diffuser of the convergent entry and divergent outlet type, but its effect can be gauged in one of the simple parallel type by moving the nozzle along the diffuser axis. This is effected by screwing a parallel tube, 12 inches long, which can be bored out to various diameters, into the diffuser plate. Readings are taken of the chamber pressure for a range of distances of the nozzle outlet from the diffuser outlet, for various conditions of operation. The results show that unless the boundary of the jet is actually clear of the diffuser altogether the diffuser length has little influence on the working of the ejector, for in each case the chamber pressure rapidly attains a minimum steady value as the nozzle is brought further from the diffuser outlet. The point where this value is reached varies somewhat with different conditions, showing their effect on the spread of the jet, but it may be assumed with safety that a parallel diffuser length of about four diffuser diameters will cover all possibilities. This feature of a steady minimum chamber pressure above a certain critical length is of much interest and importance.

From a further preliminary investigation on the subject of diffuser diameter, the critical importance of this dimension is at once apparent, and, as there is no means of making it variable, the whole series of tests to be run on the ejector will depend on the choice of a set of diffusers of suitable diameter. Later experience will show that four diffuser diameters provide sufficient optimum values for the series of tests.

There remain to be decided the convergent entrance angle and the divergence of the diffuser. A cone angle of 25° seems to be practically universal in the first case—a result of experience—while the latter is more influenced by the delivery conditions than by anything else, and has no direct effect on the ejector working. The diffuser is bored out of a 2-inch gun-metal cylinder to the first throat diameter, and the convergent and divergent sections turned out to leave a parallel portion four times the diameter in length, and an outlet area of, say, three times that at the throat. Successive diameters are obtained by further machining, the divergent cone being turned down in each case until the parallel length is equal to four diameters and the outlet area ratio is the same as before.

Pressures along the axis of the nozzle and diffuser are determined by means of a simple search-tube apparatus. A slide attached to the end of the tube moves up and down a graduated guide-rod, which is screwed to the side of the diffuser tube. Comparative rigidity of the tube is obtained by fixing guides in such positions as will result in no interference with the air stream. A $\frac{1}{32}$-in. hole is drilled in the side of the tube, and connection is made to a mercury manometer. So that the nozzle area may remain unchanged by the movement of the tube, the latter must be of exactly uniform cross-section, and it must be so long that when the hole is opposite the diffuser outlet the tube remains in the lower guide.

A table of nozzle, diffuser, and suction aperture sizes is appended.

TABLE OF DIMENSIONS (in inches).

Nozzles.				Apertures.		Diffusers.	
	d_t	d_0	l		d		Throat Dia.
1	0·289	0·369	2·01	1	0·0693	A	0·629
				2	0·1245		
2	0·289	0·436	2·00	3	0·196	B	0·694
3	0·289	0·547	1·96	4	0·251	C	0·750
				5	0·299		
Search Tube Dia.—0·135				6	0·352	D	0·788

Theory of the Ejector.—In explaining the theory of the air-operated ejector, the following symbols for the properties of the air are used:—

P = absolute pressure in lb. per sq. in.
V = volume in cubic ft. per lb.
T = absolute temperature in °F.
v = velocity in ft. per sec.
M = flow of operating air in lb. per sec.
M' = flow of suction air in lb. per sec.
E = kinetic energy per lb. operating air.
r = pressure ratio referred to the nozzle supply pressure.
r_a = pressure ratio referred to atmospheric pressure.

The suffix (1) denotes the nozzle supply,
(2) the nozzle jet before combining,
(3) the combined stream before compressing,
(4) the discharge stream.

If the inlet velocity to the nozzle be neglected, the kinetic energy at the outlet is given by

$$E_2 = \frac{144n}{n-1} P_1 V_1 \left\{1 - r^{\frac{n-1}{n}}\right\} \qquad (1)$$

from standard nozzle theory, where n is the index of expansion. It has to be modified to allow for friction losses, which are represented by the term "k_n". This term is equivalent to an energy loss just as $\left(1 - r^{\frac{n-1}{n}}\right)$ is equivalent to the energy in adiabatic expansion,[3] so that, when n is given its value of 1·40 for air, and substitutions are made,

$$E_2 = 186\, T_1 \left(1 - k_n - r^{0.286}\right) \qquad (2)$$

The loss term is obtained from the dimensional equation

$$k_n = c \int_0^l \frac{p}{A} \left(1 - k_n - r^{\frac{n-1}{n}}\right) dx \qquad (3)$$

where $\frac{A}{p}$ is the hydraulic mean depth at a point, and dx an element of length. The constant c has a value for machined nozzles of about 0·005. Values of k_n, the loss up to a point, are assumed for the moment in the integrand, and the curve of $\frac{p}{A}\left(1 - k_n - r^{\frac{n-1}{n}}\right)$ is integrated for the whole length of the nozzle, as the jet is in contact with the walls throughout its length. A small factor, estimated at 0·004, is added to include entrance losses. A closer approximation to k_n is obtained from this integration, so that a repetition of the process will enable the loss to be determined more accurately.

[3] Mellanby and Kerr, "Steam Action in Simple Nozzle Forms."—Brit. Assoc.—Sect. G, Aug., 1920.

From the principle of the conservation of momentum, it is known that the sum of the momenta of the nozzle jet and the suction air is equal to the momentum of the combined stream, in any one direction. This assumes that there is no change in pressure and no losses due to combination or to reaction from the sides of the diffuser. Since the suction air quantity is small when compared with the area of entry into the diffuser, its momentum is quite negligible, so that the momentum equation is

$$\mathrm{M}v_2 = (\mathrm{M}+\mathrm{M}')v_3 \qquad (4)$$

$$\therefore \frac{v_3^{\,2}}{2g} = \left(\frac{\mathrm{M}}{\mathrm{M}+\mathrm{M}'}\right)^2 \frac{v_2^{\,2}}{2g} \qquad (5)$$

$$\therefore \mathrm{E}_3 = \frac{\mathrm{M}}{\mathrm{M}+\mathrm{M}'}\,\mathrm{E}_2 \qquad (6)$$

This is an expression for the kinetic energy of the jet which is available for compressing the combined stream.

The state of the combined stream before compression can be determined from theory, for the nozzle outlet conditions correspond to adiabatic expansion with friction reheat, and in the combining of the streams there is a further reheat equivalent to the theoretical energy loss in combining. A theoretical value for the throat area is often calculated from the velocity of the stream at this point, for

$$\mathrm{A}_3 = 144\,(\mathrm{M}+\mathrm{M}')\,\mathrm{V}_3/v_3, \qquad (7)$$

assuming that the diffuser throat is at chamber pressure, which is seen not to be the case from pressure readings recorded later, especially when the ejector is working at high vacuum.

The kinetic energy of the combined stream absorbed by compression up to atmospheric pressure is given by the equation

$$\text{Work done per lb. combined air} = \frac{144n}{n-1}\cdot \mathrm{P}_3\mathrm{V}_3\left(\frac{1-r_a^{\frac{n-1}{n}}}{r_a^{\frac{n-1}{n}}}\right), \qquad (8)$$

$$\text{or, Work done per lb. operating air} = \frac{\mathrm{M}+\mathrm{M}'}{\mathrm{M}}\cdot\frac{144n}{n-1}\cdot \mathrm{P}_3\mathrm{V}_3\left(\frac{1-r_a^{\frac{n-1}{n}}}{r_a^{\frac{n-1}{n}}}\right). \qquad (9)$$

The diffuser friction loss term k_d is calculated in the same way as that of the nozzle, by integrating the energy curve. Since the compression loss in the diffuser is unknown, it is possible only to approximate to the energy at a point, but the friction loss is a comparatively small one so that no great error arises. With the friction loss added, and substituting for n, the equation now becomes

$$\text{Work done per lb. operating air} = \frac{\mathrm{M}+\mathrm{M}'}{\mathrm{M}}\cdot 186\mathrm{T}_3\left(\frac{1+k_d-r_a^{\,0\cdot286}}{r_a^{\,0\cdot286}}\right) \qquad (10)$$

The residual energy in the stream is deduced from the measured reaction of the ejector. If R is the reaction in lb., then

$$R = (M + M')\,v_4/g \qquad (11)$$

$$i.e.\ E_4 = \frac{gR^2}{2M(M + M')} \qquad (12)$$

The work done in compression added to the residual energy E_4 should be equal to the available energy E_3 in equation (6) if there are no other losses in the operation, so that the difference will be equivalent to the losses unaccounted for.

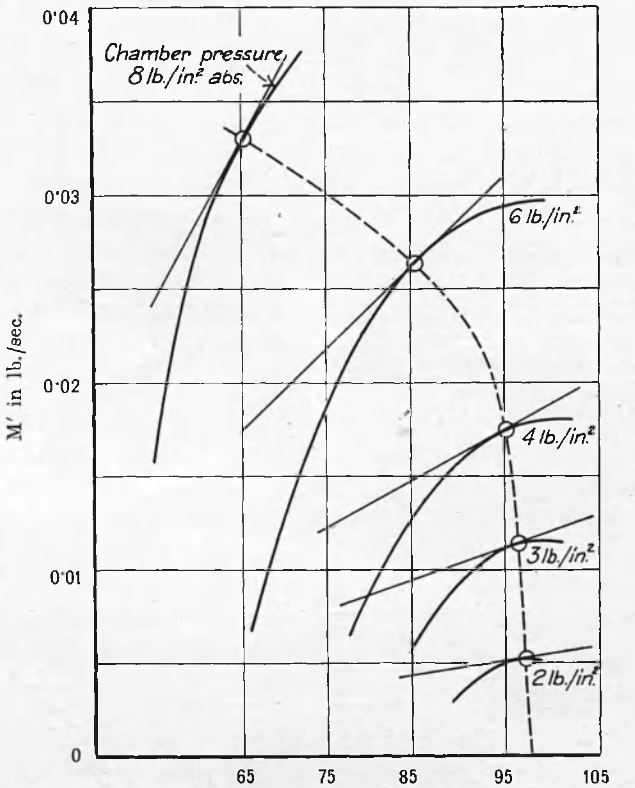


FIG. 3.—Curve of Maximum Efficiency, Diffuser "B."

Method of Testing.—Before proceeding with the tests, the temperatures for corresponding pressures at the nozzle inlet, with the compressor at normal running temperature, are plotted on a curve. The values are substituted in the flow formula $M = 0{\cdot}534\,P_1A_t/\sqrt{T_1}$, and with a discharge coefficient of 0·97 curves of the nozzle discharge on a pressure base are obtained.

The first object of the tests is to determine the operating conditions suitable for each diffuser diameter. With the first diffuser size, readings of the chamber pressure are taken at a diffuser distance from the nozzle

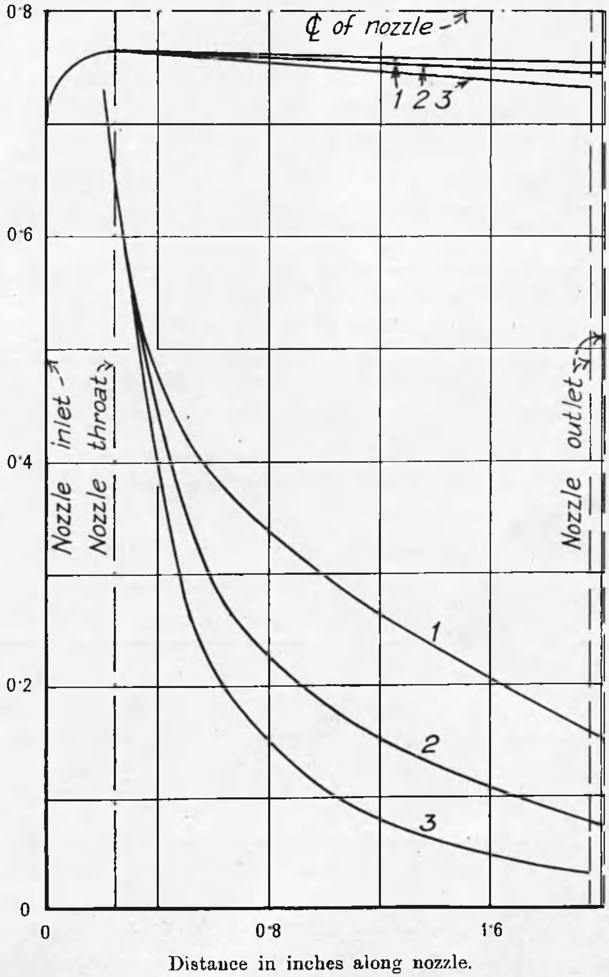


Fig. 4.—Pressure Ratio Curves, Nozzles 1, 2, and 3.

which gives the best results for a range of nozzle supply pressures and suction apertures. The chamber pressures are plotted on a base of suction quantity, and curves of constant nozzle pressure are drawn through the points. From these are derived curves of suction quantity at constant chamber pressure, on a base of nozzle supply pressures scaled off in units of energy expended per sec. by the operating air in expanding to atmospheric pressure. The points of contact of tangents drawn from the origin to the curves give readings of the suction quantity and nozzle pressure which represent the most efficient conditions of working for each chamber pressure, *i.e.*, the maximum ratio of work done to energy supplied. The reason for choosing the chamber pressure as constant will be evident, for the operating vacuum of a plant in practice is the first item to be fixed in specifications, the others being more or less a matter of choice. A curve drawn through the points of contact enables the

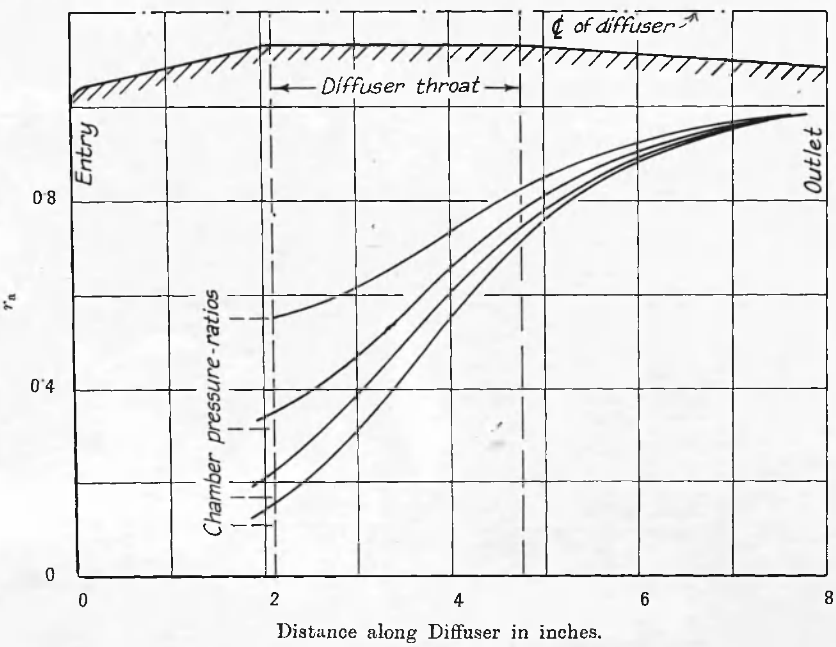


Fig. 5.—Pressure Ratio Curves, Diffuser "B."

nozzle supply pressures corresponding to the suction quantities given by the various apertures to be read off, as will be observed by referring to the specimen set of curves in Fig. 3.

Under the conditions of maximum efficiency the ejector is now operated, and readings taken of the nozzle supply pressure, the chamber pressure, the atmospheric pressure, the search tube pressures at suitable intervals along the axis of nozzle and diffuser, and, in addition, the reaction of the outlet stream. The procedure is repeated for each diffuser diameter, and the results tabulated. Thereafter the calculations indicated by theory are followed out to determine the balance of energy.

Pressure ratio curves for the nozzles and one of the diffusers are given in Figs. 4 and 5.

Effect of Dimensions.—The effect of the length of the diffuser has been dealt with in the preliminary tests, on the simple parallel type of diffuser, and the outstanding observation made was the minimum parallel length required. This value is modified to some extent by the amount of suction air; the more air, the more indefinite seems to be the boundary of the jet, and hence the required diffuser length. Increased nozzle pressure requires a slightly longer diffuser, but this is partly accounted for by an increase in the cross-section of the jet, and would not be so apparent if the diffuser diameter were enlarged accordingly. In this simple type of ejector it is the diffuser section that is usually fixed, as in some forms of exhaust blower, and the jet boundary has to conform to it by the natural process of spreading out until the walls are reached. Though this type of diffuser is used only where simplicity overrules economy, its operation is seen to give some idea of the nature of the ejector action, before a study of the more efficient type of diffuser is begun.

The most noticeable fact emerging from the dimension tests is the complete dominance of the diffuser throat diameter over the working of the ejector. It is the critical and ruling dimension. This is further emphasized when it is realized that the area of the stream varies under starting and abnormal working conditions, and there is no practical means of varying the diffuser area to correspond. For this reason the ejector, in the single stage at least, cannot be termed a flexible unit. On starting, since a large volume of air at atmospheric pressure has to be dealt with, experiment has shown that a much larger area is required than for normal running.

The sectional area of the jet is influenced by various factors. Thus, the nozzle throat diameter and the inlet pressure govern the flow of jet fluid, and to this has to be added the suction air quantity. Once the flow quantity is fixed, the area of flow is governed by the relation V_3/v_3, so that at very high vacua, while V_3 is increasing towards infinity and v_3 slowly towards a maximum, the required area will become correspondingly large, and will vary considerably for a very small change in vacuum. The importance of a correct diffuser area is illustrated in Fig. 3. It will be observed how rapidly the capacity of the ejector falls away when the nozzle supply pressure is reduced, at constant chamber pressure—the result of the jet area becoming too small for the particular diffuser size. Similarly, by increasing the pressure the diffuser becomes choked and functions improperly. It is obvious that the practice of making the diffuser oversize to assist in the starting of the ejector will mean a considerable sacrifice in efficiency when working under normal conditions.

The distance of the diffuser from the nozzle has not the same critical effect on the ejector operation. It is confined between two definite limits; when the diffuser is so far away that the jet has lost its definite boundary, on the one hand, and when it is so near that the jet does not strike the diffuser until it reaches the parallel portion, on the other. The curve in Fig. 6 records the chamber pressure for varying nozzle outlet position, and illustrates how comparatively steady it is over a considerable length. In general, it appears that for a constant nozzle pressure the distance is least when the ejector is working at high vacuum with little air suction, while increase of nozzle pressure, by increasing the dimensions of the jet, requires a proportionate increase in diffuser distance.

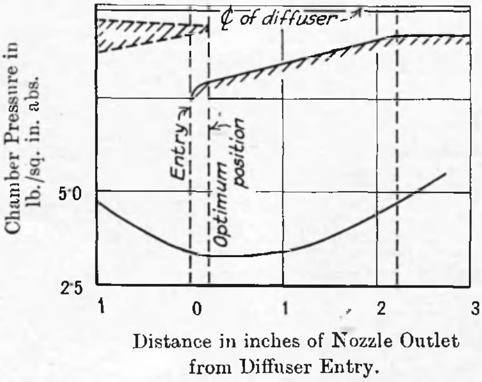


FIG. 6.—Effect of Distance between Nozzle and Diffuser. $P_1 = 75$ lb. per sq. in. abs.; Suction apertures No. 2.

The convergent entry angle to the diffuser, kept constant throughout the tests at a standard angle, cannot have much effect on the entrainment action. The divergent outlet is designed to suit the discharge requirements, and any possible effect it may have on the working of the ejector could be put to test by comparing a divergent outlet diffuser with a parallel one of the same overall length. Actually, the efficiency of the ejector is found to be quite unaffected by the rate of divergence.

From the combined results of the tests an instructive chart can be evolved, as shown in Fig. 7. It is intended to show how an actual plant could be designed from specifications, *i.e.*, the nozzle throat size, the supply pressure, the amount of air to be dealt with, and the vacuum from which it has to be pumped. From the constant nozzle pressure curves of vacuum against suction quantity, the capacity of each ejector unit in the plant can be deduced, and hence the number of units required to deal with the specified air quantity. The diffuser diameter and the throat distance from the nozzle to give the best results can then be read

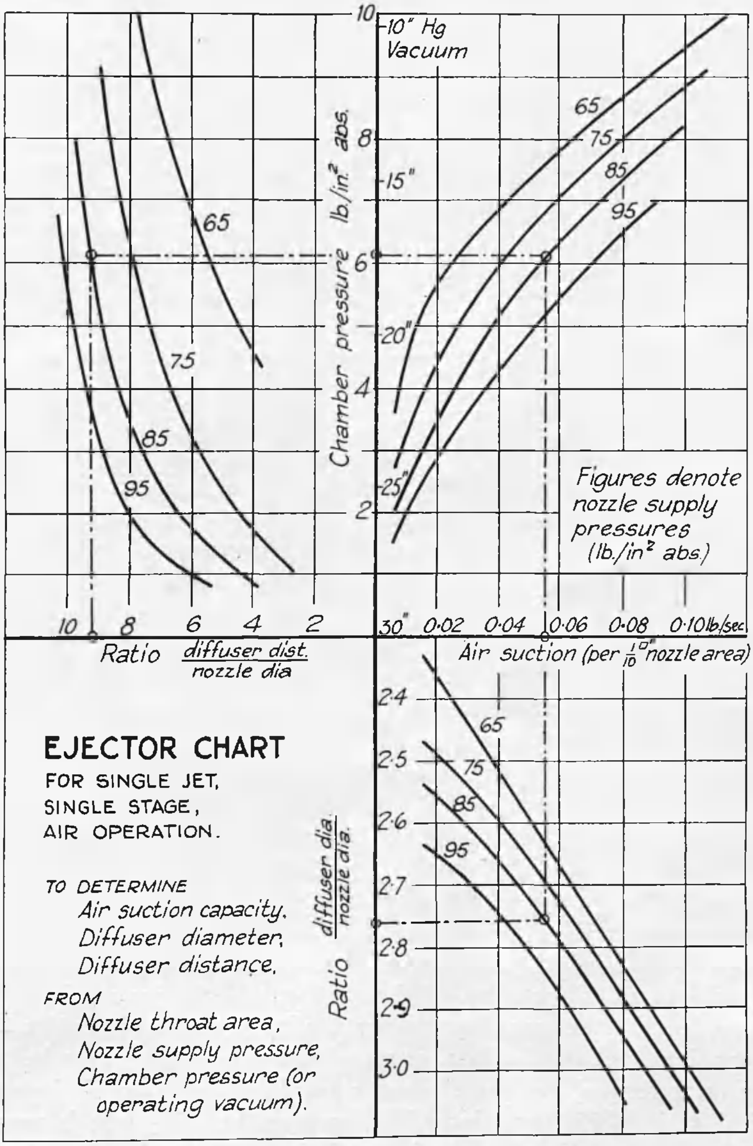


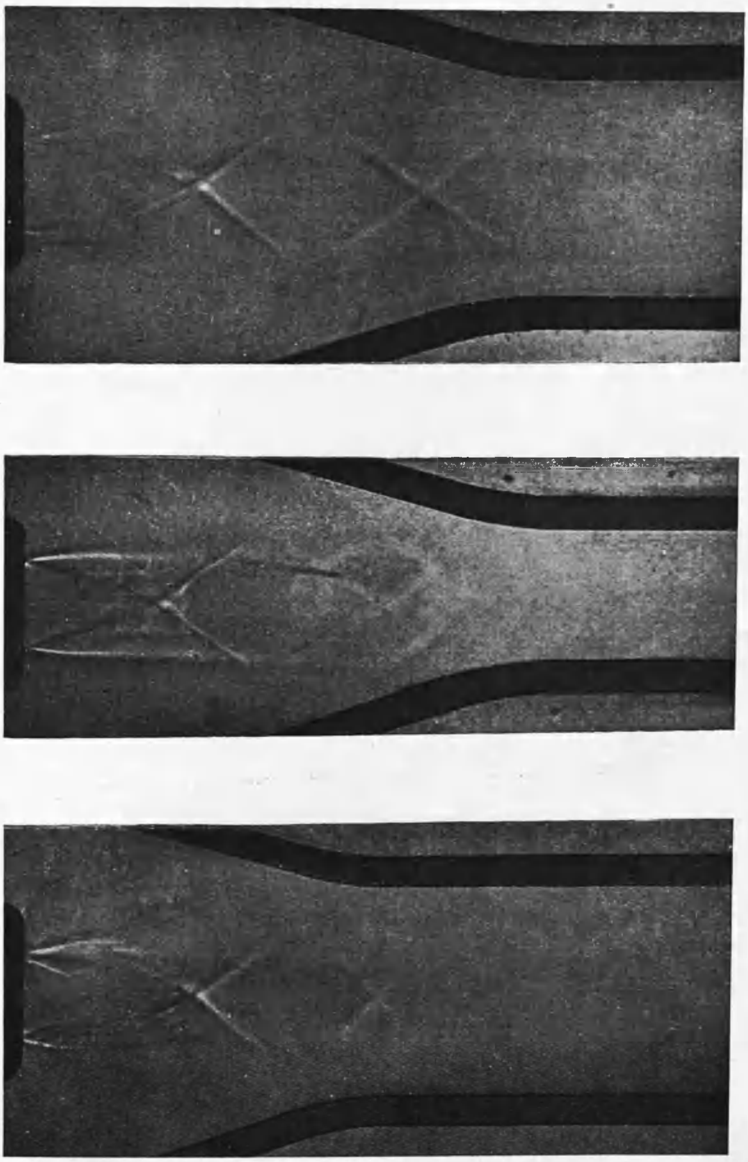
Fig. 7.

off on curves which have been derived directly from the results of the tests. For example, if the stage vacuum is to be 17·5 inches, and the nozzle supply pressure 85 lb. per sq. inch absolute, then the capacity and the dimensions for unit nozzle size are given by the dotted lines in the figure. The scope of this chart is, of course, limited by the range of the tests, and is applicable to air operation only, but there is no reason why the method could not be extended to cover practical requirements.

Further evidence of the effect of dimensions is provided by photographs of the jet, taken in the same manner as in the paper on compression losses in nozzles referred to previously. To obtain the photographs a glass-sided diffuser is required, so that the rays of light from an arc spot-lamp are enabled to pass through the combining chamber on to a screen beyond. It would be out of the question to have the diffuser of circular section, owing to the refraction caused by the glass, but a rectangular form, both of nozzle and diffuser, is sufficient to maintain a reasonable efficiency and conform closely enough with the actual ejector form for purposes of observation. The details of the diffuser and nozzle are shown in Fig. 2, each being adapted to fit the existing apparatus. The diffuser is framed by brass strips, $\frac{1}{2}$-inch wide, which are secured to the combining chamber in such a way that their distance apart, and hence the diffuser throat width, is adjustable. These strips act as distance pieces for two plates of $\frac{3}{16}$-inch optical glass, held in position by a slot in the chamber cover plate and by a clamp. As in the main tests, several nozzles of different divergence are provided, but in each case two of the sides are parallel and $\frac{1}{4}$-inch apart.

With a chosen nozzle pressure and air aperture, it is necessary first of all to find the diffuser diameter and distance to give the best results, and as these dimensions are easily varied on the apparatus, they can be determined directly by trial. An ordinary camera is used to take photographs of the shadows thrown on the screen by the refraction of the light passing through the jet. During exposure, the compressor supplies air to the nozzle at steady pressure, and great care is taken to prevent the condensation of moisture on the glass sides of the diffuser.

The first exposure represents the ejector working under the best conditions, as in Fig. 8 (*a*); in (*b*) the throat area of the diffuser is reduced by bringing the sides closer together, and in (*c*) the distance is reduced. In all three photographs the form of the jet can be followed, more or less, by the waves formed naturally at the nozzle outlet. These compressions and rarefactions, or sound waves, as they might be termed, are always formed at the outlet of a nozzle which expands beyond the critical ratio (0·528 for air). The greater the difference between the designed nozzle ratio of expansion and the actual ratio of expansion of the air, the more pronounced do the waves become, but even when the nozzle expands perfectly, the wave formation still appears. The nozzle



(*a*) Optimum diffuser distance and throat area. Chamber press. 7·0 lb. per sq. in. abs.

(*b*) Diffuser throat area reduced.

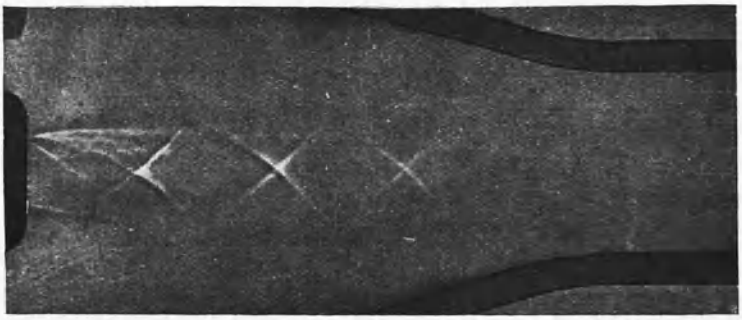
(*c*) Diffuser distance reduced.

Fig. 8.—Shadow photographs of waves in the combining chamber of a glass-faced ejector, to show the effect of dimensions on the jet form. Nozzle press. 90 lb. per sq. in. abs.; suction 0·022 lb. per sec.

outlet sections in the photographs are designed to under-expand slightly, which is usually the case in practice, and the resulting waves are fairly well defined.

No outward effect of the entrainment of the air can be observed in the photographs; in (*a*) the jet is not visibly disturbed right up to the throat section. It is interesting to note also the clearance between the visible boundary of the jet—that at which the waves are reflected—and the diffuser walls. In (*b*) the sectional area of the diffuser is considerably

(*a*) Expansion to atmosphere (sides of diffuser removed). Nozzle press. 95 lb. per sq. in. abs.



(*b*) Nozzle press. 65 lb. per sq. in. abs.; suction 0·046 lb. per sec.; chamber press. 10·0 lb. per sq. in. abs.

FIG. 9.—Shadow photographs of waves in the combining chamber of a glass-faced ejector, to compare the forms of free and combining jets of the same expansion ratio.

reduced, and its effect of choking the ejector is plainly seen in the setting up of a boundary disturbance surrounding the stream, in front of the throat. The considerable reduction in wave length is due to the increase in chamber pressure with less efficient working, and hence a reduction in the jet velocity and the amplitude of the waves. In (*c*) the effect of bringing the diffuser closer is not nearly so marked. Compression takes place sooner, of course, and the amplitude of the waves is seen to decrease rapidly as compression starts. The chamber pressure recorded is only slightly higher than that in (*a*), and this is reflected in the very similar wave form at the nozzle outlet in each case. It is significant to note that varying the diffuser distance gives a chamber pressure curve, as in Fig. 6, which shows no effect of the varying wave front at the throat section, thus proving that the waves do not influence the entrainment of air.

Losses and Entrainment Action.—The energy unaccounted for in the diffuser balance, which latter includes the nozzle and diffuser friction losses as well as the theoretical loss in combination, must be equivalent to some unknown loss which occurs between the nozzle outlet and the diffuser outlet. There may be some difference between the actual

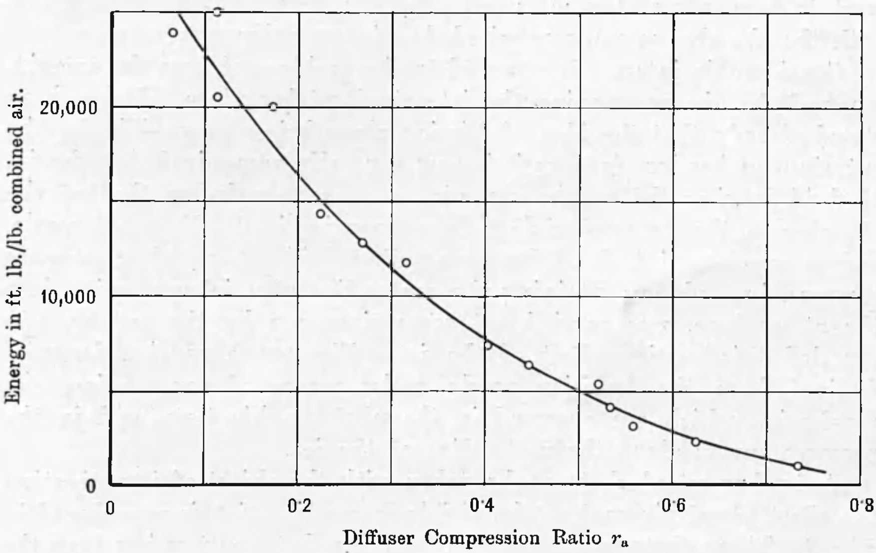


FIG. 10.—Energy unaccounted for.

combining loss and that obtained by following the law of the conservation of momentum, but it is most likely that the greatest loss occurs in the compression of the combined jet in the diffuser. It is impossible at this stage to calculate the magnitude of a compression loss, so that all that can be done is to plot the unaccounted loss on a suitable base, such as the ratio r_a of compression in the diffuser, for each test, and note the trend of the curve. Fig. 10 shows that as the ratio decreases the loss increases, a feature which is quite compatible with a compression loss.

The efficiency of the ejector is the ratio of work done on the entrained air to the energy expended overall by the operating fluid. When steam is used as the operating fluid in a condensing plant, this does not represent the actual thermal efficiency, however, as the heat of the steam in the second stage at least is recovered in the discharge to hotwell. For a steam-operated condenser plant it is customary to assume a nozzle efficiency of 90 per cent., and a diffuser efficiency of 50—75 per cent. The latter is defined as the ratio of the work done in compressing the combined jet to the difference between the kinetic energies of the combined and discharge streams. Taking a typical example from the tests, where 29·5 lb. of suction air are dealt with per hour at 25-inch vacuum, a nozzle efficiency of 91 per cent. and a diffuser efficiency of about 60 per cent. are noted, with an ejector efficiency of 5·5 per cent. The ejector efficiency as defined above is very misleading; for useful work is done only on the entrained air, and the proportion of this varies with the chamber vacuum required, but is generally very small.

Considerable interest is attached to the problem of how the entrainment of the air surrounding the jet actually takes place. It has been stated already that the idea of friction alone being responsible for the entrainment has lost favour; it is now generally recognized that the jet of fluid leaving the nozzle expands to a pressure below that of the chamber, on the same principle as over-expansion in the divergent nozzle, and so causes an inrush of the surrounding fluid. A complete pressure traverse, in a plane containing the centre line of a jet expanding into the atmosphere, was carried out some time ago by the author, and revealed this suction effect of the jet. The over-expansion is quite distinct from the familiar nozzle outlet waves of compression and rarefaction which are observed in the photographic tests, and in the pressure curves along the centre line of the jet.

A further set of photographs, taken with the object of investigating the effect of entrainment on the visible form of the jet, is reproduced in Fig. 9. These photographs show a comparison between two jets from the same nozzle working at the same expansion ratio, one jet expanding freely to atmosphere and the other entraining air in the combining chamber. In the first case (*a*) the sides of the diffuser are removed, leaving the glass faces in position, and the nozzle is supplied at the highest pressure obtainable. To find the supply pressure in (*b*) which will give the same expansion ratio, trial tests have to be carried out first, and the diffuser finally set to the correct width.

From the results the rather unexpected conclusion is reached that the jet within the visible boundary is quite undisturbed right up to the point of disappearance of the waves—very nearly to the diffuser throat—by the entrained air. No indication is given as to the manner of entrainment of the air, but there is sufficient proof that complete intermingling of jet and entrained air does not take place, until the throat section, at least, is reached.

The author acknowledges his indebtedness to Professor A. L. Mellanby, D.Sc., M.I.Mech.E., Professor W. Kerr, Ph.D., A.R.T.C., M.I.Mech.E., and Mr. J. S. Brown, M.B.E., A.R.T.C., for the use of plant and for helpful advice on various occasions.

RECOMPRESSION LOSSES
IN THE
DIVERGENT NOZZLE.

by

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(Thesis No.2 presented in compliance with the regulations for the Degree of Doctor of Philosophy of the University of Glasgow).

ROYAL TECHNICAL COLLEGE,

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ABSTRACT.

In this paper the recompression phenomenon in the divergent type of gaseous fluid nozzle is discussed with the aid of photographs taken through the faces of a glass-sided nozzle.

An attempt is made to explain the very heavy losses incurred in a compressing jet by applying the fundamental theory of fluid resistance. A fairly comprehensive range of nozzles is analysed, from data supplied by reaction and pressure ratio experiments, the analysis being greatly facilitated by the application of the loss factor introduced by Professors Mellanby and Kerr, which is particularly suited to the type of nozzle employed in the tests.

RECOMPRESSION LOSSES IN THE DIVERGENT NOZZLE.

INTRODUCTION.- Recompression is a well known feature of the divergent nozzle when it is operating with a back pressure greater than that for which it was designed. Expansion takes place independent of the back pressure down to a certain critical point, when the jet suddenly begins to compress, incurring very heavy losses in the process. In spite of the advisab::ility of avoiding it, recompression must occur under many conditions of working, such as in a nozzle which has to do duty over a comparatively wide range, or in one which is not operating under the conditions for which it was designed. Also, there is reason to believe that the over-expansion in a divergent nozzle is analogous to the action of the fluid in the ejector, for the natural tendency to over-expand can be controlled in the latter to induce and entrain fluid at the lower pressure and compress it in a suitable apparatus called the diffuser.

Until quite recently no attempt has been made to analyse the various kinds of loss occurring in a nozzle. The factors on which the losses depend undergo a very rapid change in the confined space, and to obtain reliable data under such conditions is a matter of considerable difficulty. Progress has been such, however, that the losses due to entrance effects, curvature and friction have now been isolated, while the latter loss has been expressed in a form which can be evaluated readily.

The losses due to recompression have not been given the same consideration, though Professors Mellanby and Kerr, in a paper published in the Journal of the Royal Technical College, 1925, deduced an expression to cover this loss in a general way. Available data on the reaction of divergent nozzles were employed, but, as was stated, the aim of the paper was more to indicate the possibilities of the line of attack adopted than to establish a formula, owing to the lack of sufficient data.

THE RECOMPRESSION EFFECT.- The fact that throat pressure does not depend on the back pressure in a divergent nozzle has been known for a long time, but no definite explanation of what happens to the jet has been put forward. One of the theories was that the high back pressure caused a piling up of the fluid, and the mass of the fluid following at a high speed came into contact with this slower moving body, resulting in a sudden rise of pressure.

Later investigation on jet compression showed that the outstanding feature, which seems to disprove the first theory, is the breaking away of the jet from the nozzle surface at the point of recompression. Up to this point, the jet follows the diverging contours of the nozzle, and so over-expands along the line of expansion which would be taken were the back pressure equal to or below the normal outlet pressure of the nozzle. This is the case for back pressures up to within about 0.8 of the initial, above which the throat pressure is affected. and rises rapidly for a small increase in the back pressure.

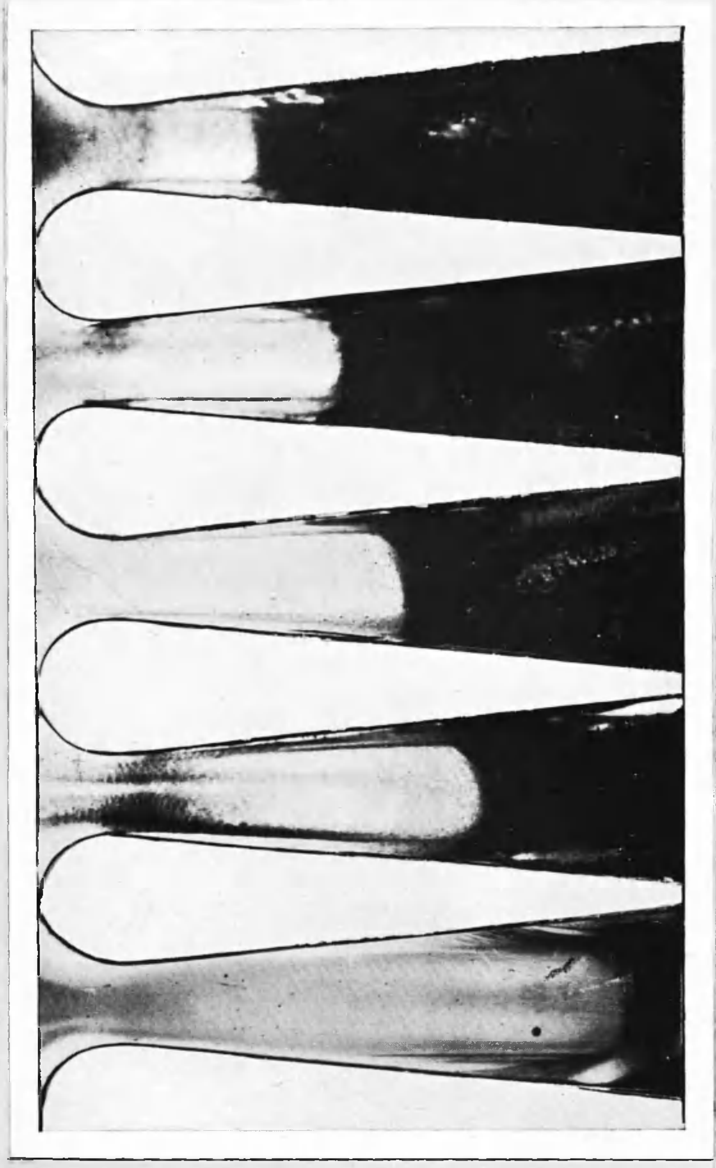


Fig. 1 – Air Jet Photographs, showing the effect of the jet on the oil-coated glass sides of a divergent nozzle.

The freeing of a jet of air is illustrated most convincingly in Fig. 1. The photographs were produced in a simple yet novel manner. They were taken through the parallel glass faces of a divergent gun-metal nozzle, $\frac{1}{4}$" square at the throat, tapering at 12° to 0·65" wide at the outlet, as seen in the diagram. These faces had been coated with a uniform film of very viscous oil, and the nozzle was placed between an arc lamp (with a ground glass screen in front to give a uniform intensity of illumination) and an ordinary camera. Exposures were made while air was being passed through the nozzle at a steady supply pressure, the air supply being obtained from a compressor through a receiving tank.

The light and dark portions in the photographs represent close contact with the walls and a free jet respectively, since the oil film is thinned or swept completely away by the contact of the jet, thus allowing light to pass through. It is noticed that in every case the jet frees itself completely from the nozzle walls at some point in the divergent portion, and that it seems to be undisturbed right up to this point. Other interesting features are that the critical stage commences at the nozzle boundary and spreads rapidly inwards towards the centre of the jet, and that in no case does the free jet seem to return to the nozzle walls before reaching the outlet. With a supply pressure P_1 of 10 lb. per sq. in. (gauge) corresponding to an outlet to inlet pressure ratio of 0·65, separation from the nozzle walls does not take place till well beyond the throat, showing without the aid of search-tube examination that the throat is

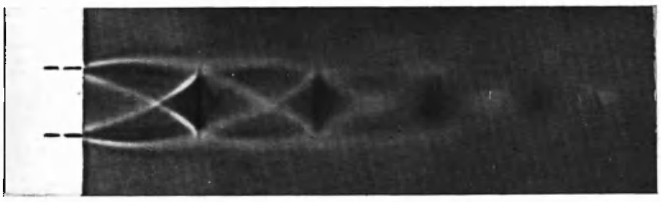


Fig. 2. — Shadow Photograph of Jet from a nozzle under expanding from 50 lb./in.^{2} gauge to atmosphere.

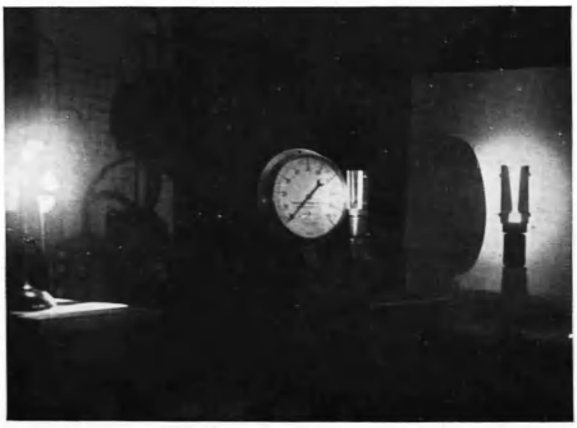


Fig. 3 — Apparatus for shadow photography of an air jet through a glass-sided nozzle.

unaffected up to a high outlet pressure ratio. At P_1 = 50 lb. per sq. in. (gauge), or pressure ratio 0.23, the nozzle is full almost to the outlet, where signs of disturbance make an appearance.

This method of featuring the recompression phenomenon gives no indication of the nature of the free jet beyond the critical point. Another method, that of passing light from a point source through the jet on to a screen, and so rendering the compressions and rarefactions visible by refraction of the rays, has been successfully employed in featuring the stationary wave formation outside the nozzle,[1] but such photographs taken through the nozzle reveal very little. The reason for this is that there is no sharp object, such as the outlet edge of the nozzle, to create a well-defined wave inside the nozzle. Instead, the converging streams meet in the centre and the ensuing shock results in the formation of waves of small amplitude, which soon die out. The illustration (Fig. 2) shows the complicated wave formation at the outlet of an under-expanding nozzle, as obtained by the refraction method.

In an effort to obtain clear photographs by "artificial" means, it was found that knife-edged projections 0·02" high on opposite sides of the nozzle at the throat section gave the best results. The apparatus employed was very similar to that in the previous experiments, except that the source of light was reduced to a pinhole in size, and the prints were exposed directly on the screen, as in Fig. 3. So long as the arc lamp was placed near enough to give sufficient light, the relative positions of arc, nozzle, and screen did not

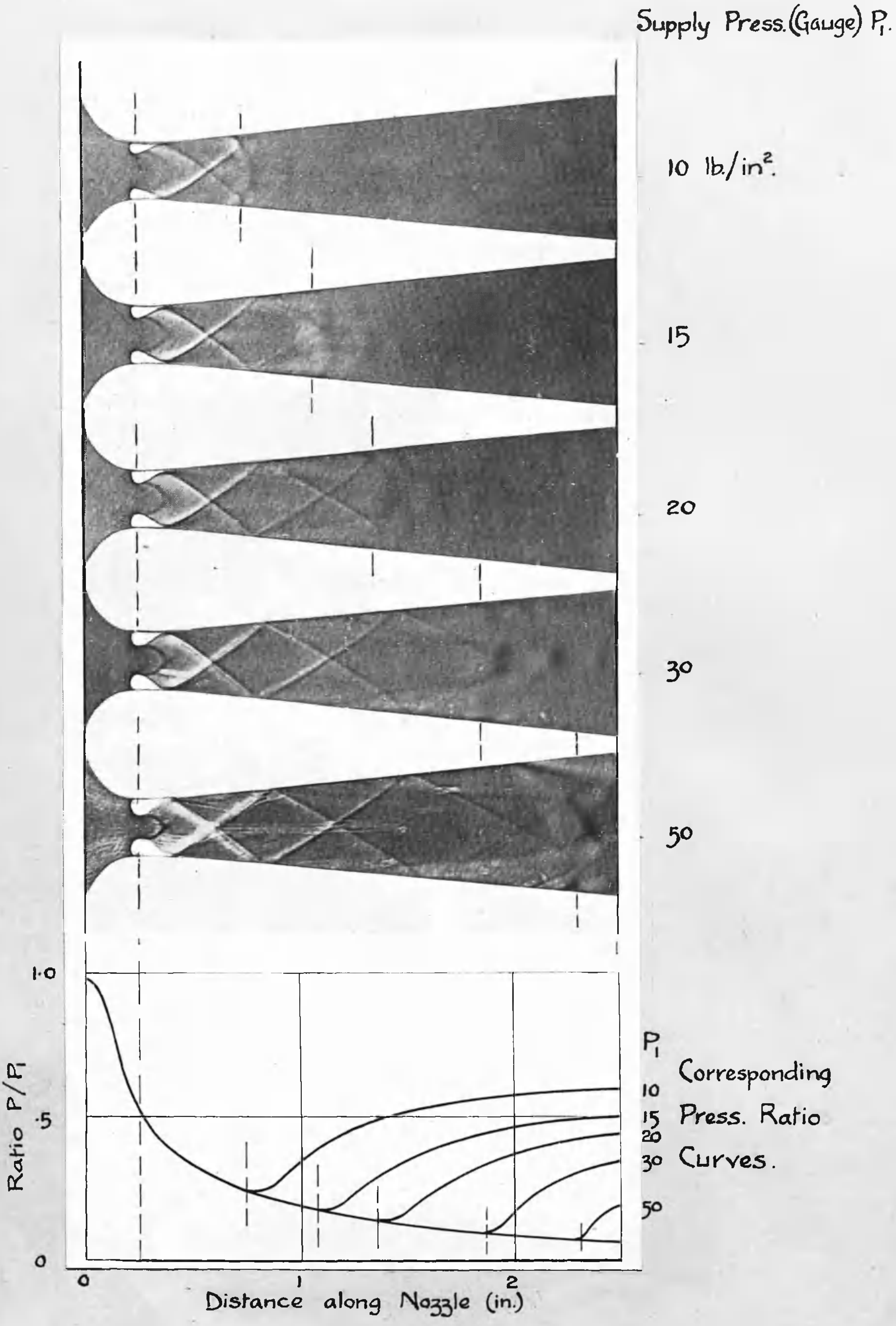


Fig. 4. — Shadow Photographs of waves produced in a glass-sided divergent nozzle by projections, 0·02 in. high, fixed at the throat.

matter much.. Great care had to be taken to eliminate oil and water vapour from the air supply, otherwise it would have been impossible to keep the glass faces of the nozzle perfectly transparent. The results of the experiment are illustrated in Fig. 4.

When the velocity of the jet is above the acoustic velocity, the projections cause waves to be propagated along the jet, the same as the compression wave (or sound wave, as it is sometimes termed) formed at the nose of a bullet, and their existence can be simply explained.[2] Referring to Fig. 5, if the bullet is moving with velocity $\underline{v}$ through the fluid in which v_c is the acoustic velocity, then in time $\underline{t}$ it will have moved a distance $\underline{vt}$, and the wave created by the compression in front of the object will have a radius $\underline{v_c t}$. It is evident that the succession of waves will be bounded by a cone, the angle α of which is such that $\sin \alpha = v_c / v$. The same wave is of course formed when the object is at rest and the fluid is in motion.

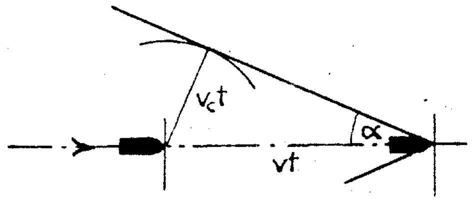


Fig. 5

The waves formed by the obstructions in the nozzle travel across the jet, to be reflected at the other side, but when the jet leaves the walls of the nozzle the wave fronts are left without any definite reflecting surface, although the borders of the jet fulfil this duty to a certain extent. This explains the sudden weakening of the waves at the point of recompression, and their ultimate disappearance. A comparison of the points of divergence obtained in this way with those in the previous experiment shows a close coincidence, in spite of the presence of the throat projections.

From the photographs it would appear that an enormous contraction of the jet takes place at the throat out of all proportion to the diminutive project:ions. If this were the case, the behaviour of the jet beyond the throat might be radically different from that with the true nozzle form, and so any deductions from the diagrams would be useless. To test the actual jet dimensions at the throat, discharge tests were carried out with and without obstructions. The flow quantities were found to be proportional to the respective throat areas, showing that no serious contraction could possibly occur. The apparent contraction must, therefore, be attributed to the magnifying effect of the refraction of the rays of light in passing through the areas of steep density gradient in close proximity to the obstructions.

Since the expansion is the same up to the compression point for all supply pressures above the critical, the wave formation should be identical, which is seen to be the case for all the photographs. The slightly curved form of each wave is due to the diverging streamlines of the jet. In the compressing part of the jet, the waves should still be visible up to the point where the fluid velocity is equal to the acoustic velocity, but the distance from the source of the waves and the imperfect reflecting surface of the free jet tend to damp them out before that.

The waves are so well defined in the expanding jet that there seems no reason why the relation $\sin \alpha = V_c/V$ should not be applied to find the jet velocity at any point in the stream, for the ratio V_c/V can be simply expressed in terms of the index of expansion, the pressure ratio, and a loss factor to be described later on. If α is measured as the angle between the wave and the direction of the jet, then the loss factor, and hence the efficiency, etc., can be determined immediately. It must be remembered, however, that the nozzle in question is only for purposes of demonstration, and would require to be modified before an analysis could be made. The effect of the obstructions on the flow could be reduced to a minimum by having several projections of almost negligible size, i.e., instead of having a strong initial wave to continue reflections along the jet, a series of waves would appear as very fine lines. An analysis of the free part of the jet presents more difficulties, but it is observed that the waves formed by the nozzle outlet in the free jet in Fig. 2 are quite definite, so that some method might be arranged to form waves of similar clarity inside the nozzle.

From these experiments it appears that the wave phenomenon must be very well developed in cast or roughly finished nozzles, where the resistance to the streams is considerable. Wave production, instead of friction alone, could be made to account for the very low efficien:
:cies of such nozzles when working at high velocities.

<u>MEASUREMENT OF THE RECOMPRESSION LOSS</u>.- Having studied the features of the recompressing jet, an attempt

can now be made to analyse the losses incurred by it. To provide the necessary data for investigating the nature of the losses, tests were made on a series of nozzles of varying divergent length and taper. These are the only features of nozzle design which, along with the nozzle diameter, can possibly have a direct influence on the recompression effect.

For such an investigation it was considered preferable to use air as the working fluid in place of steam, there being no supersaturation or condensation troubles, and no difficulty in disposing of the fluid after passing through the nozzle.

It has been pointed out by Professors Mellanby and Kerr, in their paper already referred to, that the losses can be obtained by a combination of pressure and reaction or impact measurements on the nozzles. The pressures can be read along the axis of the jet by means of a search-tube, and are necessary to define the compression range, as well as form a basis for the calculation of the frictional resistance of the nozzle walls. The discharge velocity of the nozzle, and hence its efficiency, can be calculated directly either by measurement of the nozzle reaction or the impact of the jet on a plate.

Until recently, it was thought that the impact method was quite impracticable, especially with high velocities. Trouble arose when it was found that, under the same nozzle conditions, the impact on the plate varies without apparent reason with change in its position, and at points the reading is actually in excess of the theoretical impact of the emerging jet. The presence of waves in the high velocity jet has also a very disturbing

Nozzle Dimensions. Table I

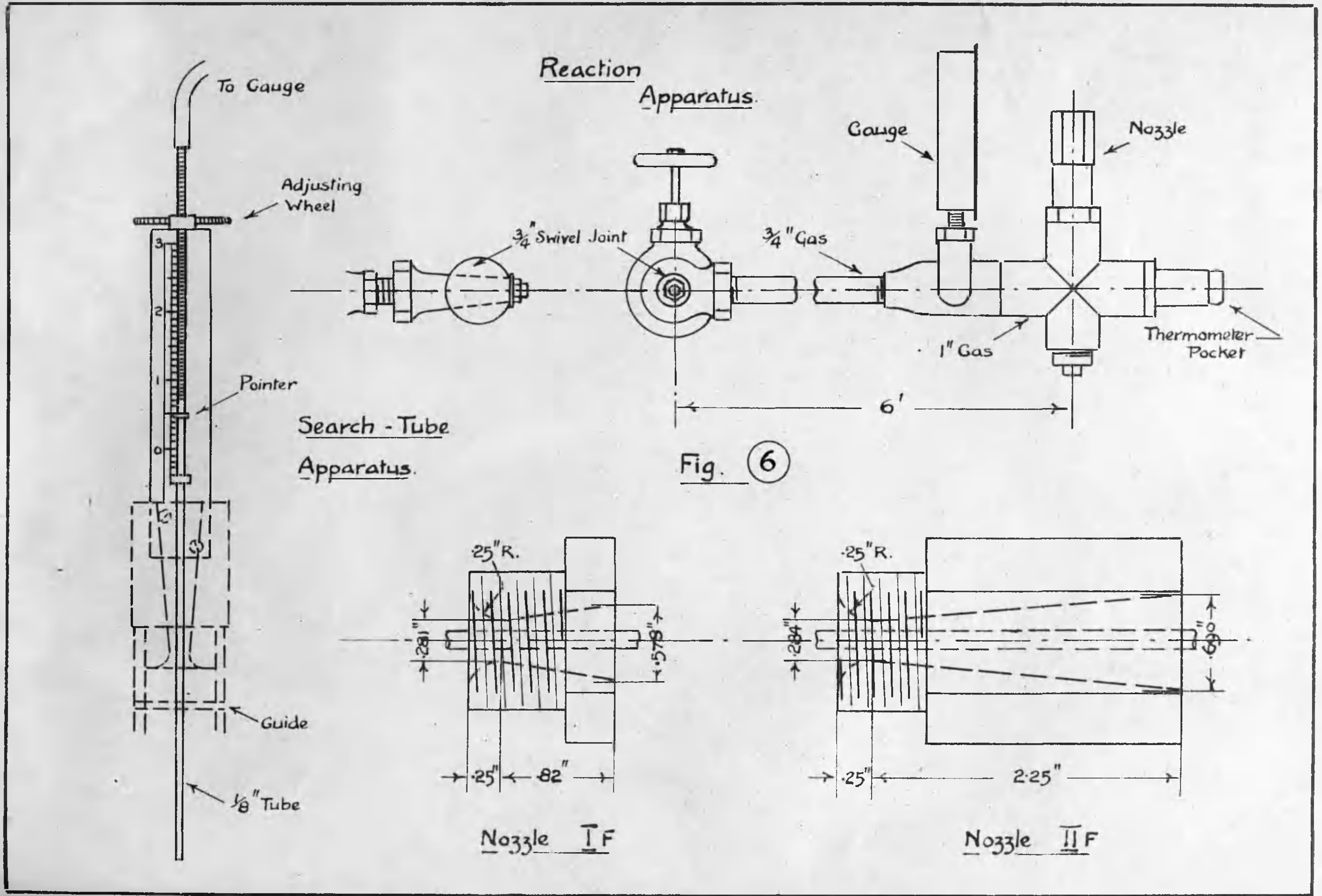
Nozzle		Throat Dia. (in)	Outlet Dia. (in)	ϕ Angle of Divergence	Ratio :- Outlet Area / Throat Area
Series I, Divergent Length, 0.82"	A	0·281	0·347	4° 36′	1·66
	B	0·281	0·374	6° 32′	2·00
	C	0·281	0·412	9° 08′	2·43
	D	0·281	0·446	11° 28′	2·90
	E	0·281	0·516	16° 20′	3·96
	F	0·281	0·573	20° 12′	4·95
Series II, Divergent Length, 2·25"	A	0·284	0·357	1° 52′	1·72
	B	0·284	0·383	2° 32′	2·02
	C	0·284	0·417	3° 24′	2·43
	D	0·286	0·485	5° 06′	3·34
	E	0·286	0·584	7° 36′	4·93
	F	0·286	0·690	10° 18′	6·98

influence on the impact, and curves on a distance base show that the waves are distinctly reflected in the readings.

The difficulties encountered at velocities below the critical were overcome after careful experiment by the Steam Nozzles Research Committee of the Institution of Mechanical Engineers. The excess impact reading was found to be due to the eddies formed by the fluid in escaping at high velocity over the edge of the plate. This and other eddy effects were avoided by covering the plate with a porous pad to "absorb" the velocity of the fluid, and by surrounding the jet with a cage of guides parallel to the plate.

Since this apparatus also tended to damp out the wave effects in a high velocity jet, it could be used with reasonable accuracy with a divergent nozzle. Tests made to determine the net reaction of a divergent nozzle, with this impact plate mounted on it, recorded an almost negligible positive difference. However, if careful consideration is given to the method of supplying the fluid to the nozzle, there can be no doubt that the reaction method of measurement remains the most suitable in the case of the divergent nozzle. It can be relied upon also to give readings of sufficient accuracy for determining losses of such magnitude as occur in a recompressing jet.

The nozzles used in the tests are gun metal, and straight, with circular cross section as in previous work of the kind. As may be seen from Table I, the entrance and throat sizes are the same for all the nozzles, the variables being the divergent length and taper. These

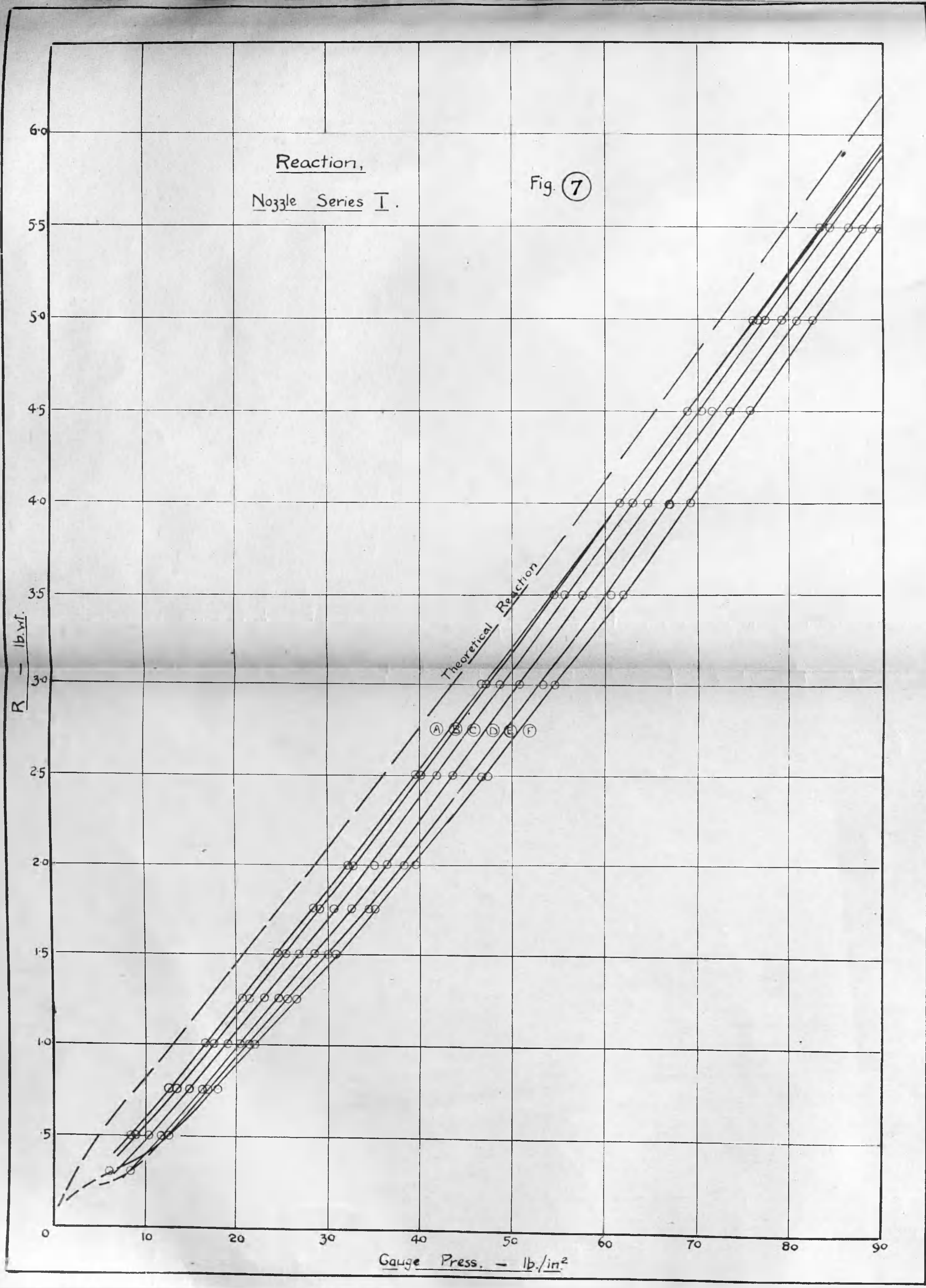

Reaction
Apparatus.
To Gauge
Adjusting
Wheel
3
2
1
0
Pointer
Search - Tube
Apparatus.
Guide
1/8" Tube
3/4" Swivel Joint
3/4" Gas
Gauge
Nozzle
1" Gas
Thermometer
Pocket
6'
·25" R.
·281"
·578"
·25"
·82"
Nozzle I F
·25" R.
·284"
·690"
·25"
2·25"
Nozzle II F
Fig. 6

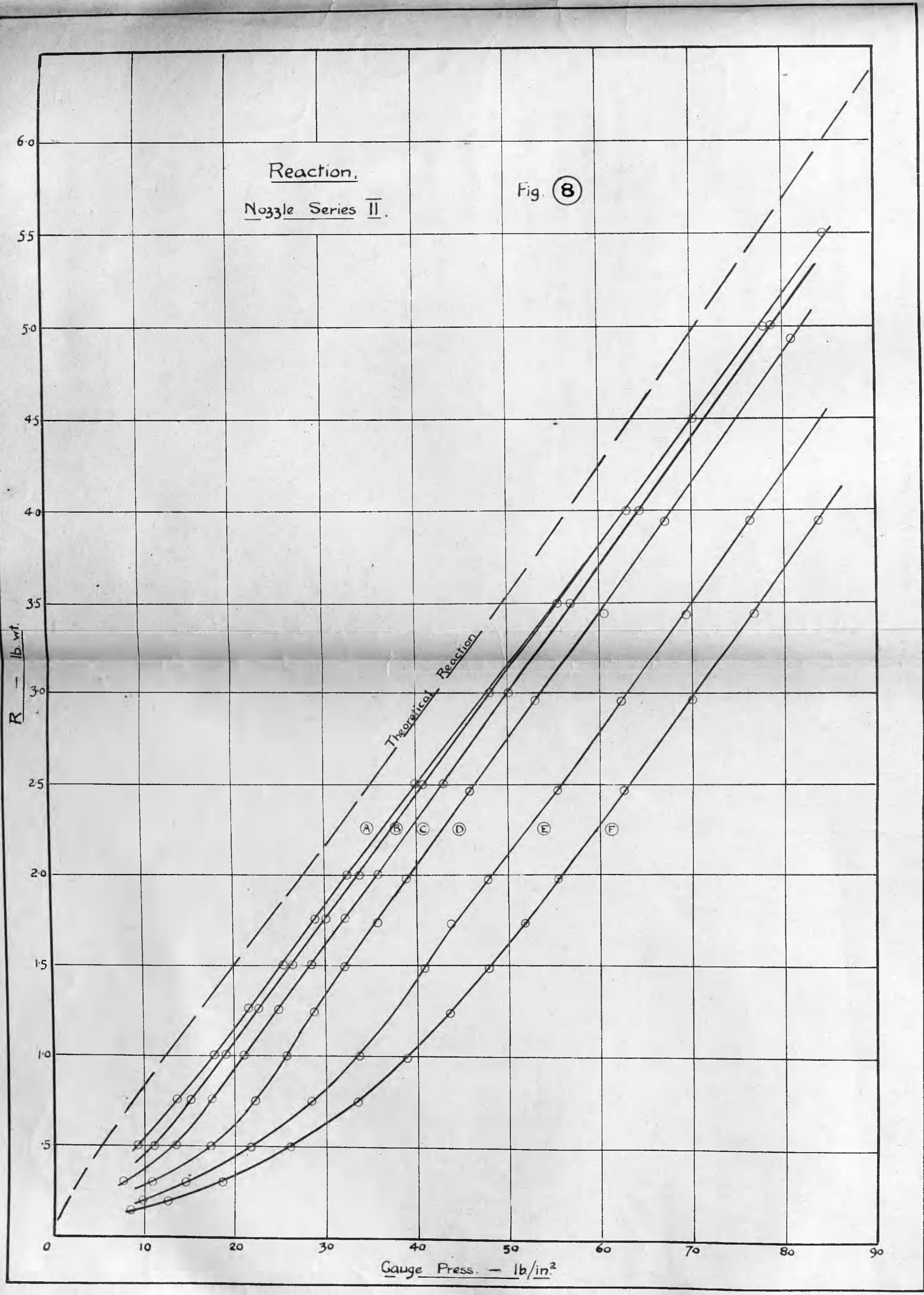
were chosen to provide a suitable range of ratios between the throat and outlet areas, at the same time including the range of taper usually to be met with in practice. The series consists of six nozzles of a short and six of a longer divergent length.

The type of reaction apparatus which could be employed left no choice in its general form - the necessity of an easy joint and a supply of air restricting the possibilities to an arm working on a swivel joint, through which passed the air supply. This was demonstrated by Morley in a paper on "The Flow of Air through Nozzles"[3], in which it was stated that a flexible connection to the nozzle could not give satisfactory results, owing to the thrust varying with the nozzle supply pressure.

The details of the apparatus are shown in Fig. 6. A length of flexible piping connected the air reservoir to the swivel joint through a stopcock, and the joint was attached to a pedestal in such a manner that it could swing freely on a vertical axis. The nozzle box consists of a cross-piece of 1¼" internal diameter, which was found to form a sufficiently large reservoir for the air. It was fitted with a guage and a thermometer pocket in the positions shown, and connected to the swivel joint through a six-foot iron pipe.

Reaction could be measured directly on a balance set below the nozzle, and the latter was fitted with a "dummy" search-tube to compensate for that required in taking the nozzle pressure readings later on. The effect of the search-tube in making the nozzle annular in section does not alter the reaction readings, as comparative tests with annual and circular sections gave reactions proport::ional to the areas, after allowing for friction.


Reaction,
Nozzle Series I.
Fig. 7
Theoretical Reaction
A B C D E F
R — lb.wt.
Gauge Press. — lb./in.²
0
.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
4.5
5.0
5.5
6.0
10
20
30
40
50
60
70
80
90


Reaction.
Nozzle Series II.
Fig. 8
Theoretical Reaction
A
B
C
D
E
F
R — lb. wt.
6·0
5·5
5·0
4·5
4·0
3·5
3·0
2·5
2·0
1·5
1·0
·5
0
10
20
30
40
50
60
70
80
90
Gauge Press. — lb/in²

Before taking a set of readings, lubricating oil was introduced into the air suction of the compressor, to provide a thin film of lubricant on the surfaces of the swivel joint. The zero balance reading was taken as that load, which, when applied to the nozzle scale pan, brought the balance pointer to the mid position. The errors due to friction are reduced to a minimum if the test readings are taken in the same way, i.e., by placing a load on the weight pan and increasing the supply pressure until balance is obtained. Below 10 lb. per sq. in. gauge supply pressure the accuracy of the readings cannot be guaranteed, so that the chosen range of pressure was from 10 to 70 lb. per sq. in. gauge for each nozzle. The results are plotted direct in Figs. 7 and 8.

As in Morley's experiments, the nozzle discharge was measured by the rate of pressure and temperature change in allowing the air to escape from the reservoir through the nozzle. If V is the volume of the reservoir, in c.ft., and M, P_1, T_1, the mass in lb. per sec., the absolute pressure in lb.per sq. in., and the absolute temperature in ^{o}F respectively of the air in the reservoir at time t secs., then

$$M = \frac{144 P_1 V}{RT_1}$$

Since the temperature of the reservoir was practically constant after the first minute, the rate of discharge

$$\frac{dM}{dt} = \frac{V}{RT_1} \cdot \frac{dP}{dt} = \frac{V}{RT_1} \cdot a P_1$$

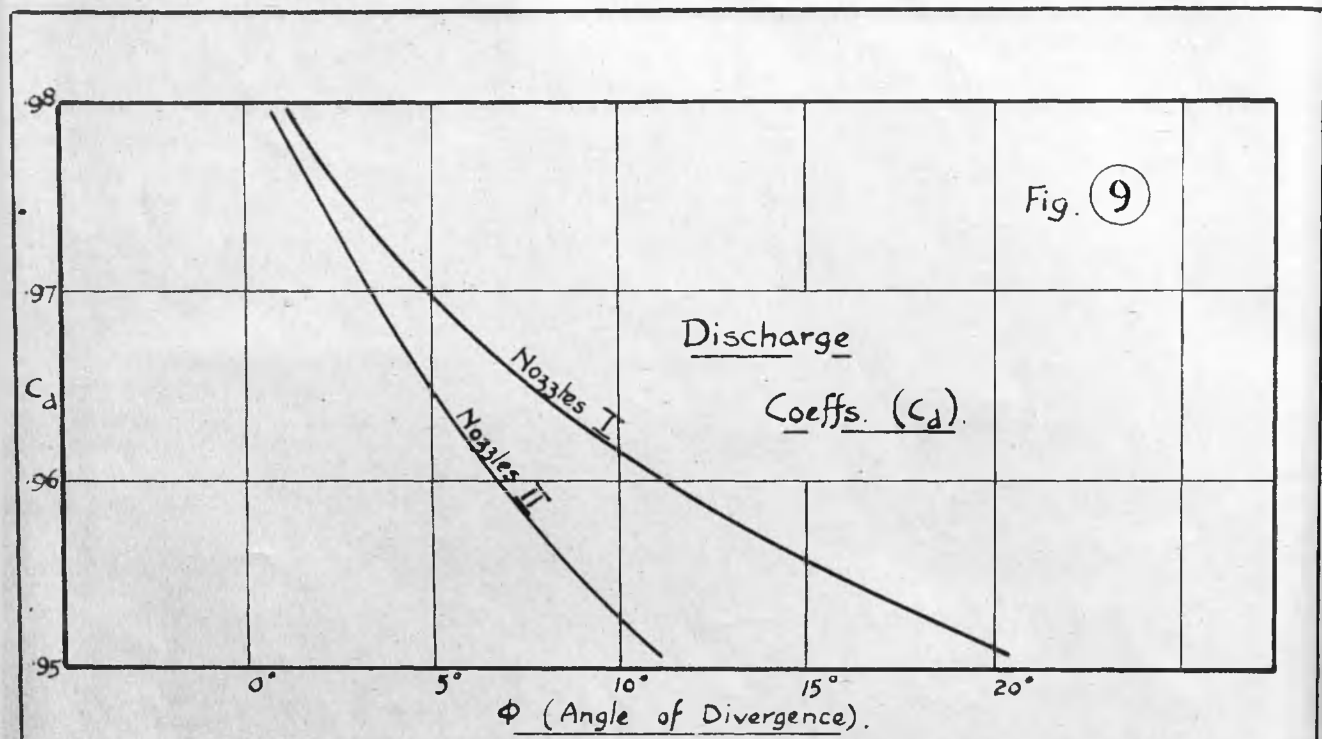
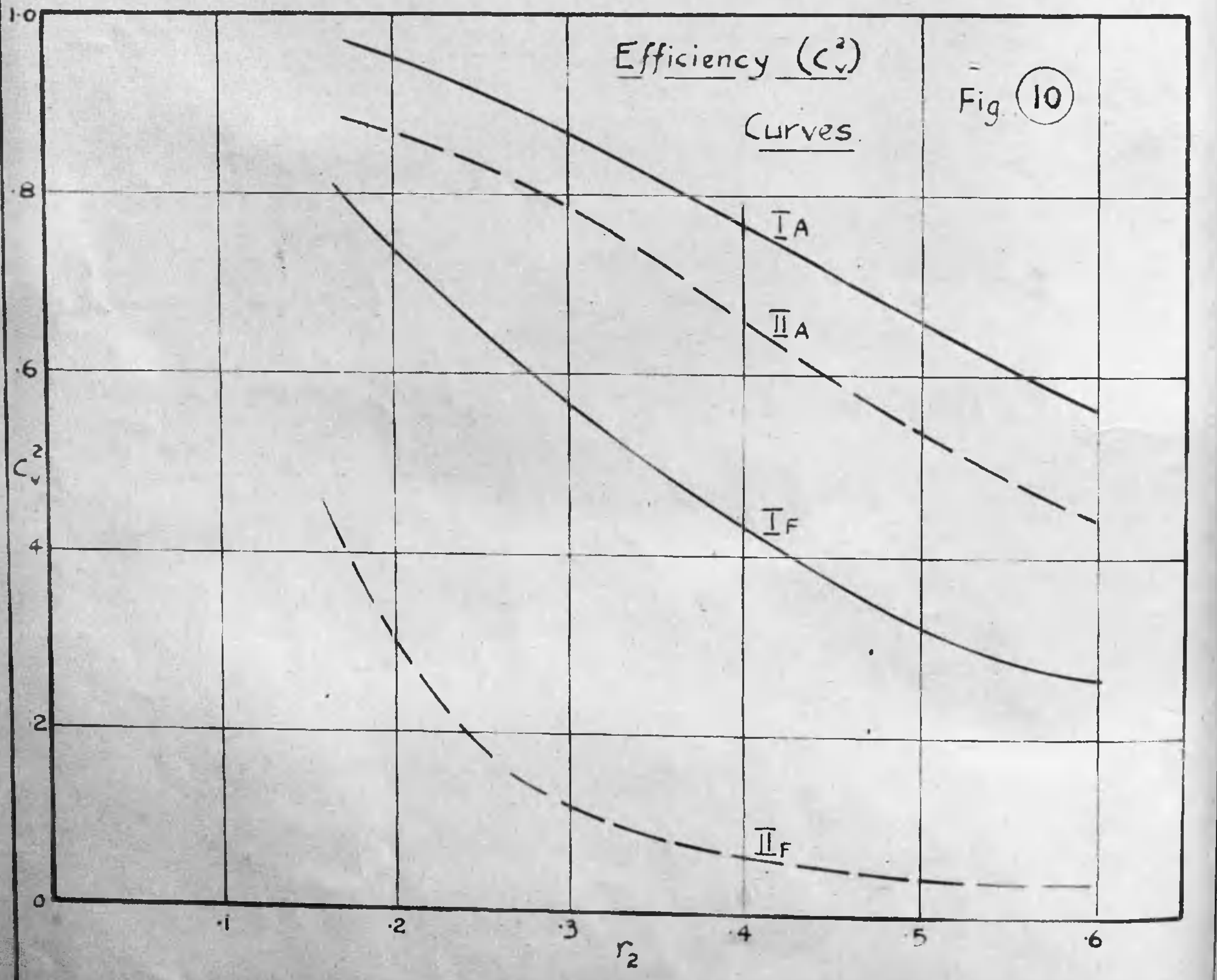


Fig. 9
Discharge
Coeffs. (C_d).
Nozzles I
Nozzles II
C_d
·98
·97
·96
·95
0°
5°
10°
15°
20°
Φ (Angle of Divergence).


Efficiency (C_v^2)
Curves.
Fig. 10
I A
II A
I F
II F
C_v^2
1·0
·8
·6
·4
·2
0
·1
·2
·3
·4
·5
·6
r_2

where $\underline{a}$ is the slope of the curve of log P_1 and $\underline{t}$.

Now the flow through the nozzle is given by

$$M = \frac{\cdot 534 P_1 A}{\sqrt{T_1}} ,$$

where A is the throat area in sq. in., so that the coefficient of discharge

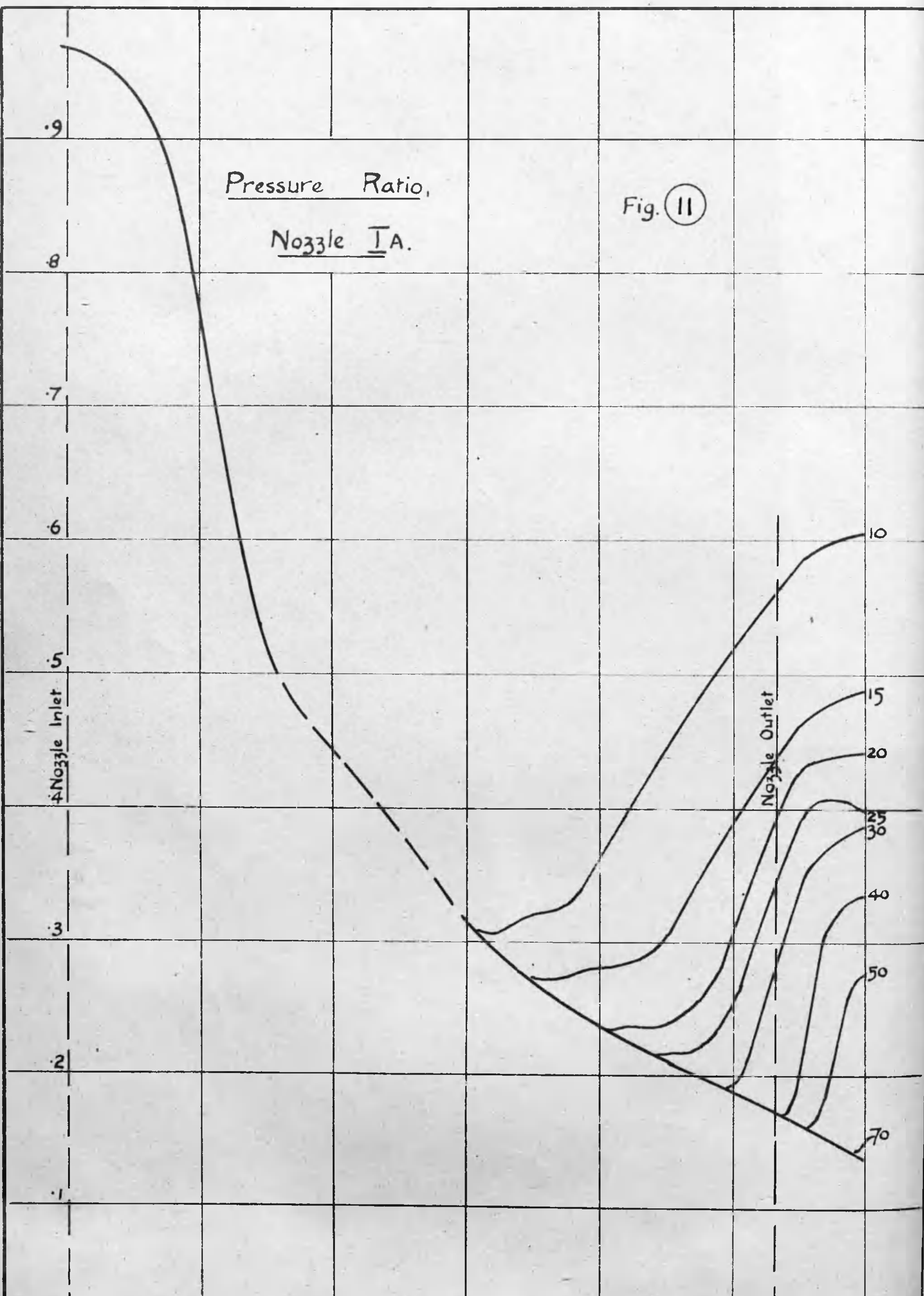
$$c_d = \frac{144 Va}{\cdot 534 AR\sqrt{T_1}} .$$

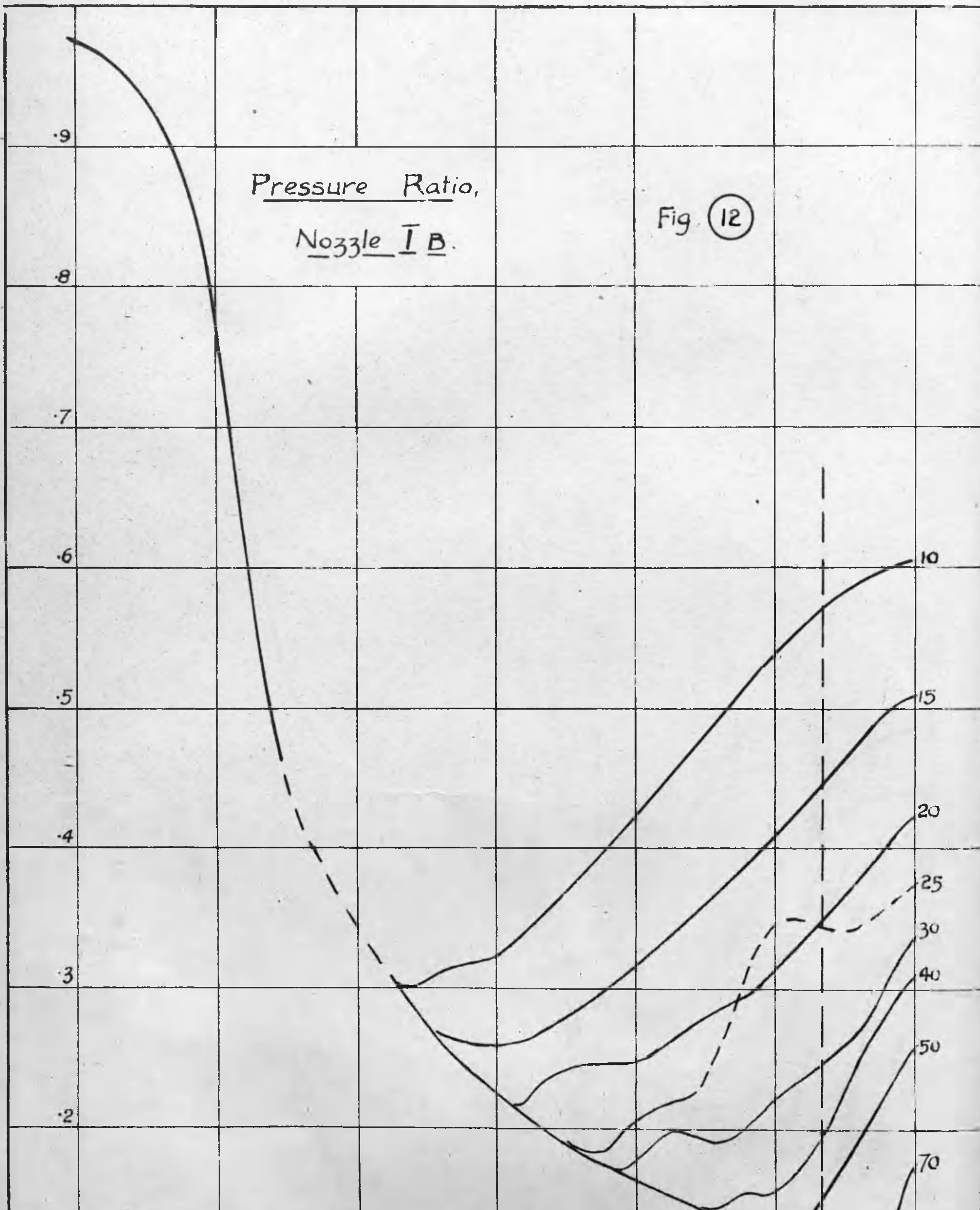
The coefficient was found to have a practically constant value for each nozzle, over the range of pressure ratios, so that the nozzle series could be represented by curves on a base of nozzle taper. Fig. 9 shows that the discharge decreases fairly rapidly with increasing taper, and also that the shorter nozzles in the first series have a higher coefficient for the same rate of divergence.

Since the nozzles were operating into the atmosphere, a search-tube apparatus for pressure measure:
:ment presented no difficulties. Accordingly the very simple gear illustrated in Fig. 6 was set up. The $\frac{1}{8}$" diameter search-tube was threaded for some distance at the top to receive the adjusting wheel, and carries a pointer which records on the scale behind. The mounting was screwed on to the side of the nozzle, additional rigidity of the tube being obtained from a guide fitted inside the nozzle box. A $1/32$" hole two inches from the end of the tube is in connection with either a pressure gauge or a mercury manometer, as required.

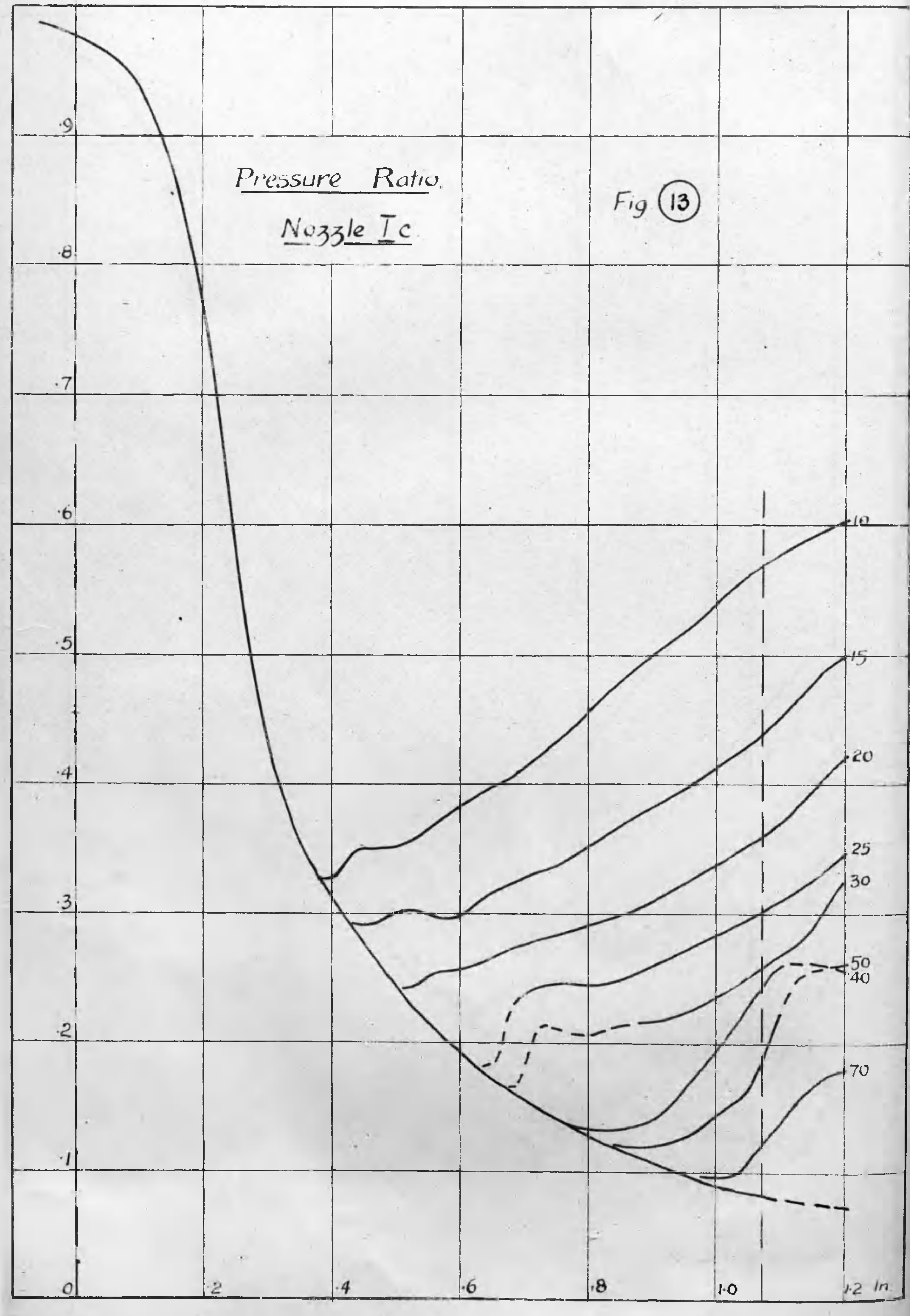
The pressure ratio curves are reproduced in Figs. 11 to 16 and Figs. 17 to 22.

<u>CALCULATION OF THE LOSSES</u>.- The theoretical reaction is equal to the rate of change of momentum of the jet, i.e., the product of the mass flow per unit

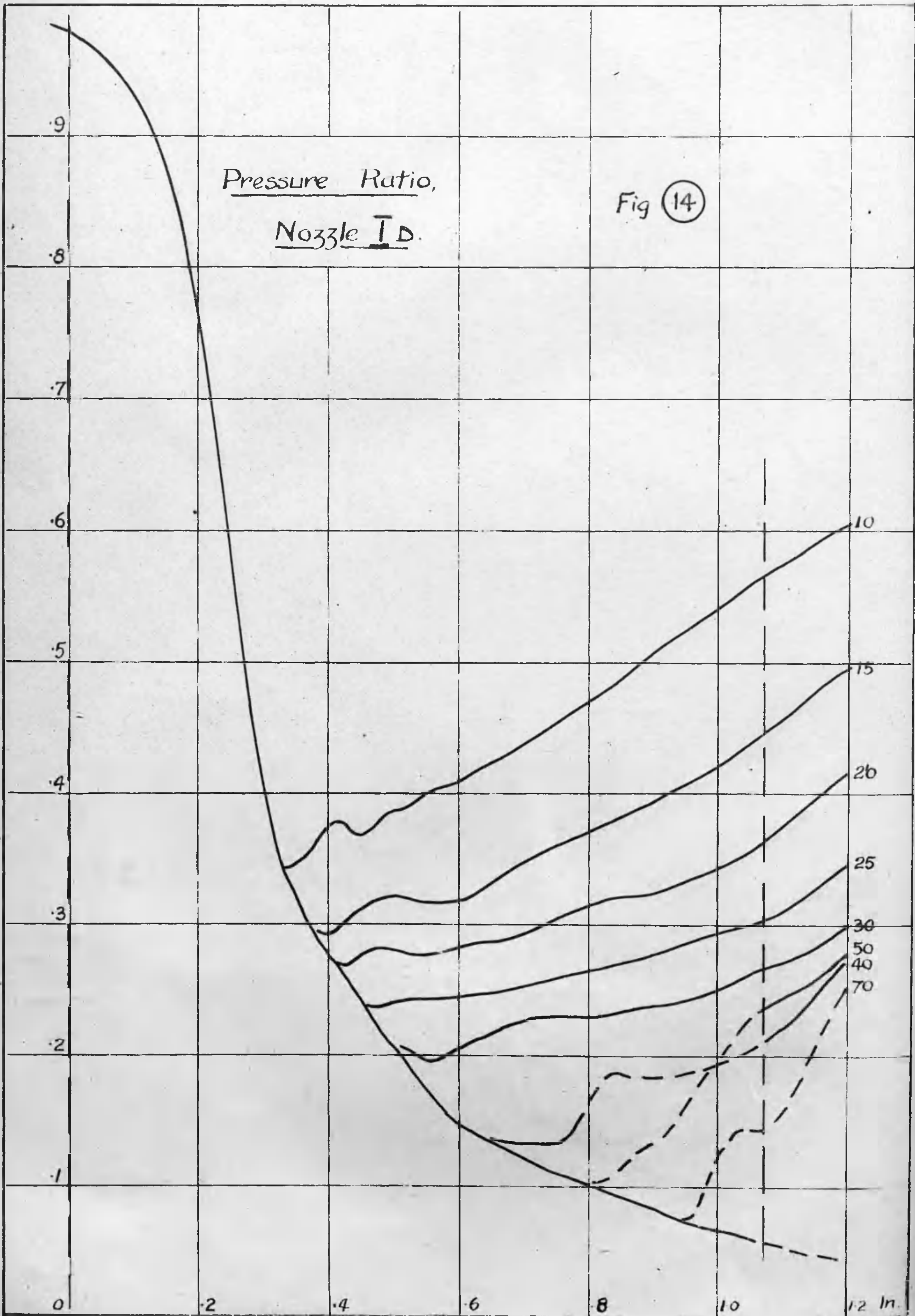

Pressure Ratio,
Nozzle I A.
Fig. 11
·9
·8
·7
·6
·5
·4 Nozzle Inlet
·3
·2
·1
0
·2
·4
·6
·8
1·0
1·2 in.
Nozzle Outlet
10
15
20
25
30
40
50
70



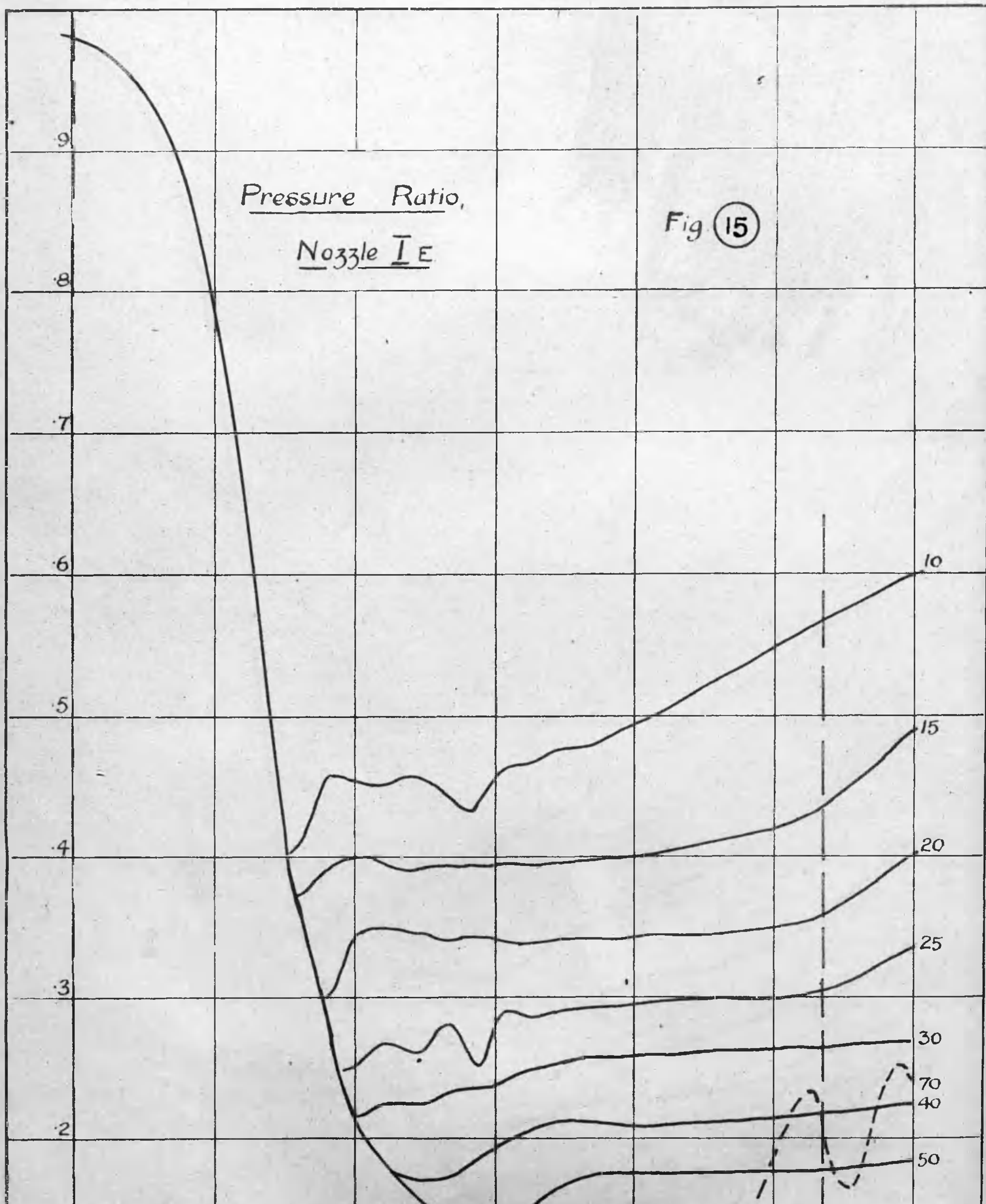
Pressure Ratio,
Nozzle I B.
Fig (12)
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·2
·4
·6
·8
1·0
1·2 In
10
15
20
25
30
40
50
70



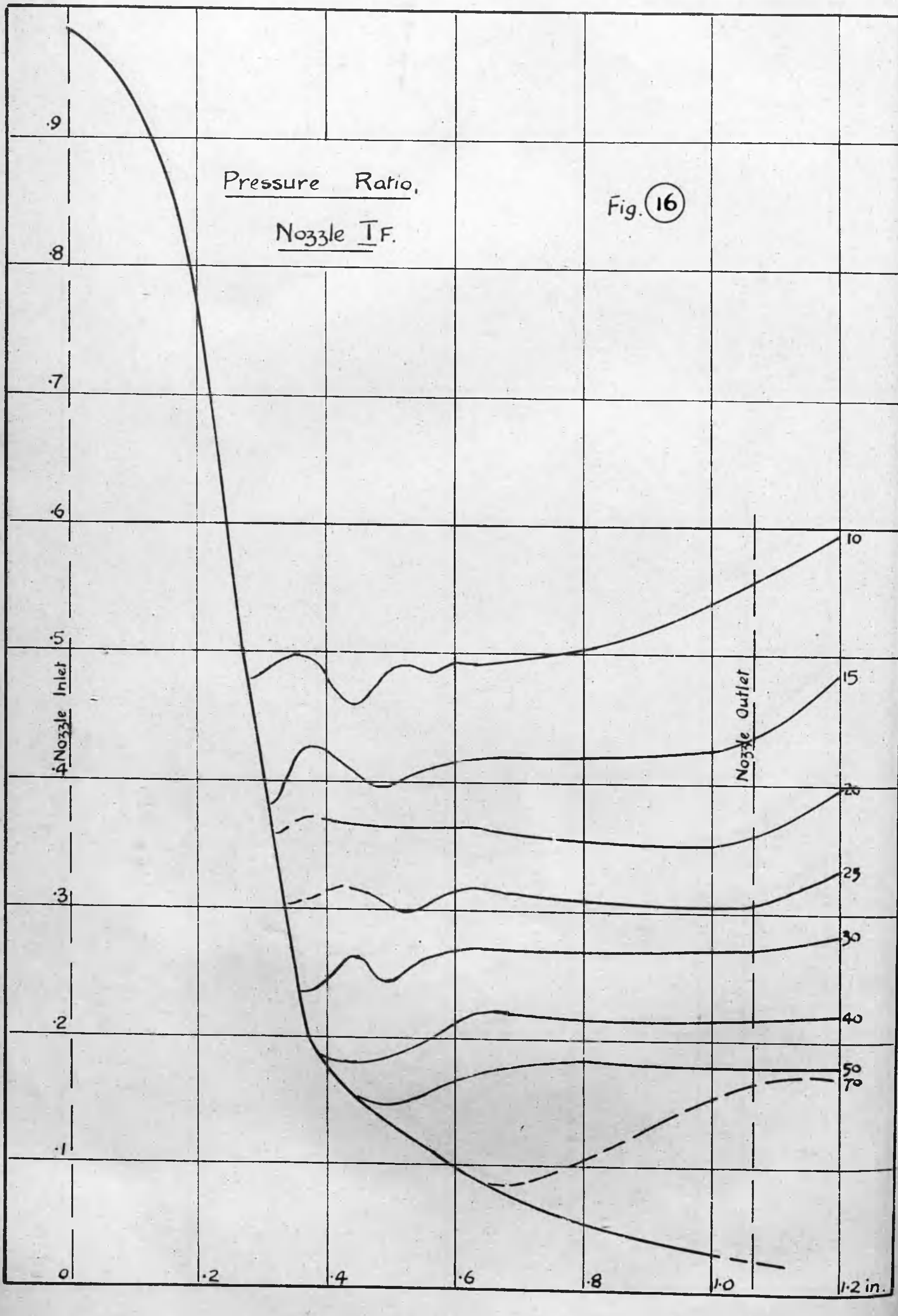
Pressure Ratio.
Nozzle Ic
Fig 13
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·2
·4
·6
·8
1·0
1·2 In.
10
15
20
25
30
50
40
70



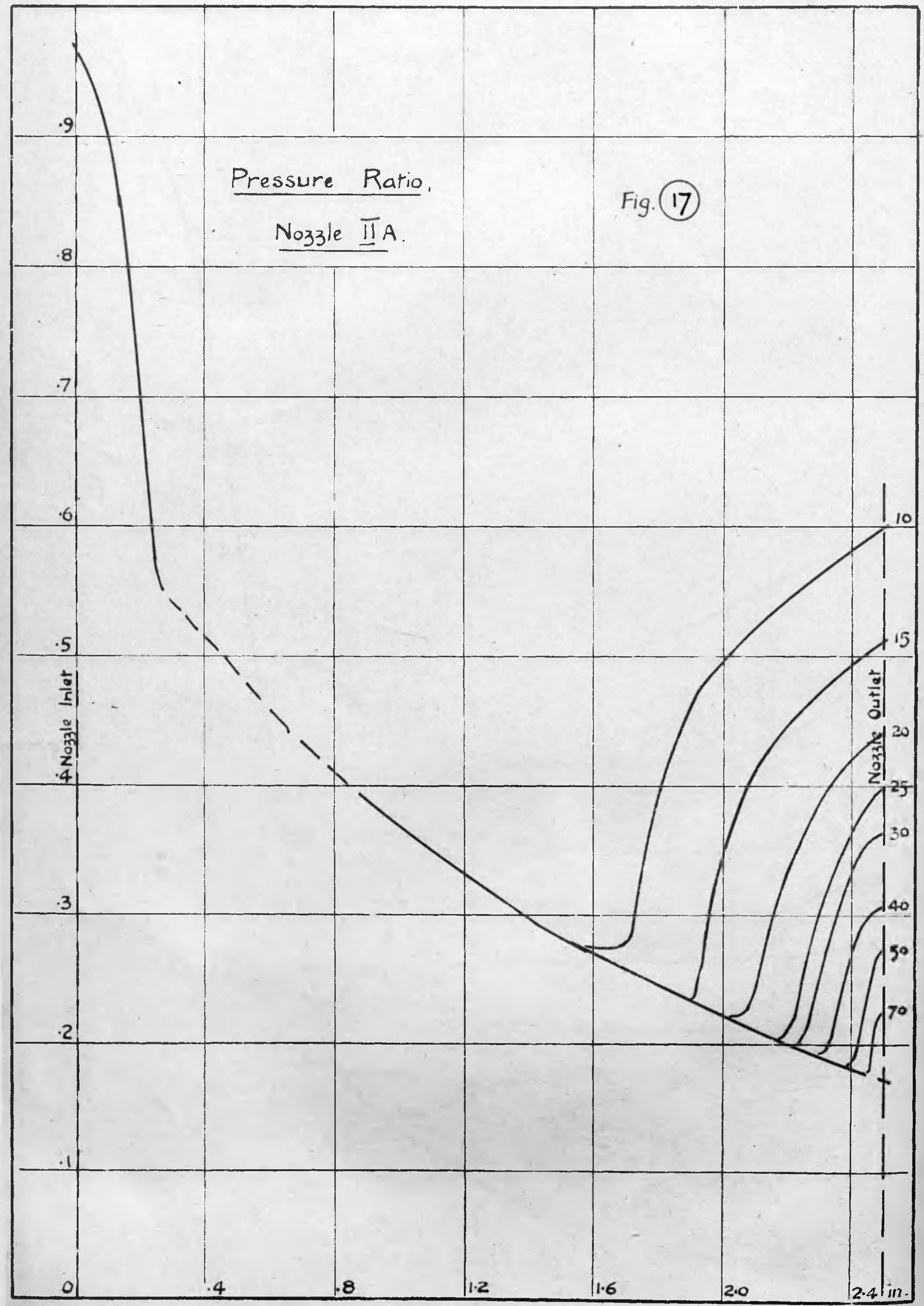
Pressure Ratio,
Nozzle I D.
Fig 14
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·2
·4
·6
·8
1·0
1·2 In.
10
15
20
25
30
50
40
70

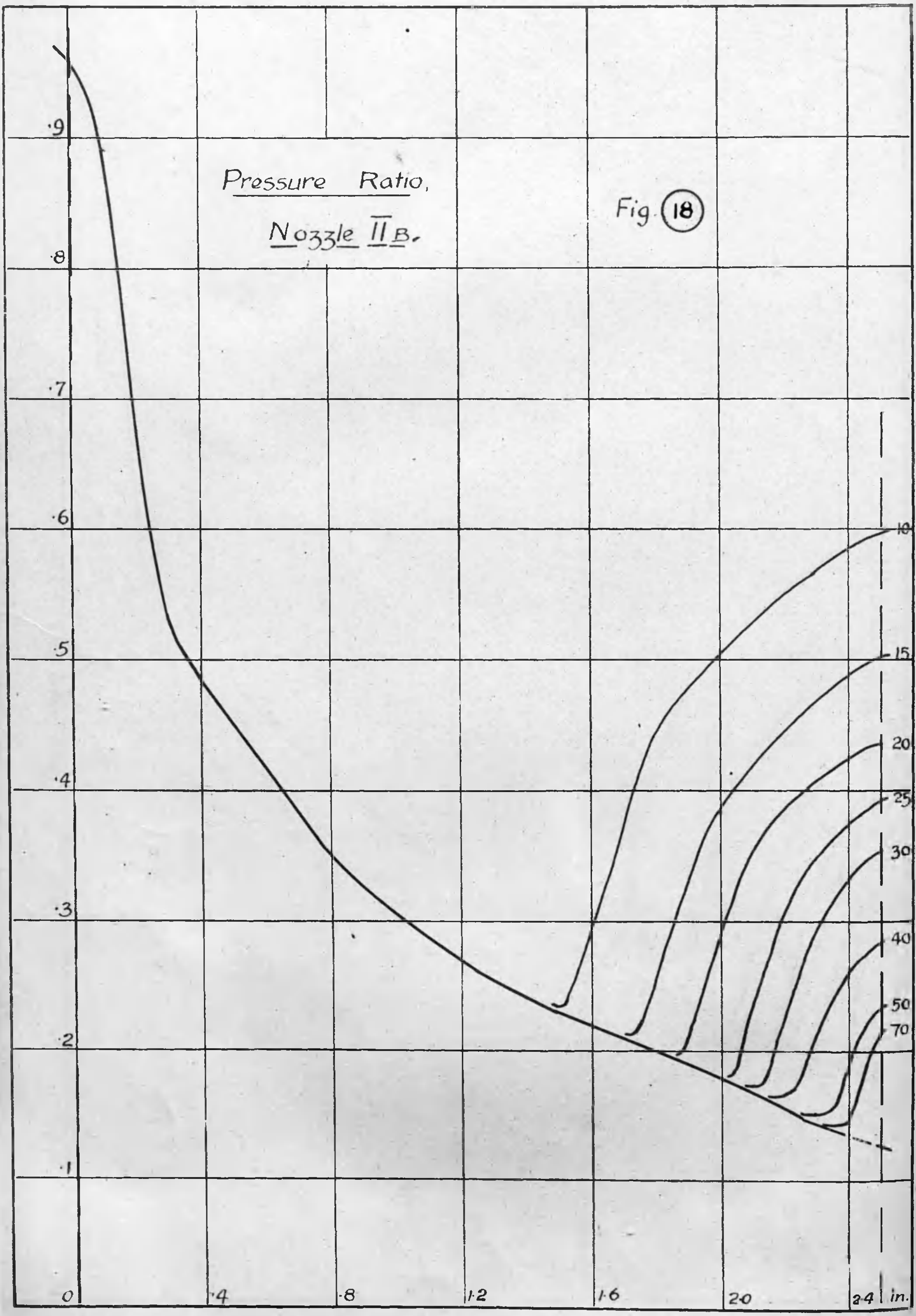


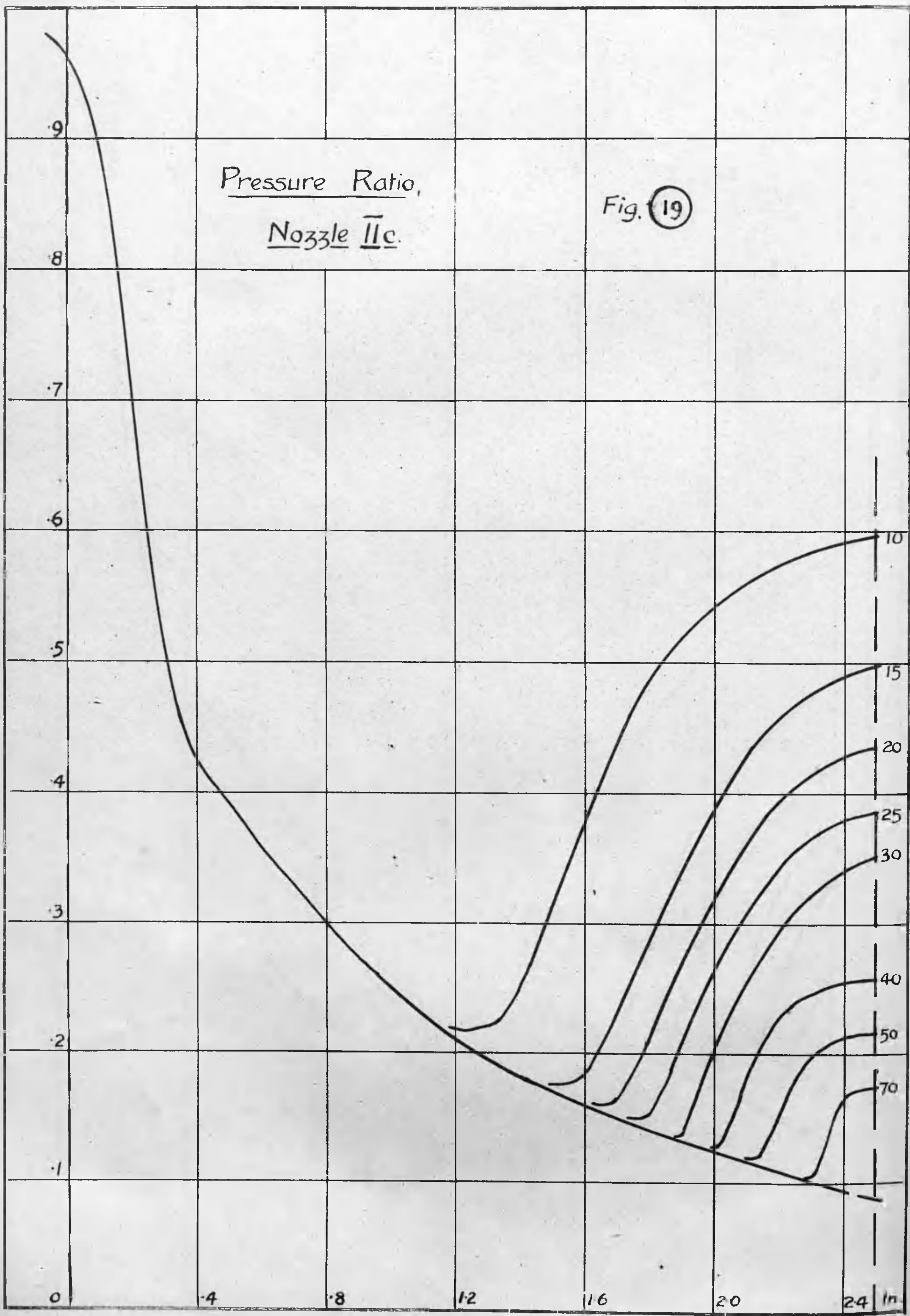
Pressure Ratio,
Nozzle I E
Fig 15
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·2
·4
·6
·8
1·0
1·2 In.
10
15
20
25
30
70
40
50

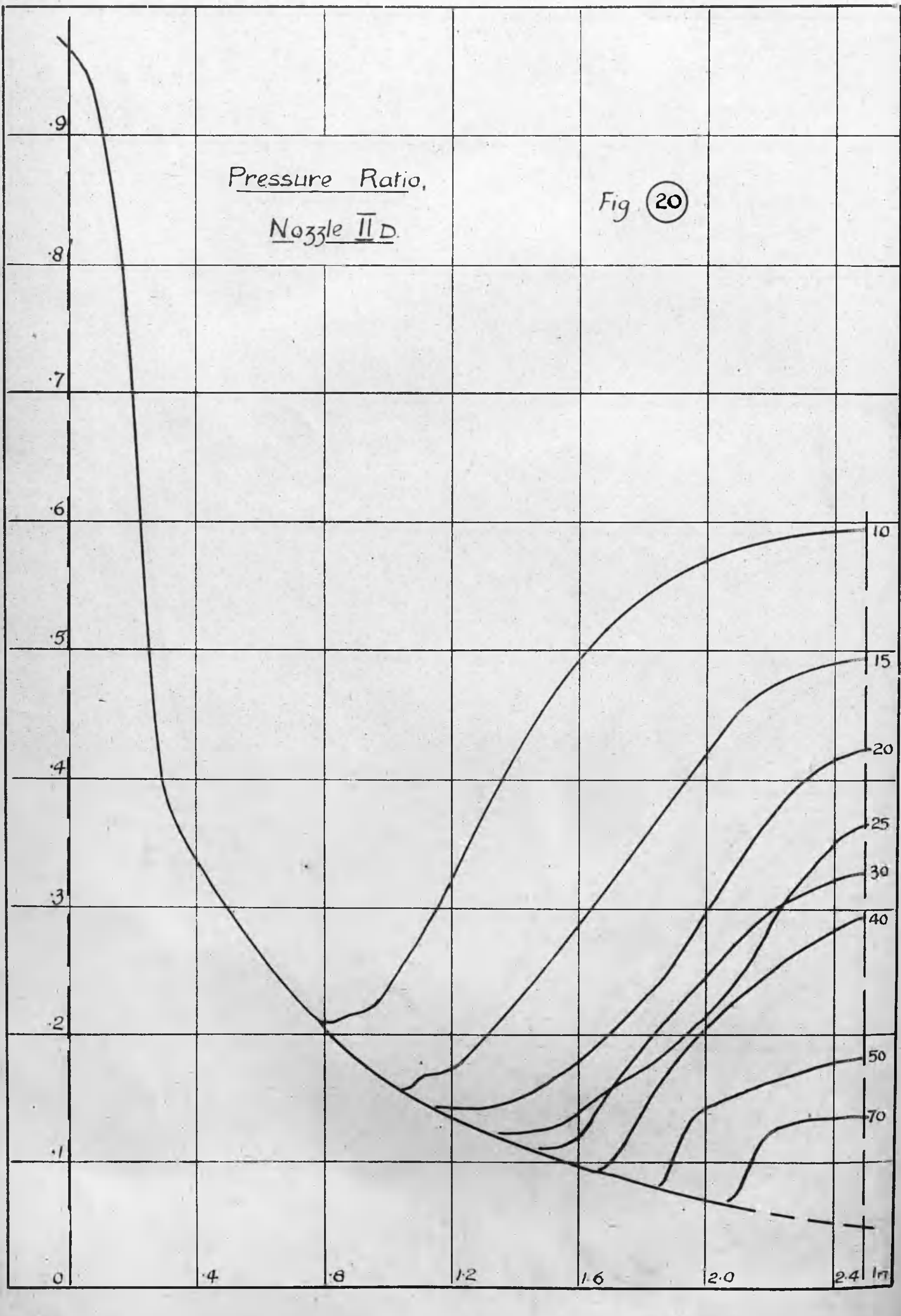


Pressure Ratio.
Nozzle I F.
Fig. 16
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·2
·4
·6
·8
1·0
1·2 in.
Nozzle Inlet
Nozzle Outlet
10
15
20
25
30
40
50
70

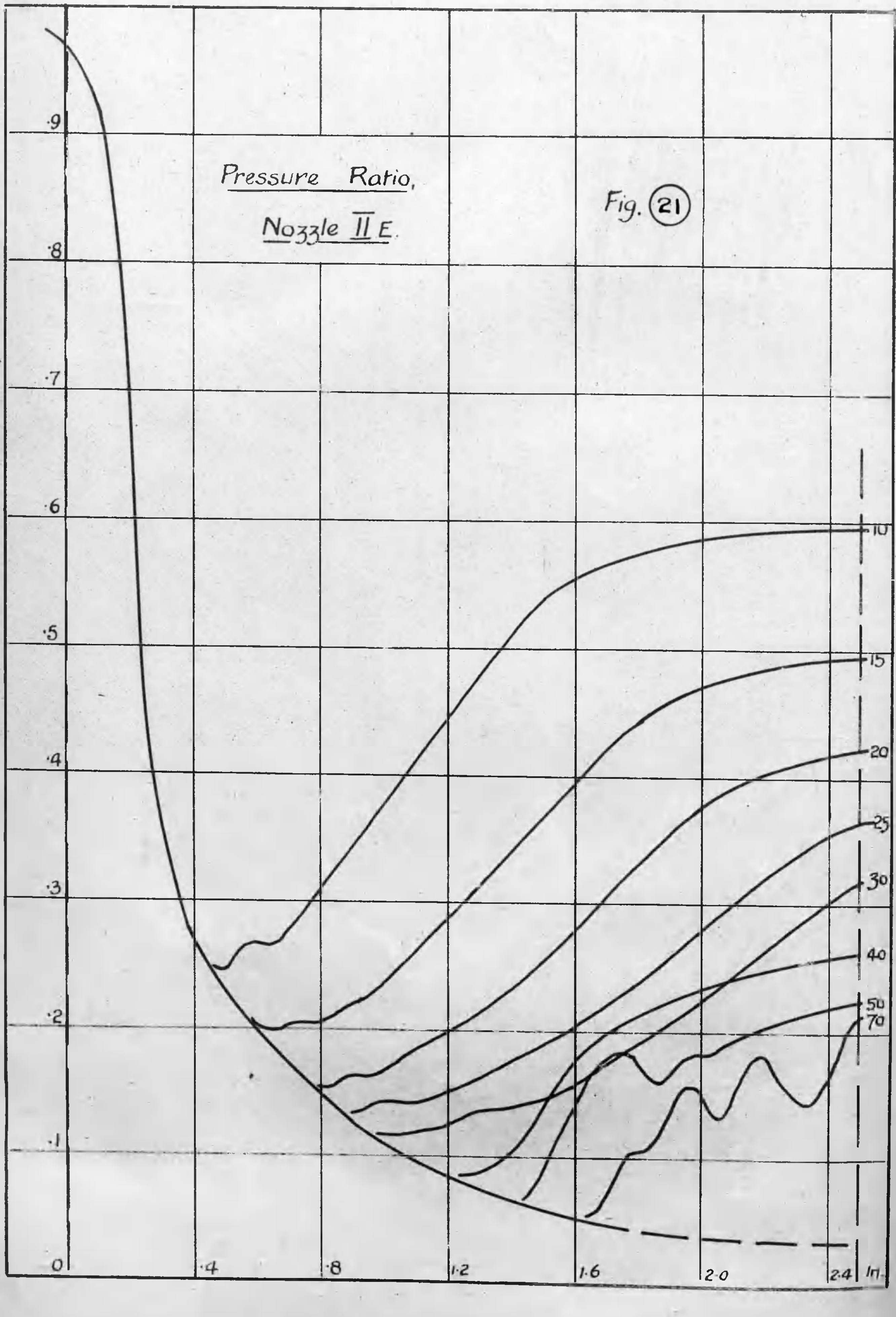

Pressure Ratio.
Nozzle II A.
Fig. 17
Nozzle Inlet
Nozzle Outlet
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·4
·8
1·2
1·6
2·0
2·4 in.
10
15
20
25
30
40
50
70


Pressure Ratio,
Nozzle II B.
Fig. 18
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·4
·8
1·2
1·6
2·0
2·4
in.
10
15
20
25
30
40
50
70

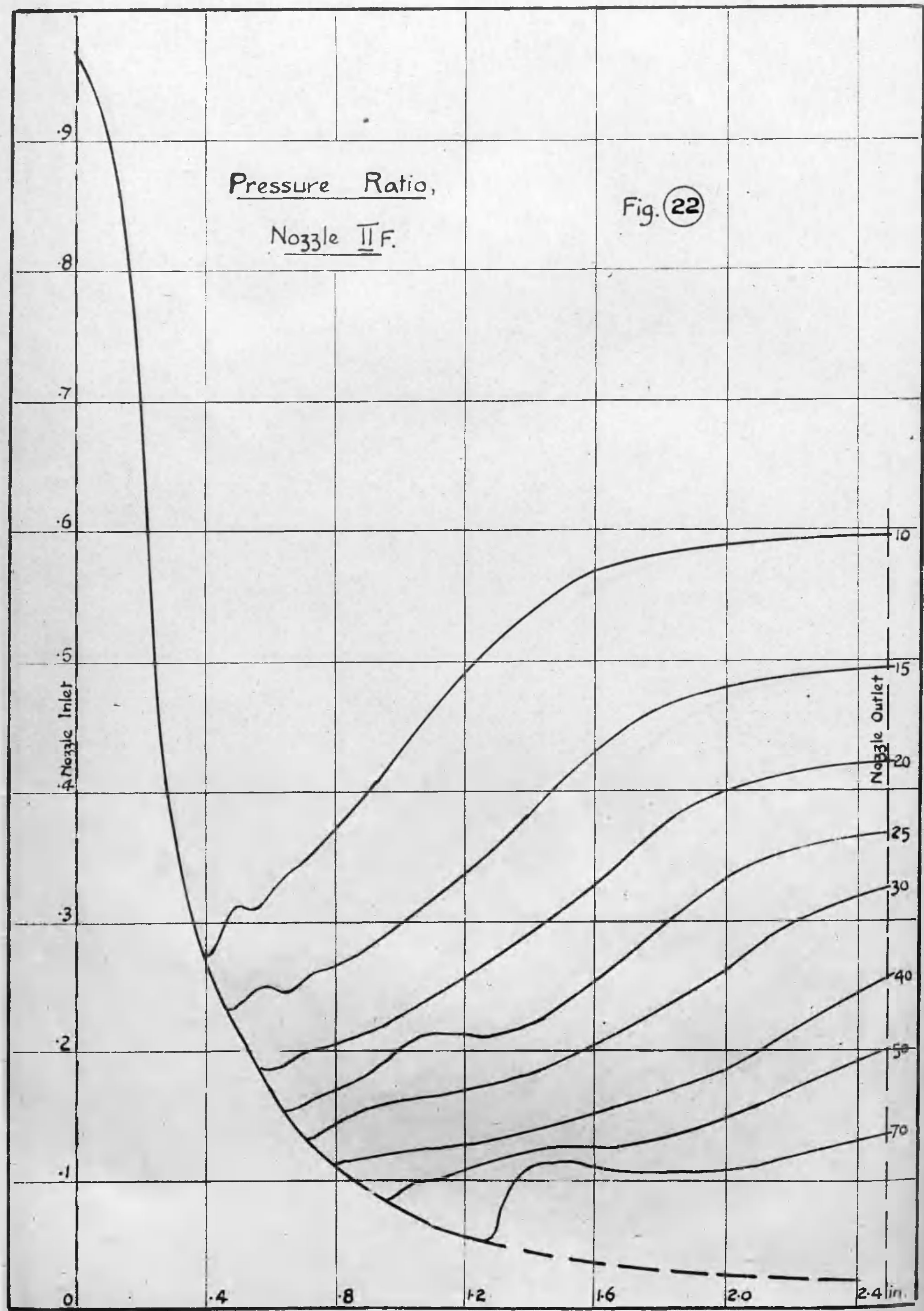

Pressure Ratio,
Nozzle IIc.
Fig. 19
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·4
·8
1·2
1·6
2·0
2·4
In.
10
15
20
25
30
40
50
70



Pressure Ratio,
Nozzle II D.
Fig 20
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·4
·8
1·2
1·6
2·0
2·4
In
10
15
20
25
30
40
50
70



Pressure Ratio,
Nozzle II E.
Fig. 21
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·4
·8
1·2
1·6
2·0
2·4
in.
10
15
20
25
30
40
50
70



Pressure Ratio,
Nozzle II F.
Fig. 22
·9
·8
·7
·6
·5
·4 Nozzle Inlet
·3
·2
·1
0
·4
·8
1·2
1·6
2·0
2·4 in.
Nozzle Outlet
10
15
20
25
30
40
50
70

time and the velocity gained by the jet. Provided the pressure ratio is below the critical (0·528 for air), the flow in lb. per sec. is given by the usual formula,

$$M = \frac{\cdot 534\, P_1 A}{\sqrt{T_1}},$$

where P_1 and T_1 represent the initial conditions of absolute pressure and temperature in lb. per sq. in. and $^{\circ}F$ respectively, and A the throat area in sq. in. The nozzle inlet velocity is negligible, so that the outlet velocity is

$$V_2 = \sqrt{11980\, T_1 \left(1 - r_2^{\frac{n-1}{n}}\right)}$$

where r_2 is the outlet pressure ratio, and n the adiabatic index (1·4 for air). Hence the reaction in lb.

$$R_T = 1{\cdot}82\, P_1 A \sqrt{\left(1 - r_2^{0{\cdot}286}\right)}$$

The theoretical reaction curve is plotted alongside the experimental results, and the total nozzle losses can now be determined. (see Table II.) Taking R_A as the actual reaction and c_d as the discharge coefficient of the nozzle, then the velocity coefficient

$$c_v = \frac{R_A}{c_d R_T}$$

It is found more convenient to express the results in terms of the total loss factor "k_t". This factor represents the energy loss in the same proportion as $1 - r^{\frac{n-1}{n}}$ does the adiabatic energy,[4] so that the nozzle efficiency

$$\eta = c_v^2 = \frac{1 - k_t - r^{\alpha}}{1 - r^{\alpha}},$$

Total Nozzle Losses.

Table II

	P_1 lb/in² abs.	r_2	R_t lb. wt	R_a A	B	C	D	E	F	C_v^2 A	B	C	D	E	F	$\frac{\delta_w}{1}$	$1-K_t-r_2^2$ A	B	C	D	E	F	K_t A	B	C	D	E	F
Series I	24·7	·596	·823	·60	·56	·48	·41	·38	·40	·564	·496	·369	·269	·233	·261	·136	·077	·058	·050	·037	·032	·036	·059	·078	·086	·099	·104	·100
	29·7	·495	1·136	·90	·85	·76	·69	·65	·61	·670	·608	·486	·400	·362	·318	·181	·121	·110	·088	·072	·066	·058	·060	·071	·093	·109	·115	·123
	34·7	·423	1·435	1·22	1·16	1·07	·98	·83	·88	·770	·706	·601	·504	·461	·418	·218	·168	·154	·131	·110	·101	·091	·050	·064	·087	·108	·117	·127
	39·7	·370	1·770	1·53	1·48	1·38	1·28	1·22	1·16	·793	·751	·660	·568	·521	·477	·249	·197	·187	·164	·141	·130	·119	·052	·062	·085	·108	·119	·130
	44·7	·329	2·100	1·87	1·81	1·71	1·60	1·50	1·44	·840	·795	·716	·627	·557	·523	·272	·228	·216	·195	·170	·151	·142	·044	·056	·077	·10[illegible]	·121	·130
	54·7	·269	2·75	2·53	2·49	2·38	2·25	2·12	2·04	·899	·875	·814	·728	·655	·612	·313	·281	·274	·255	·228	·205	·192	·032	·039	·058	·085	·108	·121
	64·7	·227	3·42	3·23	3·19	3·08	2·95	2·79	2·71	·947	·933	·881	·809	·728	·696	·346	·327	·323	·3[illegible]	·280	·252	·241	·019	·023	·041	·066	·094	·105
	84·7	·174	4·81	4·58	4·58	4·49	4·37	4·24	4·07	·961	·975	·945	·895	·853	·792	·393	·378	·385	·371	·352	·335	·311	·015	·010	·022	·041	·058	·082
Series II	24·7	·596	·846	·55	·45	·36	·28	·20	·16	·447	·301	·192	·117	·061	·039	·136	·061	·041	·026	·016	·008	·005	·075	·095	·110	·120	·128	·131
	29·7	·495	1·169	·84	·75	·60	·41	·31	·23	·544	·438	·280	·132	·076	·042	·181	·099	·079	·051	·024	·014	·008	·082	·102	·130	·157	·167	·173
	34·7	·423	1·497	1·15	1·07	·93	·62	·44	·33	·620	·542	·411	·184	·094	·053	·218	·135	·118	·090	·040	·020	·012	·083	·100	·128	·178	·198	·206
	39·7	·370	1·820	1·48	1·40	1·27	·93	·60	·47	·695	·629	·517	·280	·118	·073	·249	·173	·157	·129	·070	·029	·018	·076	·092	·12[illegible]	·179	·220	·231
	44·7	·329	2·16	1·83	1·73	1·60	1·33	·82	·62	·755	·681	·583	·407	·156	·090	·272	·205	·185	·159	·111	·042	·024	·067	·087	·113	·161	·232	·248
	54·7	·269	2·83	2·52	2·44	2·29	2·05	1·42	1·04	·834	·790	·696	·563	·273	·148	·313	·261	·247	·218	·176	·085	·046	·052	·068	·095	·137	·228	·267
	64·7	·227	3·52	3·17	3·14	3·00	2·75	2·12	1·62	·855	·845	·774	·657	·393	·232	·346	·295	·292	·268	·227	·136	·080	·051	·054	·078	·119	·210	·266
	84·7	·174	4·95	4·48	4·54	4·40	4·14	3·48	2·97	·862	·894	·839	·751	·536	·394	·393	·339	·351	·330	·295	·211	·155	·054	·042	·063	·098	·182	·238

where $\alpha = \frac{n-1}{n}$ and thus k_t is derived. This total loss is plotted on a base of r_2 in Figs. 23 and 24, and some of the efficiency curves are shown in Fig. 10.

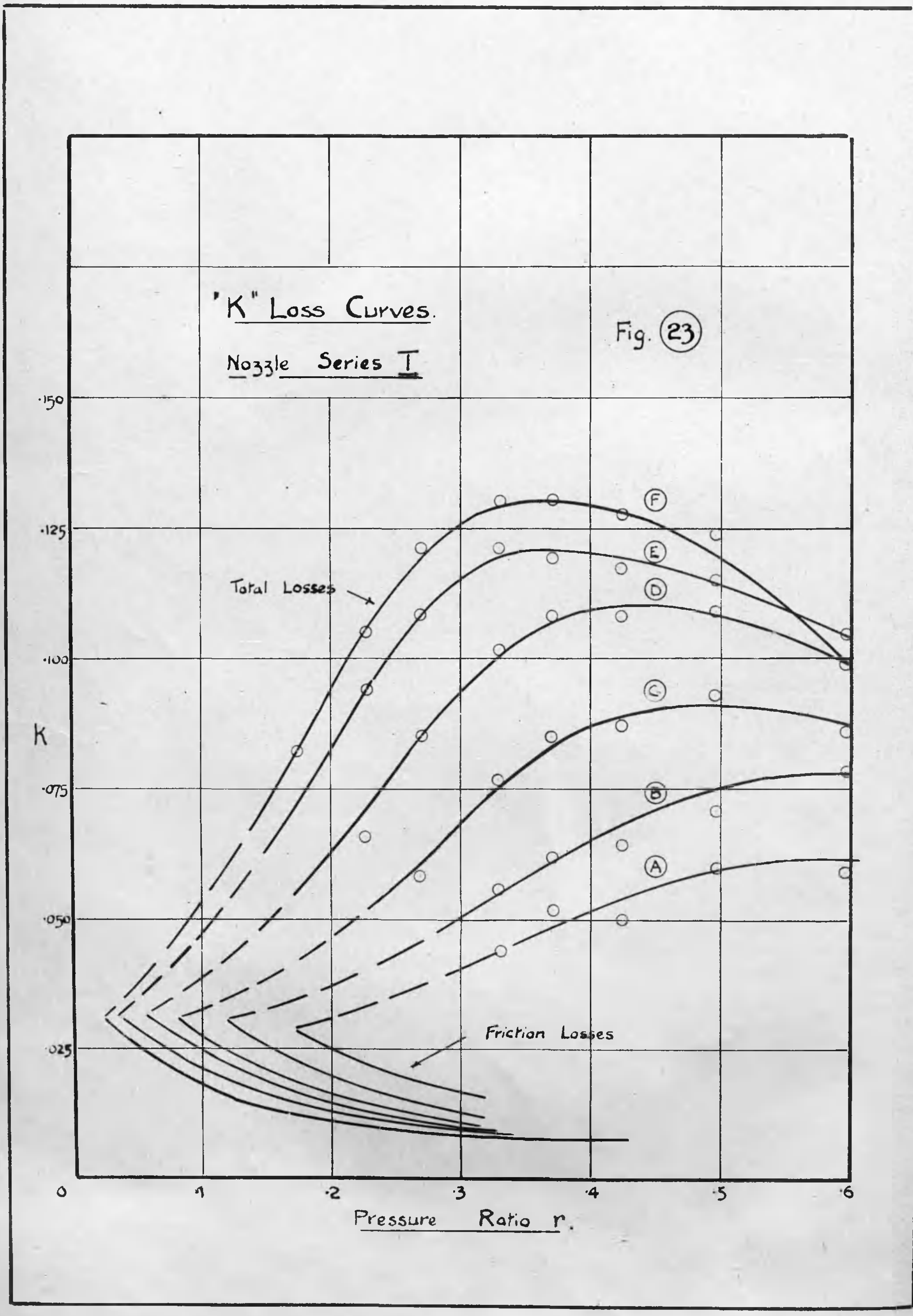
The surface friction of the nozzle is now taken into account, it being also expressed as a "k" loss. The general equation is

$$K_s = c\int_{l_1}^{l_c} \frac{P}{A}\left(1 - K_t - r^{\alpha}\right) dx ,$$

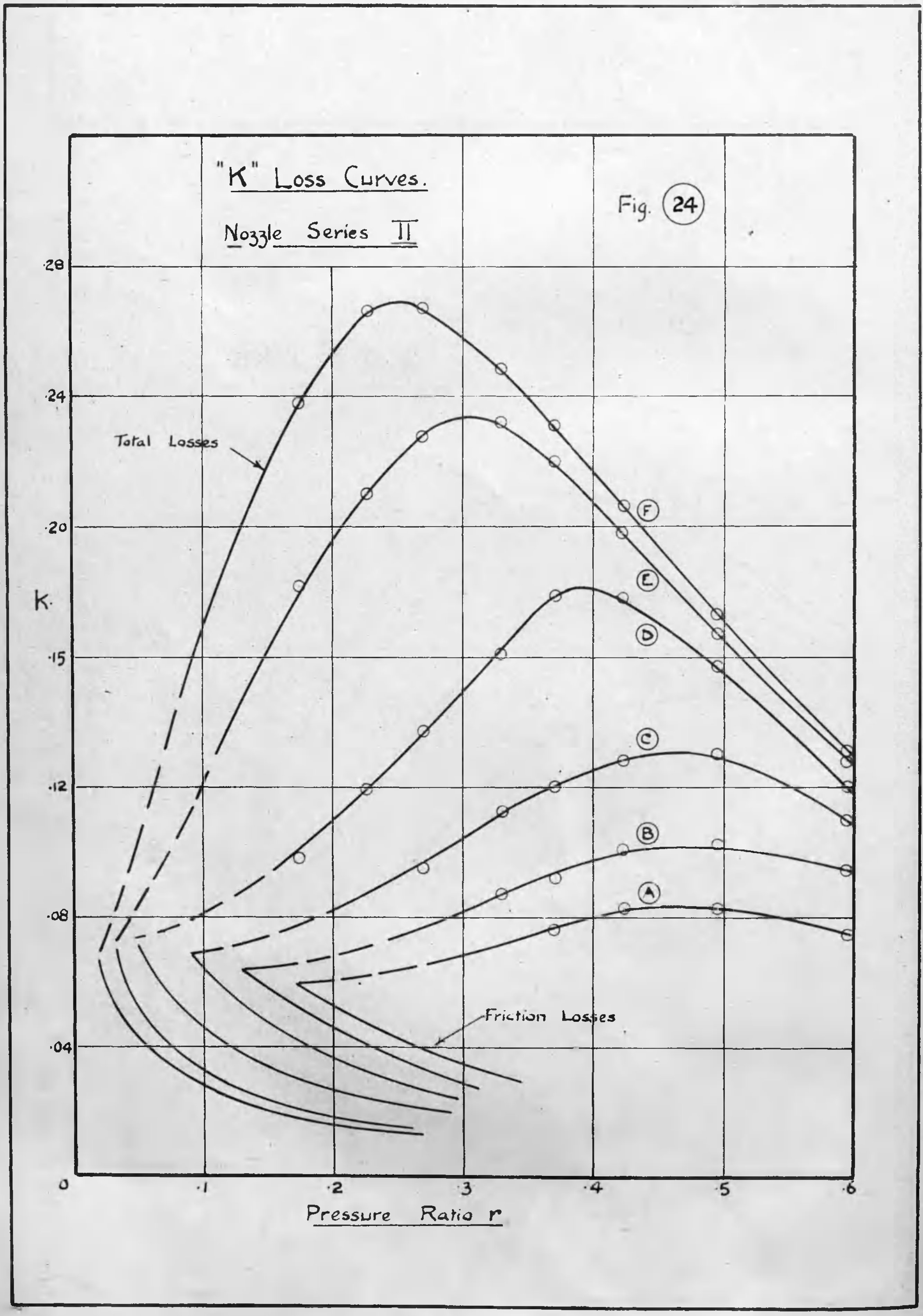
$\frac{A}{P}$ where $\cancel{P/A}$ is the hydraulic mean depth at a point, and dx an element of length. The value of C for a machined nozzle as used in the tests may be taken as 0·005. The curve of $P/A\left(1 - K_t - r^{\alpha}\right)$ is integrated from the nozzle entrance (l_1) up to the point of recompression (l_c), beyond which the nozzle surface has no effect on the jet. In addition to the integration, a small factor, estimated at 0·004 for each nozzle, is added to include entrance losses.

These losses are also plotted in Figs. 23 and 24 on a base of r_c (the ratio of recompression), and since the pressure ratio curves give the relation between r_c and r_2 , it is an easy matter to deduce the compression loss k_c .

LOSS THEORY.- If the loss is incurred during compression, then it should be possible to account for it by the application of fundamental theory to the flow over this stage; the inclusion of any other loss, such as might be caused by some form of shock at the point of recompression, would make it impossible to find such a



"K" Loss Curves.
Nozzle Series I
Fig. 23
Total Losses
Friction Losses
F
E
D
C
B
A
·150
·125
·100
·075
·050
·025
K
0
·1
·2
·3
·4
·5
·6
Pressure Ratio r.



"K" Loss Curves.
Nozzle Series II
Fig. 24
Total Losses
Friction Losses
F
E
D
C
B
A
K.
·28
·24
·20
·15
·12
·08
·04
0
·1
·2
·3
·4
·5
·6
Pressure Ratio r

relation between experiment and theory. In the following analysis, therefore, the loss is considered to be a compression one, so that the relationship, if any, can be found. All the dimensions are given in ft. lb. sec. units.

The resistance to viscous incompressible fluid flow is represented by the dimensional equation

$$R = \frac{\rho v^2}{2g} l^2 \cdot f\left(\frac{\rho v l}{\mu}\right),$$

where ρ = fluid density,

v = fluid velocity,

l = linear dimension,

and μ = viscosity.

The single function $f\left(\frac{\rho v l}{\mu}\right)$ becomes insufficient, if the fluid is compressible, at velocities approaching the acoustic. The resistance is then influ:enced by the formation of waves, and can be expressed by the addition of a compressibility term v/v_c so that the equation becomes

$$R = \frac{\rho v^2}{2g} l^2 \cdot f\left(\frac{\rho v l}{\mu}, \frac{v}{v_c}\right)$$

v_c being the acoustic velocity. Converting this to energy loss per unit volume of nozzle per unit time,

$$E = \frac{\rho v^3}{2gl} \cdot f\left(\frac{\rho v l}{\mu}, \frac{v}{v_c}\right).$$

Now, if M is the mass flow, A the area, and V the specific volume

$$M = \frac{Av}{V},$$

so that the energy loss per unit mass in an element of length dx is

$$de = \frac{E}{M} \cdot A \cdot dx = \frac{v^2}{2gl} \cdot f\left(\frac{\rho v l}{\mu}, \frac{v}{v_c}\right) . dx .$$

Also,

$$de = \frac{v^2}{2g} \cdot \frac{dk}{1 - K_t - r^\alpha} ,$$

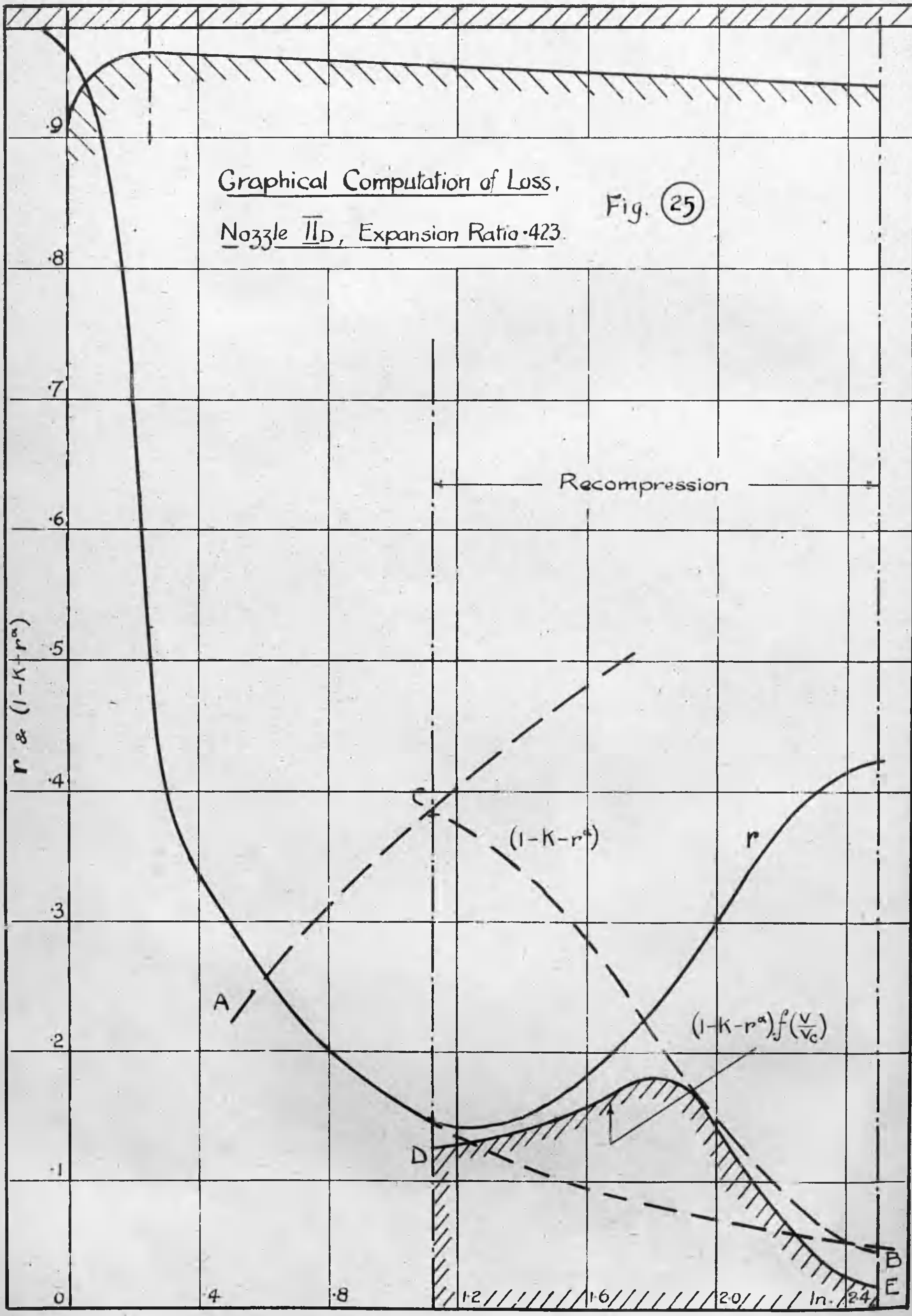
$$\therefore \; dK = \frac{b}{l}(1 - K_t - r^\alpha) . f\left(\frac{\rho v l}{\mu}, \frac{v}{v_c}\right) dx .$$

In a nozzle running full the compressibility term is considered to have little effect, but, as in the tests, when the jet is free and the velocity any: :where in the region of the acoustic, it becomes the predominating factor. Under such conditions, the viscosity term , which gradually becomes less with increase of velocity, is of little importance, compara: :tively speaking, and may be left out of the discussion. Hence the equation reduces to

$$dK = \frac{b}{l}(1 - K_t - r^\alpha) \cdot f\left(\frac{v}{v_c}\right) dx .$$

This loss is to be considered over the free part of the jet, i.e., from the point of recompression onwards, so that

$$K_c = \frac{b}{l} \int_{l_c}^{l_2} (1 - K_t - r^\alpha) . f\left(\frac{v}{v_c}\right) dx ,$$



Graphical Computation of Loss,
Nozzle IID, Expansion Ratio ·423
Fig. 25
Recompression
r & (1-K-r^a)
(1-K-r^a)
r
(1-K-r^a)f(V/Vc)
A
B
C
D
E
·9
·8
·7
·6
·5
·4
·3
·2
·1
0
·4
·8
1·2
1·6
2·0
2·4
In.

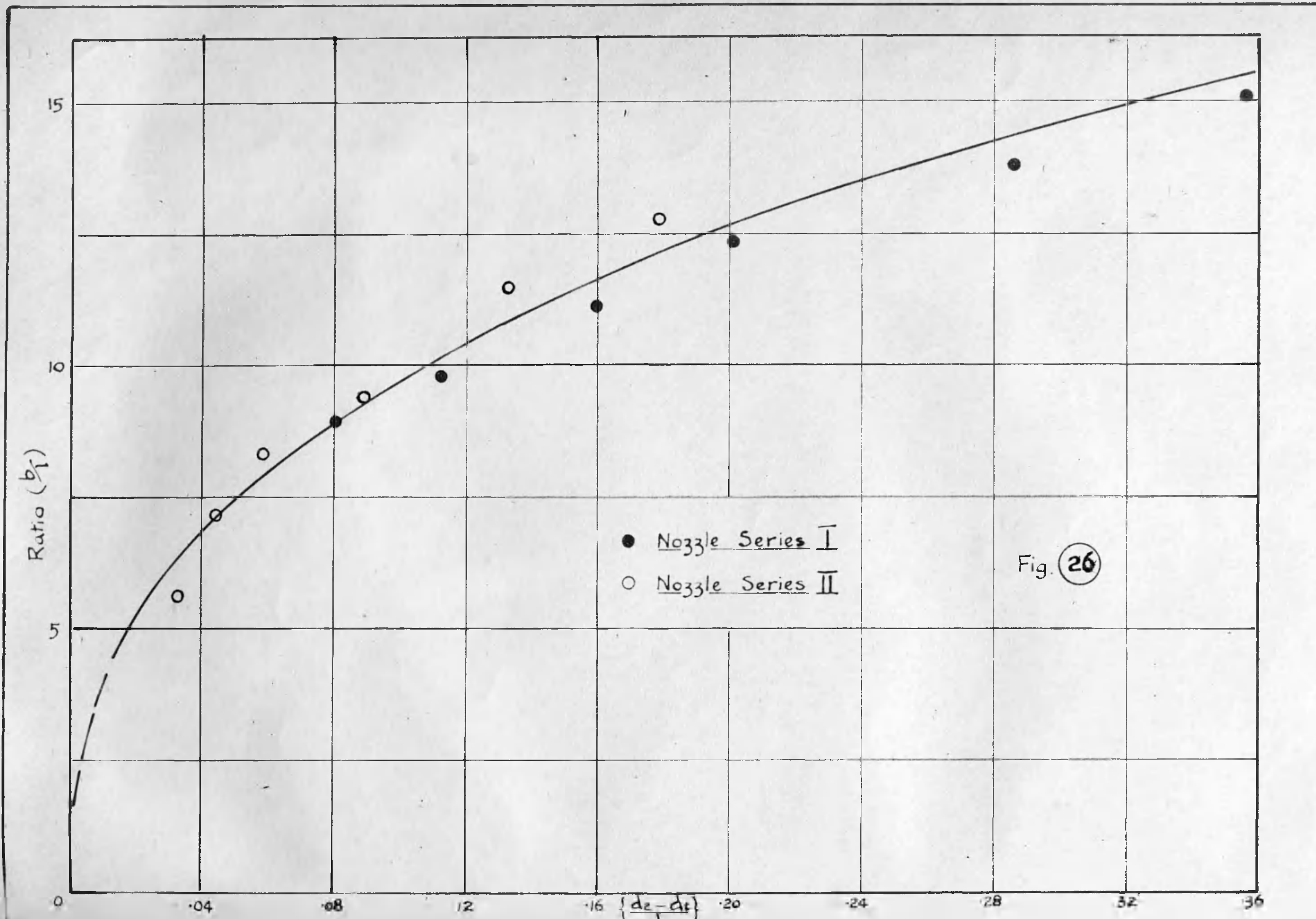
where $l_2 - l_c$ represents the recompression range of the nozzle.

Having deduced the equation from theory thus far, it remains now to evaluate the constant b, the term of dimensions $1/l$, and a suitable function of v/V_c. Considering one particular nozzle,

$$K_c \propto \int_{l_c}^{l_2} (1 - K_t - r^{\alpha}) \cdot f\left(\frac{v}{V_c}\right) dx,$$

since b/l is constant. The function $\frac{v}{V_c}$ is deduced from this relation by graphical construction as in Fig. 25. A curve CB of $(1-K_t-r^{\alpha})$ over the recompression range for each pressure ratio is set down. The k_t value at the commencement of recompression is the entrance and friction loss up to that point, and that at the nozzle outlet may be taken as the total nozzle loss recorded from the experimental data, while values for two or more intermediate points have to be assumed for the moment.

The next step is to choose a function of v/V_c which will modify the areas under the $(1-K_t-r^{\alpha})$ curves to correspond with the loss, as required by the above relation. Some guidance as to the form of the function is obtained from previous investigations on fluid flow, which have roughly established it as rising from low velocities up to a maximum at the acoustic. The estimated function is then used to form the curve DE of $(1-K_t-r^{\alpha}) \cdot f\left(\frac{v}{V_c}\right)$, and since the area under the curve is proportional to the loss, k_t can now be deter::mined more accurately for intermediate points. The $(1-K_t-r^{\alpha})$ curves are corrected for the new k values, and the v/V_c function is finally established. The equation



Ratio (b/l)
15
10
5
0
·04
·08
·12
·16
·20
·24
·28
·32
·36
{(de − dt)/l}
Nozzle Series I
Nozzle Series II

Fig. 26

$$f\left(\frac{v}{v_c}\right) = \frac{0 \cdot 8}{7 \cdot 9\left(\frac{v}{v_c}-1\right)^2+1} + 0 \cdot 2$$

satisfies the conditions with sufficient accuracy, so that the loss equation is now in the form

$$k_c = \frac{b}{l}\int_{l_c}^{l_2}\left(1-K_t-r^{\alpha}\right)\left(\frac{0 \cdot 8}{7 \cdot 9\left(\frac{v}{v_c}-1\right)^2+1} + 0 \cdot 2\right)dx.$$

A plot of k_c against the integrand of this equation gives a ratio $\left(\frac{b}{l}\right)$ for each nozzle, from which the effect of nozzle dimensions can be deduced. The only variables in the tests are the nozzle length, the taper, and the area ratio, and as the length is included in a factor in the integrand, the choice lies with the last two. A plot (Fig. 26) of $\left(\frac{b}{l}\right)$ on a base of nozzle taper $(d_2-d_t)/l$ where d_2 and d_t are the outlet and throat diameters respectively, and $\underline{l}$ the divergent length, gives a single curve corresponding to the equation

$$\left(\frac{b}{l}\right) = 23 \cdot 2\left(\frac{d_2-d_t}{l}\right)^{0 \cdot 38}$$

It will be observed that this is a dimensionless factor, giving no account of the dimension l/l required, therefore it is concluded that some dimension which remains constant throughout the tests requires represent: :ation. The nozzle diameter provides the necessary solution, and it may be assumed that the characteristic term $\frac{p}{A}$ is applicable here, p and A being the perimeter and area respectively, of the nozzle throat. This brings the loss equation to its final form

$$K_c = 0{\cdot}137\left(\frac{d_2-d_t}{l}\right)^{0\cdot38}\frac{p}{A}\int_{l_c}^{l_2}(1-K_t-r^{\alpha})\left\{\frac{0{\cdot}8}{7{\cdot}9(\frac{v}{V_c}-1)^2+1}+0{\cdot}2\right\}dx,$$

or, expressed as an efficiency

$$\eta = 1-\varepsilon - \frac{0{\cdot}137\left(\frac{d_2-d_t}{l}\right)^{0\cdot38}\frac{p}{A}\int_{l_c}^{l_2}(1-K_t-r^{\alpha})\left\{\frac{0{\cdot}8}{7{\cdot}9(\frac{v}{V_c}-1)^2+1}+0{\cdot}2\right\}dx}{1-r_2^{\alpha}}$$

where ε is the fractional loss due to friction, etc.

In such a fundamental investigation, it has been necessary to make a few assumptions. For instance, it may be argued that the pressure recorded by the search tube in the centre of the stream does not represent the mean pressure over the section, but any possible effect on the results would be slight. Of more importance is the fixing of the limits of integrat::ion in applying the theory to determine the loss. The point of recompression gives no trouble; a search-tube exploration over the section shows that the change takes place over the whole section very nearly at once. Since by theory the measurement of the jet reaction gives its momentum on reaching atmospheric pressure, the other limit in the integration should be at the point where this pressure is reached by the jet. This requirement is fulfilled with sufficient accuracy by taking the nozzle outlet section as limit, for in most cases atmospheric pressure is reached at this point, or, as search tube examination shows, very soon after it.

Though the equation applies to the wide range of nozzles tested with sufficient accuracy for all practical purposes, and, in addition, gives some idea of the nature of the loss, a more simple type would be necessary for a practical application. Attempts made in this direction did not lead to any satisfactory result, however, and it may be concluded that the resistance to fluid flow at these velocities is so involved that a simple working formula to determine the losses in a nozzle working under given conditions could hardly be obtained.

It has already been stated that air was used as the working fluid in the tests for experimental reasons, but this does not necessarily limit the application of the equation to air flow. Since the theory is based on fundamental principles, the losses incurred in steam flow could be similarly described.

<u>THE RESISTANCE FUNCTION</u>.- In pipe flow, where the fluid velocities are generally less than half that of sound, the resistance depends entirely on the well-known "Reynolds", or viscosity function. Since this function is comparatively small and decreases steadily as the velocity increases towards the acoustic, and the $(1 - K_t - r^{\alpha})$ factor in the loss function tends to zero as the velocity decreases, the effect of viscosity may be disregarded over the velocity range of the compressing jet.

In contrast with the viscosity function, little or nothing appears to be known about the compressibility function, in spite of the widespread application of the theory of fluid flow in recent years. There is some evidence to show that it rises slowly at first, then rapidly up to the acoustic velocity, but even this has not been definitely established. Recently research has been commenced on ballistic problems, where the velocities are of the same order as in nozzle flow, but little information can be gained beyond the existence of a critical change in the resistance of projectiles at the acoustic velocity. Tests on various types of bullets all gave resistance curves which rose to a maximum at the acoustic velocity and fell away again slowly. The function employed in the loss equation shows the same general features.

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