A STUDY OF PRIMING PROCESSES IN A CENTRIFUGAL PUMP

M.Sc. Thesis

by

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Summary

A search through literature reveals that there is no method for converting a standard centrifugal pump with a shrouded impeller, to self-priming operation without the use of valves or large loss in efficiency.

A review of work carried out by Dr. Ing. A. Welte in Germany shows that, in the case of open shrouded impellers the pump operation is subject to instability in early stages of priming, if the re-fluxing jet is directed against the direction of rotation of the impeller. The conclusions of this work are, that the maximum attainable vacuum is a direct function of the impeller speed alone, ar the air pumping capacity depends not only on the impeller width but also on the method of introducing the re-circulating fluid. Furthermore the turbulent nature of flow renders it unsuitable for theoretical treatment.

The investigation described in the thesis is carrie out in two stages.

The first, a feasibility study, is concerned with the alternative methods for energising a mixture of air and water by an impeller. It is found that the best method is by directing a jet at right angles against the suction end shroud of the impeller. An investigation of the jet positioning and casing geometry leads to the conclusion that the jet must be placed 180° from the casing tongue and a baffle should be inserted into the discharge branch to prevent the direct re-circulation of air bubble bearing mixture. It has also been found that the quantity of air entrained is almost directly proportional to the volume of re-circulating water and that the rise in water temperature during the priming time is not excessive. The reduction in pump efficiency resulting from the adoption of continuous re-circulation is of the order of 3½%.

The second stage, a development and detail study, deals with subsequent improvements of the jet arrangement dispensing with the baffle.

The priming process is found to consist of two basic stages. The first is due to jet action alone, with air entrainment taking place in the space between the impeller shroud and the pump end cover. The air suction capacity decreases sharply as the vacuum rises and the cavity fills completely with water. The second stage commences when the water in the volute casing is drawn sufficiently close to the impeller tip by the vacuum, to cause a formation of a secondary cavity at the casing tongue. The formation of this cavity results in a sharp initial increase in air pumping capacity, which gradually falls off with rising vacuum, as the impeller periphery becomes increasingly water logged. Velocity traverses carried in the volute at various vacua confirm this theory and explain to some extent the instability during the change over from one stage to the other.

The final results show that the suggested priming system is suitable for adoption with standard shrouded pump impellers and that it compares favourably with all existing units, except the "Hannibal" in respect of water pumping performance, while the priming performance is equally good.

In view of A. Welte's conclusion, as well as actual flow observations, no attempt is made to produce a rigorous mathematical treatment.

A CENTRIFUGAL PUMP. "

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

The inspiration for this treatise came from the tireless effor of Prof. Wilhelm Schulz and Dr. Ing. A. Welte, both of the Technische Hochschule of Hannover, who contributed materially to our knowledge of principles involved in the construction of self-priming centrifugal pump

Prof. A. S. T. Thomson granted the experimental facilities in Department of Mechanical Engineering of the University of Strathclyde, and Dr.H. L. McBroom counselled me on many occasions throughout the tortuous path of experimentation.

Much of the equipment and physical help, in the initial stages was given through the courtesy of C. H. Carslaw, B.Sc., M.I.Mech.E., and I. Paterson, B.Sc., A.M.I.Mech.E., who are Technical Director and Chief Research Engineer respectively, of Messrs. Drysdale & Co. Ltd., of Yoker, Glasgow.

Last but not least, I should like to acknowledge the infinite patience and skill of the Technicians of Fluid Mechanics Laboratories in construction and operation of the experimental equipment.

The speedy conclusion of my efforts is a result of the co-operation of all those directly concerned inside and outwith the University, and in this respect I may only add that their helpfulness would not be easily bettered.

P R E F A C E

The outcome of any development research has as its ultimate aim not only the provision of a satisfactory answer to a problem in hand, but also an understanding of the scientific principles involved in producing such a solution. Unfortunately the latter aim is frequently more difficult to accomplish than the former, and one has to be content with a machine working for a number of reasons, only a few of which can be ascertained by scientific measurements. The remainde are a matter of conjecture and inspired guesswork. The investigation of the self-priming process in a centrifugal pump was undertaken by the writer with a view of producing an efficient adaptation of a standa end suction centrifugal pump to self-priming conditions, without use of valves or changes in the impeller and volute design. It was hoped to produce a design based on a deeper understanding of the physical phenomena taking place in the pump during priming, the knowledge of which is not available, even on an empirical basis.

At the time of writing this simple account, one has to admit defeat on the score of a complete understanding of the process of self-priming, in spite of remarkable success on the practical side. It is appreciated that this is far from satisfactory, but, barring extremely expensive new equipment in the form of a transparent pump and impeller, there is little hope of throwing more light on the subject. Even then the prospects of producing a rigorous mathematical treatment are extremely remote, and one would tend rather to proceed more with trial and error variants of geometry, than any other approach.

In view of these prospects, the attempts described in this treatise should be viewed not as the most sophisticated possible, but the most expedient under the circumstances. It should be added that the final form of the pump is now both, subject to a patent, and being seriously considered for production.

The experimentation was carried out principally at the Hydraulic Machinery Laboratory of the Royal College of Science and Technology, which has now become the University of Strathclyde, during the period 1963 - 1965.

S U M M A R Y

The subject of this dissertation is an investigation and development of what is thought to be a novel method of self-priming of a centrifugal pump. It is not proposed to deal with the fundamentals of the well known established variants of self-priming pump layout except in the way of a review contained in Part I. This will also contain a brief analysis of attempts to put the comparison of priming performance characteristics on some form of rational basis.

Part II contains the description of experimental apparatus and testing procedure employed in the development of the self-priming pump.

Part III consists of experimental findings and their analysis as well as some attempts to correlate the results with those of Dr. Welte's experiments.

A recommendation for the general form of adaptation of a shrouded impeller pump is made, in view of experience to date.

PART I

If one discards the variety of ingenious devices, which can be incorporated into a centrifugal pump to produce self-priming, there are only two basic methods left for consideration.

A. Priming by external recirculation from the pump discharge into the impeller eye. See Fig. I

B. Priming by an internally incorporated passage returning the fluid from the discharge into the pump volute, either against the direction of rotation of the impeller, Fig. 2a , or in the same direction, Fig. 2b .

All pseudo-centrifugal pumps such as Sihi, peripheral circulation pumps and many others, depend on the repetitive recirculation of the liquid mixture through the blading, which renders them unsuitable for incorporating into standard pumps. The scope of the review is thus strictly limited to types A and B. The general difficulty in generating a priming head with a centrifugal machine lies in the fact that the vortex head is a direct function of the density of the working fluid. It thus becomes impossible for a water pump to generate sufficient priming head when operating with air. To overcome this contingency one has to resort to the use of an air-water mixture, formed by a small quantity of liquid in the pump, to generate enough head and separate the two fluids after passing through the unit. The air is allowed to escape, while the water is re-used to entrain more air in the machine and form a new mixture.

The system A, shown in Fig. I, returns the liquid into the impeller eye, where it is mixed with the gas from the suction line by splash and turbulence, and the mixture is then discharged through the impeller before the air can separate out. The re-fluxing jet energy is a result of the pressure differential between the discharge and impeller inlet, and can represent a considerable loss of pump output if it is not shut off after priming has been completed, when full pump operation commences. The disadvantage of this method lies in the limited size of the nozzle passage and the necessity to use some form of shut-off device in the return passage to maintain reasonable pump efficiency. The mechanism of gas entrainment is very haphazard and is usually arrived at by trial and error. This method, while fairly common, is not recommended for units of reasonable efficiency and dependability.

The system B, Fig. 2a and 2b, depends on internal recirculation in the pump casing, with the impeller tip and shroud providing the mixing impulse and mixture ejecting energy. The mixture is discharged into a separator, whence the gas is exhausted, whereas the liquid is returned back to the impeller tip for further gas entrainment. Once the priming has been completed and the impeller discharges liquid only, the direction of flow in the return passage may reverse and it acts as an additional discharge taper. (This will, of course, only apply to system shown in Fig. 2a).

An outstandingly interesting variant of the type B is a machine invented and patented by Prof. Schulz under the name of "Hannibal System" shown in Fig. 3 . Here the casing has been fitted with an insert which, together with the impeller periphery, forms a venturi shaped nozzle. The purpose of the insert is to direct some of the fluid dragged along by the impeller, in the form of a purging jet, into the impeller passages. This results in a vigorous air mixture formation, which is then extracted from the impeller by velocity recovery in the diverging part of the insert, and directed into the discharge taper, whence the usual separation takes place. Pumps of this type have been known to have a water pumping efficiency of 75% , while retaining their self priming characteristic, which is a most remarkable achievement. On the whole, however, units employing method B require much larger casings and special impellers, usually of the open-shrouded type and with 3 to 4 blades only, in order to permit proper mixture generation. Thus multiblade, shrouded, narrow, high efficiency impellers are not suitable for method B, which is the only valveless variant of the normal centrifugal pump.

An extremely thorough investigation into the priming process of the B type unit was carried out by Dr. Ing. A. Welte at the Technische Hochscule of Hannover. This work is the only systematic on record, and since the findings are of general significance, it is proposed to review Welte's results in some detail. In order to form a general picture of performance characteristics, three standard pumps were tested under a variety of conditions. Fig. 4, 5 & 6 show the pump details, while Fig. 7 gives the air evacuation performance curves of these units. An experimental pump with a series of open shrouded impellers shown in Fig. 8 & 9 was built to enable the experimenter to vary both the casing and impeller geometry, and to obtain detailed pressure distribution and air and water circulation data. Figs. 10, 11 & 12 show a typical set of experimental results obtained during the investigation. It is not intended to reproduce here any more of Welte's results, but the general findings pointed to the following characteristics of a pump with an internal recirculation partition wall.

A. CONTRA FLOW ARRANGEMENT.

The best impeller outlet angles are 60^{o} & 120^{o}, with the performance falling off in-between and reducing towards zero for angles tending to 0^{o} and 180^{o}.

The effect of impeller speed in most cases of impeller blades was that of increase in air pumping capacity with speed, for a range of 1400 2600 r.p.m., with a tendency for the increase to flatten out to either side of the 60^{o} outlet angle. There is a somewhat unexplained tendency for the air pumping capacity to fall at an outlet angle of 120^{o} with increase in speed. Another feature of the air flow/speed diagram Fig. 10

is the peak value obtained at 600 r.p.m. except for the 120^{o} setting.

If we examine a typical constant speed test shown in Fig. II the general pattern of liquid mixture behaviour becomes fairly clear. The pressure of two break-waters produces two low pressure peaks in the impeller passages during a full revolution. This implies radial surging of the free water surface inside the impeller passages or rapid changes in circumferential velocity, both of which will result in turbulence at the free surface and air mixture formation. If this surface is sufficiently close to the impeller tip some portion of the mixture eddy will be ejected upward at the top break-water into the separator.

B. PARALLEL FLOW ARRANGEMENT (FIG. II)

The outstanding difference between this and the contra flow arrangement is the fact that there is only one low pressure peak at the impeller periphery, which occurs at the upper break-water. It is noteworthy that the pressure distribution indicates a much larger cavity extending over a greater length of the casing wall. In fact, the region of sub-atmospheric pressure in the casing extends over almost 180^{o} compared with just about 90^{o} for the contra flow arrangement. This seems to explain to a large degree the reason why the D-Pump has an air pumping capacity of more than twice that of the other units, as can be seen from Fig. 7. The general conclusions formed on the basis of Welte's experiments can be summarised under the following headings :-

1) DEGREE OF PUMP CASING FILLING OR FREE SURFACE POSITION ABOVE THE CENTRE LINE OF THE IMPELLER.

In the case of a typical parallel flow pump the air pumping of capacity appears to be fairly independent ~~from~~ the separator level location above the centre line of the impeller, within the range of 10 in. to 16 in. There is a rise to a peak of about 10% in excess of the flow of these limits. The reasons for this are two-fold. One is the necessity to remo as far as possible the separation surface of bubble mixture from the point of re-entrainment, while the other is the need to produce as large a drivin head as possible for the re-entraining jet. It is worth noting that the latter becomes less important with the drop of pressure in the impeller eye but the former actually increases in importance under these circumstances in view of less favourable conditions for gas bubble separation. These tend to be dragged down into the jet with increasing velocity of re-circulation.

2) EFFECT OF OUTLET ANGLE.

It is quite clearly demonstrated in Fig. 10 that for a usual outlet angle in a conventional pump the peak of the performance is not likely to be reached in the contra flow arrangement, since it appears to take place at 60° and 120°, with a drop between those two values, depending on the speed. No information is available for the parallel flow layout. If we take the maximum permissible water pump outlet angle not to exceed 45°, then it is fair to say that for self-priming performance the angle should be kept as large as possible.

3) IMPELLER TIP SPEED.

This effect can be considered two-fold. On one hand the maximum vacuum is directly a function of the speed alone, as can be seen from Fig. 13. On the other hand the rate of evacuation depends not only on the rate of mixture generation, but also on the size of air bubbles in this mixture. If these happen to be large, the separator will be more effective. If, on the other hand, they are small, inevitably some will be re-entrained in the re-fluxing jet and reduce air pumping capacity. This could explain the peak at between 400 - 600 r.p.m. followed by decrease in air flow, as seen in Fig. 10. Once the mixture composition has become stable, a rise in speed is accompanied by an almost linear increase in the air pumping capacity, the rate of which depends on the impeller and casing geometry. The limit to this capacity is fixed by both suction pressure and tip speed, in view of cavitation and the forced vortex head generated in the impeller.

4) IMPELLER OUTLET AREA EFFECTS.

It is quite obvious that the drag exerted by the impeller on the re-fluxing fluid is a direct function of both speed and the area of the impeller in contact with the fluid. One can, therefore, think of an Air Flow Co-efficient $\varphi \text{ air} = \frac{V\text{air}}{b_o \times D_o \times u_o}$, for use as a parameter to compare the various machines under the same suction conditions, since such number incorporates both the area and velocity supplying energy to the fluid.

i.e. – The greater φ air the better the pump.

To separate the speed effect we can plot the suction pressure against the ratio of $\frac{V_{air}}{b_o \times D_o}$. The resulting curve is shown in Fig. 14. The rectangular area A represents the ideal pump performance, i.e. independent of suction pressure, whereas the area C under the curve represents the actual pump performance at an instant speed. The ratio of these give an overall efficiency of operation over a complete possible range of suction pressures. This latter concept introduced by Welte under the name of "Capacity Co-efficient" is independent of speed, and is a very useful criterion of comparison between various self-priming units, providing the impeller periphery forms the only energising surface. A reasonably valid criticism of this concept is that it has no direct bearing on the actual air pumping capacity, which is of paramount importance in the view of the fact that the priming time is very short anyway, the loss of power being rather immaterial, nor does it take into account the change of air density due to pressure reduction. In general it has been found desirable to maintain a wide impeller, to facilitate the radial penetration of the re-fluxing jet. The actual sizes of the impellers tested by Welte varied from 20 to 25 mm. in width.

5) INSTABILITY IN THE AIR VOLUME/SUCTION PRESSURE CHARCTERISTIC.

The investigation shows clearly that, under certain conditions, depending on the geometry of the pump, there is a period of instability in this characteristic, see Fig. 7, produced by flow phenomena inside the pump. In particular, a steep striking angle of the fluid at the break water or an excessive impeller outlet angle can produce undue turbulence and impact in the case of contra flow or diffuser arrangements, such as in type G and M, whereas the parallel flow of type D is not subject to th peculiarity. It is the rapidly increasing depth of jet penetration into t impeller, which causes the contra flow arrangement to exhibit this particul characteristic by producing a low pressure on the downstream portion of the tongue, in combination with choking effects in the mixture discharge passag into the separator, Fig. 11. The latter is due to the fact that the high velocity of the re-fluxing stream operates against the energising effect of the impeller, instead of being additive to it as in the case of parallel fl unit. The pressure distribution on the downstream portion of the tongue t displays characteristics which are similar to those associated with a heavi throttled centrifugal pump.

PART II

The experimental equipment and techniques were develop under two separate headings.

A. Feasibility study.

B. Development and investigation of detail.

Prior to dealing more closely with these two procedures, it would be of some interest to describe in greater detail the machine itself and the min requirements set out to make the project of practical significance.

The pump, U - II, which is a standard close coupled, end suction unit, with a 4" suction and 3" discharge, operates at 2900 r.p.m. It may be fitted with alternative impellers of $7\frac{1}{4}$" , $6\frac{1}{2}$" , and 6" diameter to extend its range. The impeller passage width is II mm., and the total axial width is 16 mm. The sectional arrangement of the unit is shown in Fig. 15 together with the basic dimensions of the volute and impeller, which has 6 blades and a 22^{0} outlet angle. The N.P.S.H. is of the order of 12 ft. and it was decided that the priming criterion should be, the raising of a water level in a 3" suction line to a height of 10 ft., in a time of 3 - 4 minutes. Maximum efficiency drop on the water pumping duty was not to exceed 5%. The permissible additions to the pump were, a suction sealing bend or cover, air separator and some form of reflux passage into the volute. No changes in the impeller or volute shapes were permissible, nor were the use of any valves.

A. FEASIBILITY STUDY.

On the basis of the available information, discussed in the previous chapter, it was realised that, with the existing impeller, it would be very difficult to produce an eddy inside the impeller passages. Accordingly two ideas were explored. A small jet striking the impeller radially was expected to produce an ejector effect in the throat of the casing or, alternatively, the jet was expected to penetrate sufficiently into the impeller passages to produce adequate air entrainment. The position of the jet was arranged at a number of radial tappings, as shown in Fig. 16 . The suction branch was blanked off by a Perspex flange, and a barrel separator shown in Fig. 17 was fitted to the discharge branch. A suction line through a gas meter was connected to the inlet line, and thus the first experimental layout was arranged, as indicated in Fig. 18. The tests have shown immediately that the scheme has no prospect of succeeding for two reasons. One of these was poor separator efficiency, while the other was a totally inadequate recirculation and energy transfer from the impeller. Accordingly, a rectangular box separator was substituted, Fig. 30. Tests were also conducted with an impeller being rotated in open air, to gain insight into the best method of energising the recirculating jet. These attempts, shown in Fig. 19a, b and c, have suggested that a jet should be directed against the shroud face, thus permitting the use of a larger jet. Accordingly, after some experiments with smaller jets, it was decided to use the maximum possible diameter, which was 1", led in at 90^{o} to the shroud through the suction end cover. This jet was supplied either from the separator or from an external source, while the separator level was maintained constant by an overflow weir. The circumferential position of this jet could be varied by moving the suction cover through a stud pitch, as shown in Fig. 20, a, b, c, which gives the final arrangements of the feasibility study equipment for air evacuation tests.

To ascertain that the recirculation does not impair unduly the water pumping performance, a somewhat different hydraulic circuit, shown in Fig. 20 d, was arranged, with a watt-meter in the motor circuit, to facilitate efficiency calculations. The other equipment employed were manometers, vee-notch and metal baffles, see Fig. 29 , used to improve the performance of the rig, in order to reach the minimum requirements. It may be added that a priming time of 3 min. 25 seconds was eventually reached with a drop in water pumping efficiency of the order of 3% , and it was thus decided to proceed with further development and detail investigation.

B. DEVELOPMENT AND DETAIL INVESTIGATION.

The following description will cover an investigation of the phenomena taking place in the pump itself. No development has been pursued in respect of a commercial air separator design, which is thus retained in the form of a rectangular box for the present. The experience of previous investigators, as well as the feasibility study, have shown that the following features are desirable in a re-fluxing pump investigation.

I. Flow visualisation.

2. Recirculating flow rate measurements, as well as gas flow measurements.

3. Velocity and pressure measurements at the appropriate points in the flow passages.

4. Input power and speed of rotation control.

The limitation of time and cost required to fulfil these, while using a standard pump, led to some degree of compromise on various scores.

In its final form, the rig shown in Fig. 21 is arranged for a water pumping test, and consists of the pump, in this case mounted on a pedestal and driven by a variable speed D.C. torque mounted motor. Fig. 22 shows the same pump arranged for an air pumping test. A Perspex cover with a number of axial holes is used in place of the standard suction cover, but the shape of the internal casing is unaltered, except for a provision to reduce the shroud end clearance by inserting a spacer disc, see Fig. 23 a. The end cover fits inside the retaining stud circle, which facilitates the random positioning of the re-fluxing jets. It thus becomes possible to observe the occurances at the jet energising surface, although the events in the volute casing cannot be seen. A steel end cover, with an appropriate suction opening is used in place of the Perspex one for water tests, as shown in Fig. 23 b.

Three basic methods were used to assess the air pumping performance. The first, which was both qualitative and quantitative, employed a dry gas meter for large flows, and a wet gas meter at small rates of flow connected to the pump suction. The pressure control valve, which was of the rubber diaphragm type, was inserted between the meter and the pump.

The second method consisted of timing the rate of fall of pressure in a container of fixed capacity, 2·4 ft. The rate of change of pressure can be converted into a rate of loss of mass and thus the rate of evacuation. This method was extensively used for a direct qualitative comparison of various changes in pump geometry, see Figs. 46 and 47, on the basis of time/pressure fall comparison.

Finally, the third way of assessing the performance was by observing the rise of a free surface in a vertical water pipe with its open end submerged, and the closed one connected directly to the pump suction.

It is worth observing at this stage that neither the second nor the third method gave sufficient indication of instability, and hence the first method was used exclusively to obtain this important feature of the pump characteristic.

The short and curved length of recirculating passages posed a flow metering problem, which was overcome by terminating them with two straight lengths of 1" copper piping at the air separator end. These ends, which were fitted in flush with the bottom of the separator, had each a pressure tapping located 4" from the inlet. The pressure drop between the free surface in the separator and the tapping point was used as a rate of flow indicator, and Fig. 24 shows the calibration of this device, carried out against an orifice plate with the aid of another pump.

An additional pressure tapping was provided at the entry point of the recirculation passage into the pump in order to estimate the friction losses in the re-fluxing liquid. Both the casing and end cover are fitted with a number of pressure tapping cum Pitot insertion points distributed round the periphery, as shown in orientation sketch in Fig. 25. To provide additional entry points for velocity traverses, use was made of the end cover studs which were drilled right through the centre, providing the necessary guidance for the Pitot cylinder. Pressure distribution measurements were recorded on a multitube mercury manometer, but these had to be confined to the wall pressures. The pressure distribution across the stream was found impossible to determine.

To obtain velocity distributions a small Pitot Cylinder was manufactured, with a spirit level located protractor, to assist in estimating the flow direction. The instrument is shown in Fig. ~~26~~ 27. It will not be out of place to mention the use to which this particular device was put, in spite of its unconventional form.

The usual type of instrument for ascertaining velocity and direction would have two radial holes subtending ~~and~~ arc of 78^{0}, each hole being connected to one limb of a manometer. The size of the probe, 0·105 " dia., and the viscosity of the fluid, made it necessary to use only one relatively large radial hole 0.041" dia., which in turn meant alternativ directional rotation of the probe about its axis to obtain equal pressure readings. Tests carried out in a high speed air jet stream have shown the probe to be remarkably sensitive to the direction of the oncoming stream, in spite of the large size of the radial hole relative to the probe diameter. The velocity head coefficient was found to be I. In actual use, in conjunction with a mercury manometer, it was found that, when operating in the bubble mixture region, the change in pressure resulting from the rotation of the tube was too small to give a positive manometer indication. This phenomenon, resulting from the low stream density, rendered the tube insensitive to detecting direction, turning it at the same time into a probe for the position of the free water boundary inside the casing. This, in fact, was the only method available to gain any insight as to the location of and shape of the free water surface in the casing under air evacuation conditions.

PART III

EXPERIMENTAL RESULTS OF FEASIBILITY STUDY.

The results of the radial small jet injection hardly justify a mention, since they can be described as totally negative.

The shroud method of jet acceleration shown in Fig. 19 a and b indicates that the actual jet spread is not greatly affected by the speed of the jet. This is fairly understandable, since the impeller speed is of the order of 95 ft./sec., whereas the jet speed is 2·6 ft./sec. and 5·2 ft./sec. in the corresponding pictures; and the jet orifice distance from the shroud is slightly more than $\frac{1}{4}$ jet diameter.

Further pursuit of these characteristics was not desirable, since the jet would operate in a closed casing, possibly with different discharge characteristics. Tests carried out with a jet in situ have shown that the effect of the impeller rotation was small, with a slight tendency to increase at bigger flows, see Fig. 28. This was primarily due to a vortex formation in the shroud space, which increased the effective head across the jet orifice. The formation of this vortex is naturally a function of the degree of filling of the clearance space, which takes place rather suddenly at a certain stage, as will be described later.

Tests were now carried out to ascertain the best circumferential position of the re-fluxing jet fed from an external source, while varying air suction pressure. It became immediately apparent that the air separation and flow in the discharge taper were very unsatisfactory, due to the presence of eddying effect in both the axial and circumferential plane. This was accompanied by pulsation of flow into the separator. As the simplest expedient, a number of axial baffle arrangements, shown in Fig. 29, were tried, and it was found that baffle 'c' reaching down to the impeller centre line, in a plane parallel to the axis of rotation, was most effective. A feature of this layout was that most of the air bubbles came up in the outer portion of the discharge taper, split in two by the presence of the baffle. This suggested the presence of a free water surface in the casing outwith the impeller, but no steps were taken to investigate the matter further at the time.

Air pumping tests were carried out with external recirculation to avoid air re-entrainment effects. These results shown in Figs. 31 to 36 underline the importance of the recirculation on the air entrainment. The relationship is almost linear for all jet positions, and vacua, up to 8" of mercury, with few exceptions. It is also self evident that the jet positions 4 and 5 are most advantageous, and these are almost diametrally opposite from the casing tongue.

In the case of the $7\frac{1}{4}$" impeller, the jet position appears to be of lesser importance at higher vacua, Fig. 32, whereas with a 6" impeller, and thus clearance of the tongue of the order of $\frac{5}{8}$", the jet positioning is of greater significance, Fig. 36. This points to the importance of the events taking place at the tongue of the pump and suggest a somewhat different entrainment mechanism at higher vacua.

In fact, contrary to expectation, the 6" impeller appeared to give a larger air pumping capacity as the vacuum increases up to 8" Hg for the best positions of the jet, i.e. 4 and 5.

A possible reason for this course of events was that the water surface was subject to increased turbulence due to a part of the jet extending beyond the impeller periphery. In addition, the tongue of the pump was too far from the impeller tip to produce a low pressure region which would be less effective in distorting the free entraining surface position, aiding the entrainment to a greater extent, than the actual positioning of the jet for small tongue clearance with a larger impeller.

The low pressure peak at the tongue displayed in all tests carried out by Welte suggests that such reasoning is not out of place.

To return to the quantity effect, it can also be expected that, since the quantity is connected with both the momentum available for compressing air, and surface area of the water stream, it can have a direct bearing on the air pumping capacity. These two effects must not be confused since the first will only effect the energy available for compression, while the second controls the actual mechanism of entrainment and is largely dependent on the geometry of the pump; hence all attempts to use wide impellers with few blades, in order to get a large entrainment area, Fig. 37, shows the air pumping capacity against vacuum for the two impellers with recirculating flow, and it is fairly obvious that the larger impeller is more effective at low vacua, though for the range of 4" Hg. to 8" Hg. the advantage is rather doubtful.

The water pumping performance shown in Fig. 38 is self explanatory, except for improved efficiency beyond a flow of 260 g.p.m. At maximum efficiency with a total head of 100 ft. the pressure differential across the re-fluxing passage is not likely to exceed 30 ft., giving a re-fluxing flow of about 25 g.p.m.

Hence the proportion of output energy of the pump being wasted ought to amount to $\approx \frac{25 \times 30}{260 \times 100} \approx 3\%$, which corresponds fairly closely to the actual reduction in efficiency.

The final test of evacuation rate with a vertical 10 ft. 3" bore line gave an evacuation time of 3 ~~m/~~ min. with a temperature rise of approximate 3° F for a water content of 3 gallons. A test was then carried out to find the effect of water temperature on priming time. This effect, shown in Fig. 39, indicates a 50% extension of the priming time for a 3" bo 10 ft. high column, which would correspond to the pump operation for a temperature rise of 25° F. up to 95° F.

If the pump starts with water at 70° F. it could operate for 25 minutes without the priming time being increased beyond 4 minutes. The project was thus proved feasible for commercial development and accordingly a much more refined approach, with more elaborate instrumentatio was adopted. The findings of the feasibility study can thus be summarised as follows :-

1. The pump can be made self priming without use of valves.
2. The effect of the impeller tip clearance at the tongue is not clearly defined at vacua of 4" Hg. to 8" Hg.
3. The refluxing flow should be maintained as large as possible.
4. The refluxing jet should be positioned diametrally opposite to the volute tongue.
5. With a 3 gallons water retention volume, the temperature rise during priming of a 10 ft. 3" bore suction column is of the order of 3^{o} F.
6. Priming time increases from 3 min. to 4 min. as the water temperature rises from 75^{o} F. to 105^{o} F.
7. The drop in water pumping efficiency is of the order of 4%.

EXPERIMENTAL FINDINGS OF DEVELOPMENT AND DETAIL STUDY.

The major proportion of this effort was directed toward securing a much improved machine, and only after this was accomplished some time was directed to a more detailed exploration of the flow pattern.

The use of a Perspex end-cover revealed immediately the reason for the effect of the discharge baffle. In view of the low pressure generated at the tongue of the casing, a vigorous recirculation takes place from the throat of the taper, back into the volute beyond the tongue in the direction of rotation. This is shown in Fig. 40 which displays the vigorous recirculation even with the baffle in position. It is also noticeable that the space between the shroud and end cover is not fully occupied by the jet, which only covers an arc of about 120^{0}, the majority of which extends in the direction of rotation. The jet entrains vigorously air drawn partly via sealing rings by splash, and this continues until the shroud space becomes completely filled with water, in consequence of rising vacuum and increased refluxing jet flow, as well as inflow, from the volute itself at some points.

This type of entrainment accounts for a large proportion of the air pumping capacity at small vacua, as has been ascertained by running tests with the impeller eye completely sealed off. The results in Fig. 41 show that, while displaying a small degree of instability, there is a marked tendency to a reduction in the maximum attainable vacuum, if the impeller eye is closed, and baffle removed.

As a deduction from the observations, it was decided to employ a second refluxing jet, positioned in such a way as to prevent internal recirculation without the use of a baffle. The result of this geometry was quite startling. The recirculation was arrested and the air pumping capacity was similar to that with a baffle over the full range of vacua. The air entrainment method did not differ from that of the single jet case, but, due to both greater entrainment surface and momentum, there was an appreciable gain in both air flow and maximum vacuum.

Fig. 41 shows the appropriate performance curves. To improve the energy transfer from the impeller shroud to refluxing fluid, it was decided to reduce the clearance between the end cover and the impeller to 3/32" at the point of closest proximity. The improvement was quite startling, but all characteristics continued to display an instability in the region of suction pressures 3' to 4', as can be seen from Figs. 42 and 43. Figs. 44 a, b, c, show the events in the shroud clearance space. These are, a vigorous double jet entrainment followed by complete filling of the space with clear fluid. The clouding at high vacua is produced by re-entrainment of air bubbles from the separator. There is no recirculation in this shroud space.

The priming time for a 4" bore column 10 ft. high was

2 min. 23 secs., which was an almost 50% improvement on tests carried out with a 3" bore column during the Feasibility Tests. The water temperature rise was only $1\frac{1}{2}^{o}$ F., which was also very satisfactory in view of the reduction of the filling volume from 3 gallons to $2\frac{1}{2}$ gallons.

A number of jet positioning combinations and tests with two impellers of reduced diameter were also carried out and the results are shown in Figs. 45, 46, 47, 48 and 49.

In general it was found that the best jet positioning was at the tongue and diametrally opposite. The impeller diameter was found to be most important, though it is not possible to ascertain whether this was principally on account of varying casing tongue clearance or merely due to more effective energising of the recirculating jet.

Water pumping performance tests were carried out with all three impellers and recirculating flow. The results shown in Fig. 50, 51 and 52 give a drop in peak efficiency of 9%, which is still considered acceptable, although it may be improved by further development work.

A feature of these tests was the drowning of the pump inlet to a depth of 1 ft., to offset any possible onset of cavitation at large flows. At this stage the development work was considered adequate for design purposes.

The only remaining problem was to try to throw some light on the instability and volute flow conditions. Accordingly, detailed explorations of the casing pressures and velocities were carried out wi the $7\frac{1}{4}$" Dia. impeller, and the most efficient twin jet layout. The tes were conducted at zero pressure and suction pressures below instability point and above it. In spite of repeated efforts to build up a compl picture of the flow pattern, the conditions at various parts of the casing were so turbulent that it was impossible to produce a complete survey. However, Figs. 54, 55, 56 and 57 give a general picture of the events in the pump during priming.

In the Pitot traverses it was assumed that when the tube is insensitive to rotation, it is in the region of low density flow or an empty cavity. It is realised that such assumptions may not appeal to some experimenters, but there was no alternative to such deductions, save the building of a transparent pump, which was out of the question at the time.

The work involved in the collection of this information was extremely time consuming, and called for a great deal of improvisation of details, which are not mentioned here, since the obvious purpose of the investigation was to develop a pump and understand its operation and not to engage in manufacture of experimental equipment.

The other unfortunate feature is an almost total absence of a basis for mathematical analysis due to the excessive turbulence of the flow pattern.

The development and detail investigation conclusions are as follows :-

a) It is desirable to reduce the impeller shroud clearance in the casing to improve energy transfer to the refluxing jets and arrest as much as possible the recirculation of the air water mixture from the casing throat past the tongue.

b) Two jets, one at the tongue and the other diametrally opposite, appear to be adequate for the purpose of producing a satisfactor gas entraining recirculation.

c) The priming process consists of two basic stages. In the first the flow energised by the shrouds causes the gas to be entrained in the shroud clearance space as well as at the free surface in the casing due to turbulance. This is obvious from Fig. 55 where the velocity actually increases towards the stationary casing wall and falls to almost zero within $\frac{1}{4}$" of the periphery of the impeller, which is moving at 95 ft. /sec. As the free water surface is drawn inward radially by increasing vacuum, the impeller drag becomes apparent until at 28" Hg. vacuum the free water surface actually enters the impeller passages and air pumping ceases. There is no doubt that the inflow from the shroud space delays this event but cannot prevent it.

e/

In an examination of Figs. 56 and 57 this event can also be observed. In fact at the lowest vacua the refluxing jet water is the only apparent source of energy, the velocity falling towards both axial casing walls, right at the periphery of the impeller. Fig. 57, which shows the axial traverse $\frac{3}{4}$" outward from the impeller tip, confirms these characteristics to a lesser extent, but even there we seem to have a lower velocity at the centre due to the absence of impeller outlet area drag. This traverse is not quite as informative as Fig. 56, probably in view of the complexity of flow in the volute.

The attempt to ascertain the direction of flow either at the impeller boundary or in the volute itself have shown that the flow is invariably circumferential and in the radial plane. No axial components could be detected.

If we now turn to the second stage, which is reached as soon as the impeller drag becomes apparent, we become immediately aware of a vacuum being created at the tongue of the pump casing, which is a result of the formation of a cavity. The boundaries of this cavity are formed by the casing, whereas the highly turbulant inflow from the shroud spaces provides the air entraining medium. The cavity is in direct communication with the impeller eye at the tip of the tongue, as can be seen from the extrapolation of the casing pressure curves beyond the actual measuring point to the tip of the breakwater - see Fig.5

The two curious bumps at 160^0 and 280^0 may be due to the acceleration of volute flow at a point downstream from the main jet entry blockage effect.

d) It would perhaps not be out of place to throw some light on the power taken by the pump in priming. The quantities of air flowing are so negligible that the power absorbed in doing work on this medium can be overlooked. If we take the total HP and the net energy supplied to the recirculating liquid - see Fig. 58 - it becomes self evident that the pump is, in fact, an excellent water brake under priming conditions. The temperature rise confirms the estimate of these losses, pointing to most unnatural flow conditions inside the machine. These conditions disappear instantly when the water pumping duty commences in spite of the continuous presence of the re-fluxing stream.

e) The change in the water pumping characteristics, especially with the $7\frac{1}{4}$" impeller, is explained by the reversal of re-fluxing flow, thu creating an additional outlet area.

f) Comparison with results of tests carried out by Welte shows that the U - II is better than D and G types, and only inferior to D type which is of the parallel variety with a water pumping efficiency of only 45% compared with 62% of U - II.

As regards the effect of tip speed on attainable vacuum, it is obvious from Fig. 13 that the tip speed is the predominant factor because all four pumps , in spite of vastly different constructional details, atta[illegible] a maximum vacuum of about 9·4 m.H_2O at a tip speed of the order of 18 m/se[illegible] (59 ft./sec.) . The vacua attained at lower speeds do not vary significa[illegible] from one pump to another. This establishes positively the shroud re-flu[illegible] method as a practical possibility, and Fig. 59 shows a highly satisfactory priming time for up to 20 ft. lift on a 4" bore suction.

g) To see that comparison with water jet pumps is rather pointless, let us examine Fig. 60 . This shows the ratio of air volume entrained at suction pressure to the volume of entraining liquid for a water jet pump using 1·6 $ft.^3/min.$ of water at a differential pressure of $30^{lb}/in.^2$ across the water jet nozzle, in comparison with a similar ratio between air and recirculating water volume for U -II. The power required to drive the water ejector is $\frac{1}{5}$ H.P., whereas the U -II would absorb between $1\frac{1}{2}$ - $3\frac{1}{2}$ H.P. for the same range. This striking comparison is meaningless for engineering purposes since the ejector is meant for continuous duty, whereas the priming time constitutes an infinitesimal percentage of the life of a pump.

The only valid observation to be drawn from f) & g) is that the shroud re-fluxing method can produce overloading of a motor, on a low powered water pump, and must not be used in an indiscriminate fashion.

C O N C L U S I O N S

The work described in the preceding chapters has shown the reasons for difficulties in producing a self-priming adaptation of a centrifugal pump. This can be summed up as a shortcircuiting of the tongue cavity to the discharge taper throat through the shroud clearance space, which is the crux of the problem. The method of energising the re-fluxing fluid is a matter of choice, but has been devised to serve a dual purpose. Firstly it improves air pumping capacity at low vacua due to direct jet action; secondly it arrests recirculation of entrained air within the casing itself.

The re-fluxing jet passages can be incorporated in a standard end cover and left plugged up when not in use. The only additional item thus becomes the air separator. Design studies, under way at present, indicate that such a device can be produced in a small enough size to make the unit commercially viable. The only extra items to be carried by the pump maker, would thus be an air separator and a simple water retention flap.

As regards the analysis of the self-priming process itself, it is quite obvious why the pump will not prime if a high speed, shrouded multiblade impeller is used in a casing with a reasonable shroud clearance

The importance of the findings lies in recognition of the fact that the only really active portion of the pump is that of the diverging casing portion at the tongue, which causes the formation of an entraining cavity. The M pump, Fig. 4 , has six such cavities (!), but the jets are not as effective as in the suggested layout with the volute and impeller of U -II.

A further study of the entrainment in this cavity is a subject, which ought to lend itself to much theoretical and experimental study at a future date. Such work need not necessarily be done on the pump itself, which will assist materially in the control of boundary conditions.

The design of an air bubble separator is a study in itself and can be investigated in conjunction with both uniform and non-uniform mixture flow into this device. The behaviour of this mixture is a subject, which has been widely explored in the field of Chemical Engineering and no doubt an intensive search through existing information will reveal much relevant data.

At a guess, one would hope to improve the present output by at least 50% through further development, though such an expectation will not be easy to realise without fairly extensive modification of impeller clearances in the casing.

It is hoped that the information presented in this brief treatise will settle some of the doubts about factors affecting the self-priming of a centrifugal pump, and suggest to others a fresh approach to this rather fascinating problem.

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I N D E X

LIST ' SYMBOLS USED

Vair – Rate of air-flow ft^3/sec.

Uo – Impeller outlet tip speed.

bo – Impeller outlet width.

Do – Impeller outlet tip diameter.

φ air – Air flow coefficient.

INDEX OF ILLUSTRATIONS AND GRAPHS.

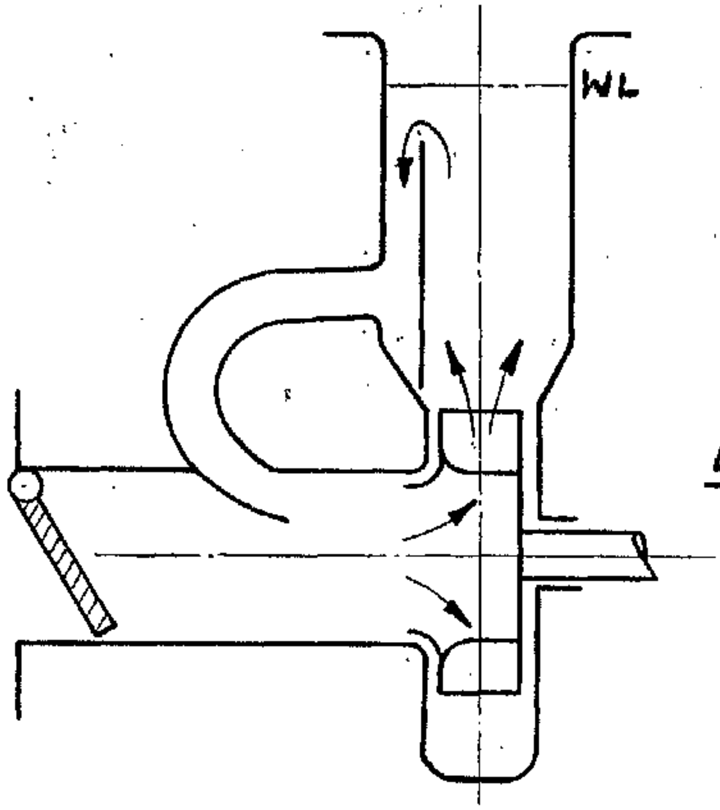
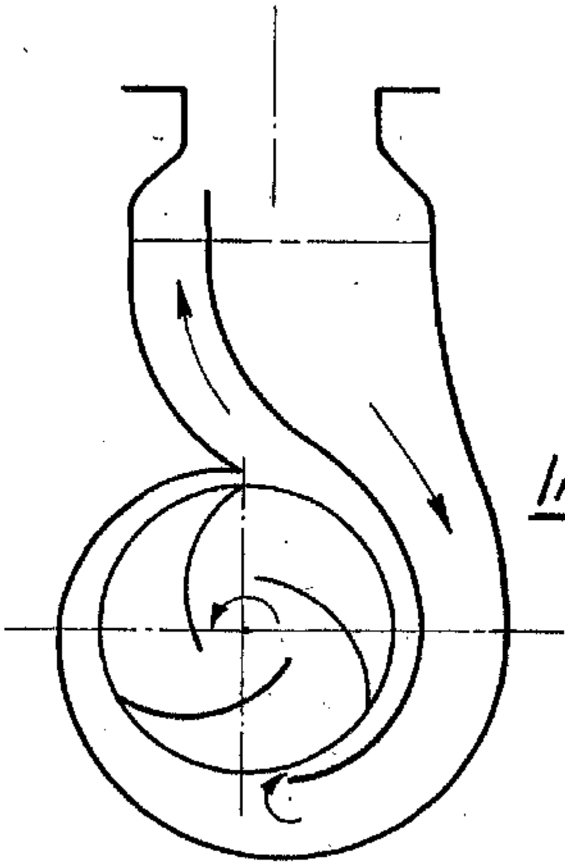


Fig. 1

External Refluxing



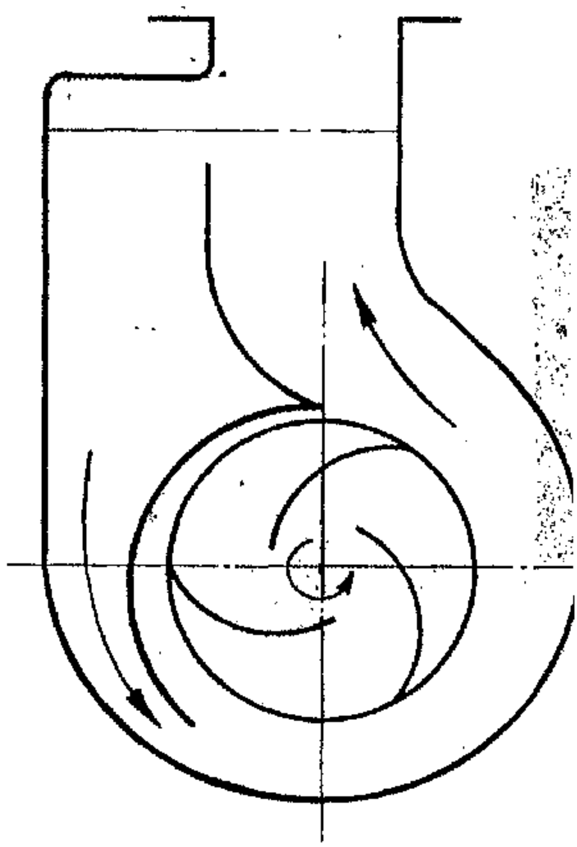


Fig. 2

Internal Refluxing

a

Contra-Flow

b

Parallel Flow

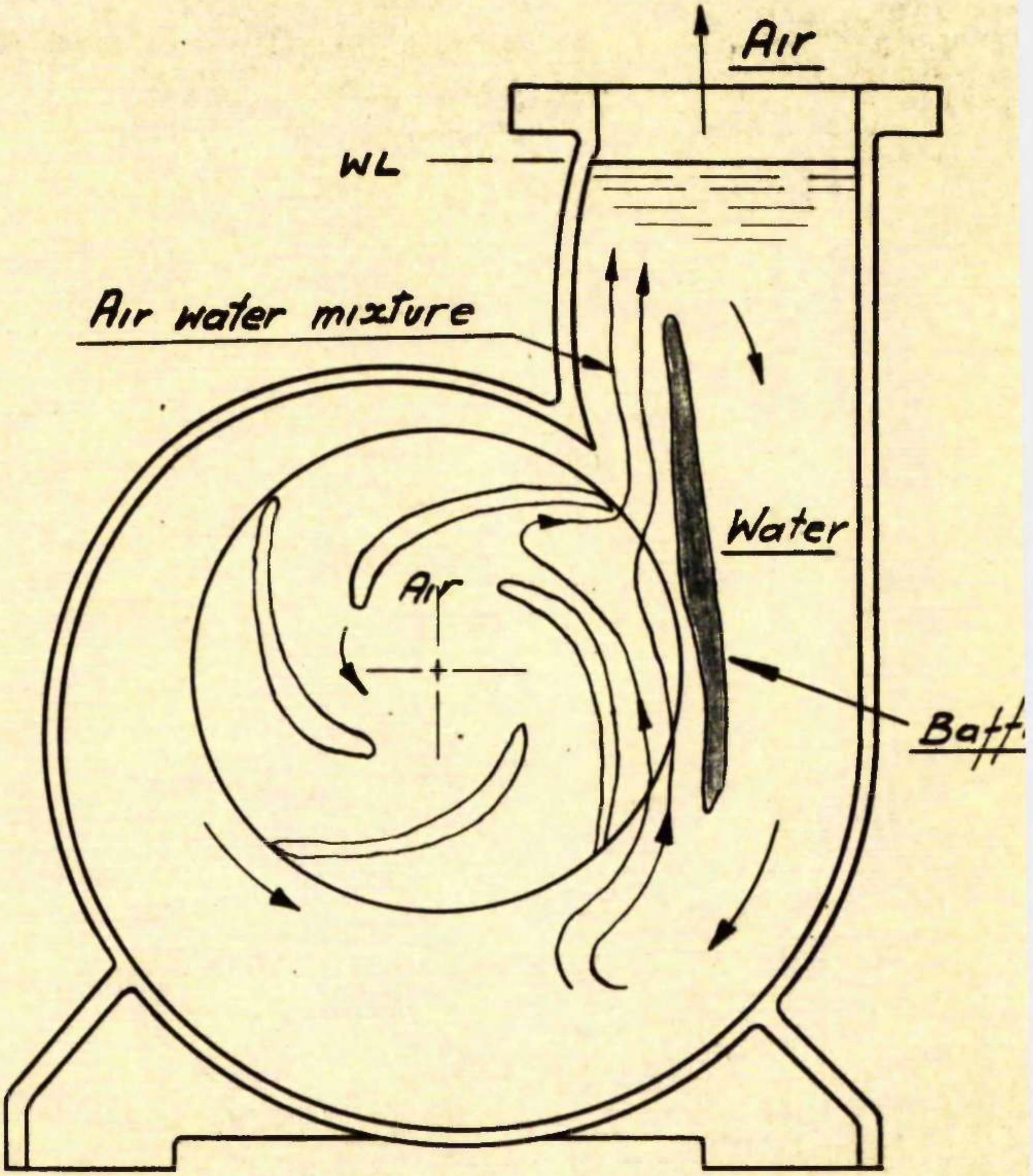
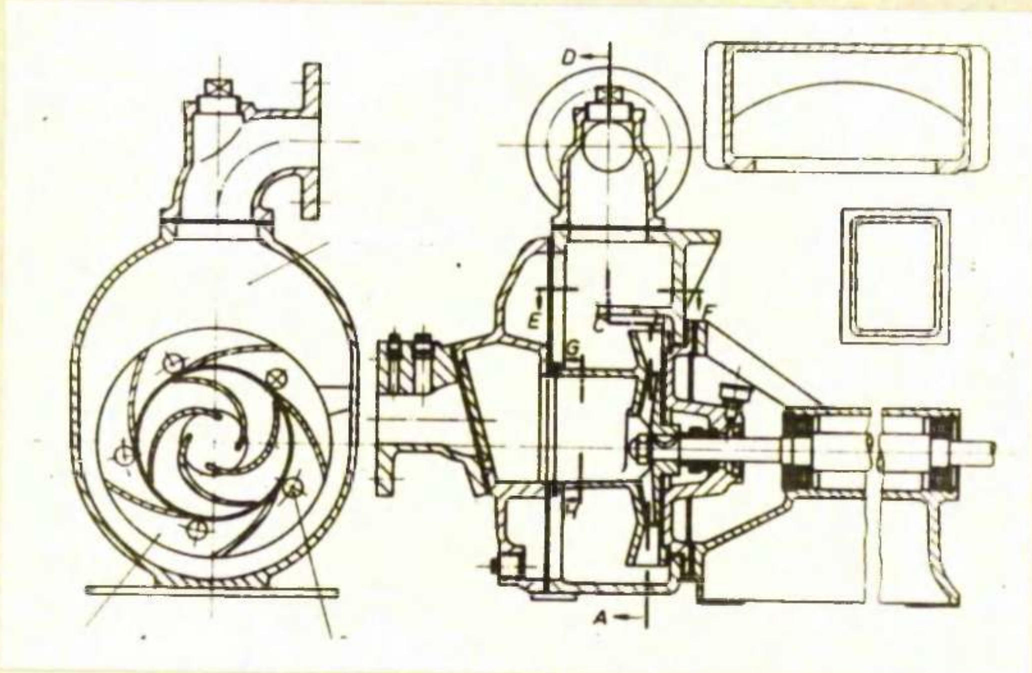


Fig. 3

The "Hannibal" self-priming pump



M - Pump

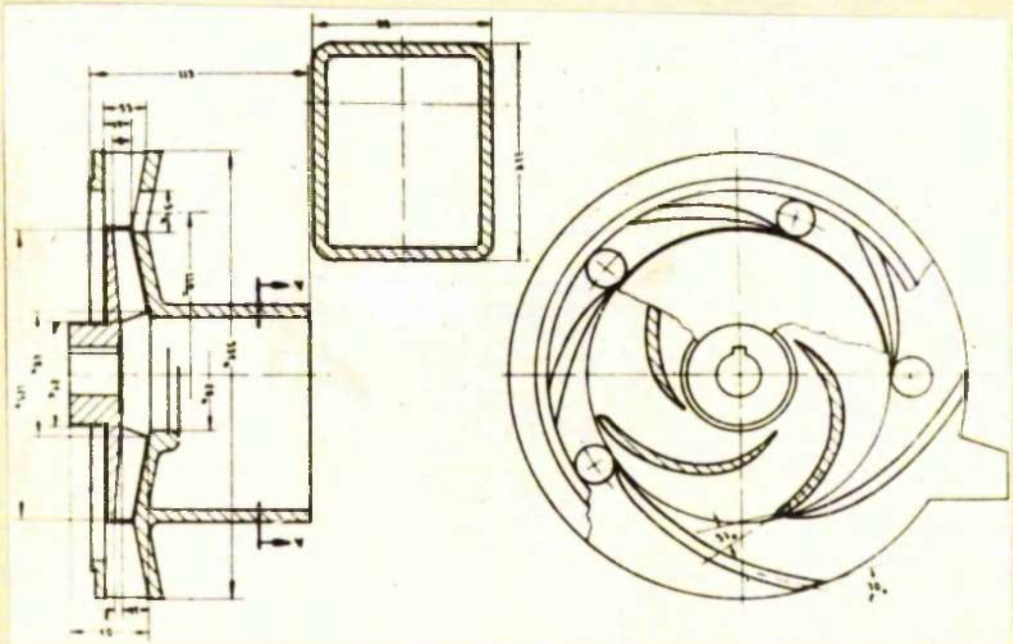
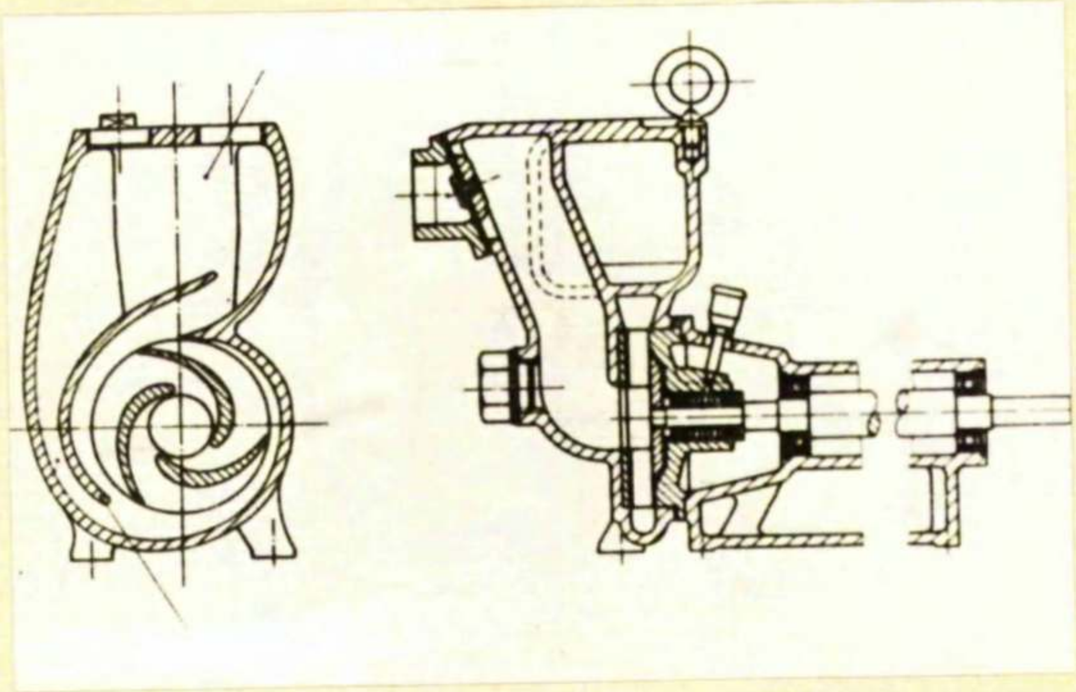


Fig. 4



G – Pump

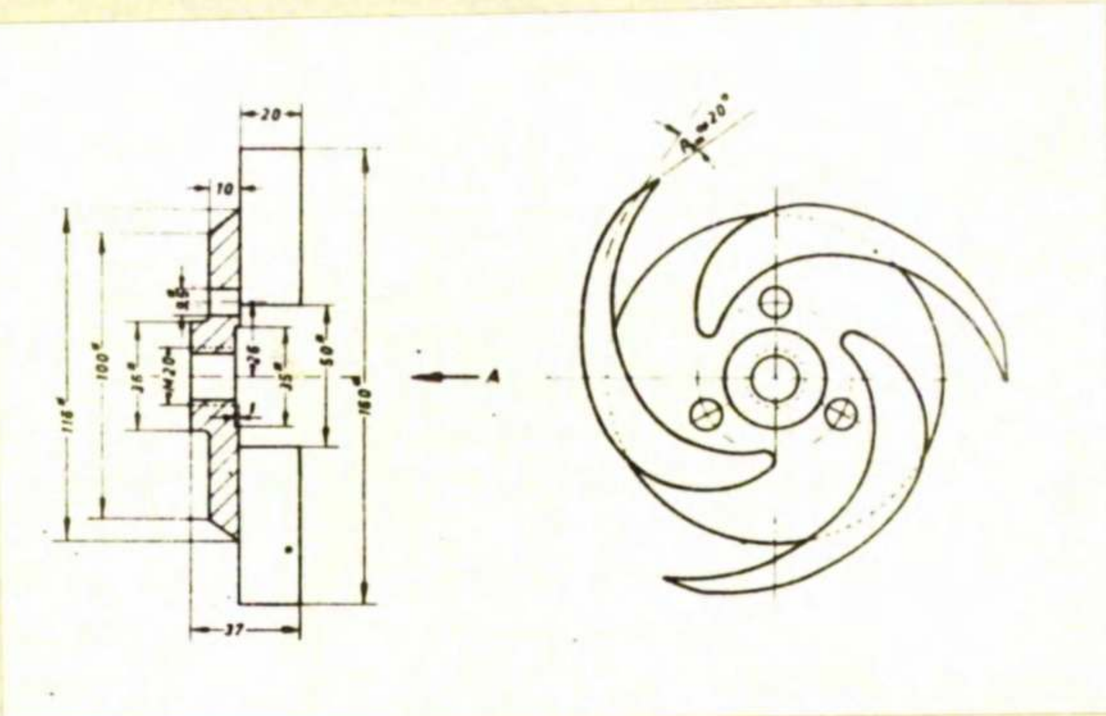
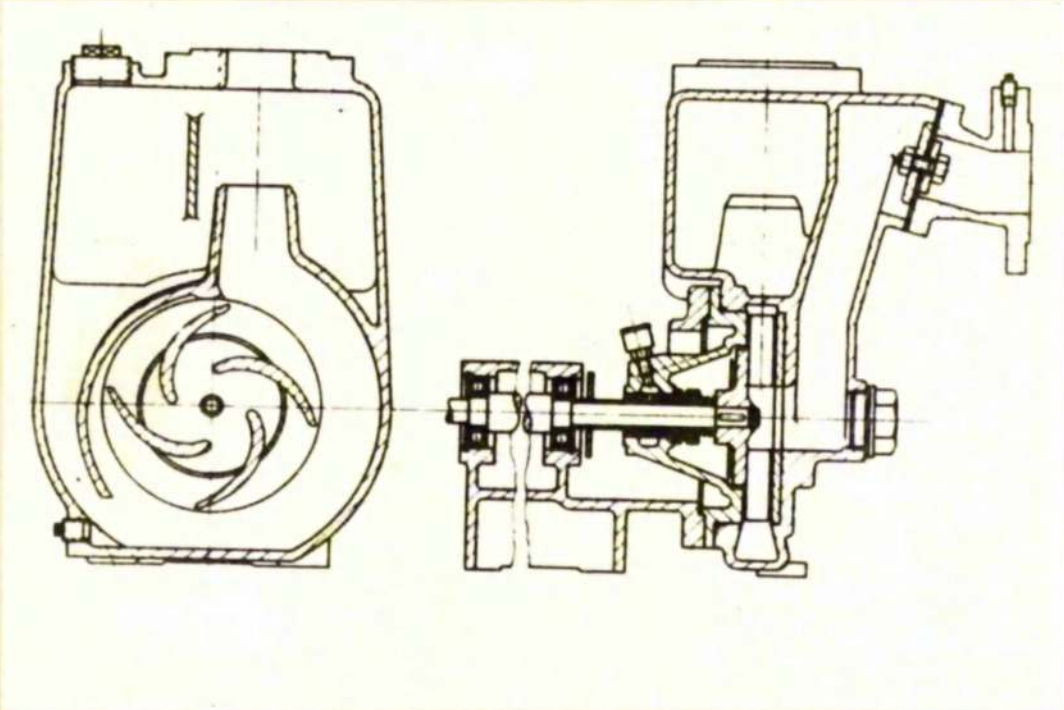


Fig.5



D - Pump

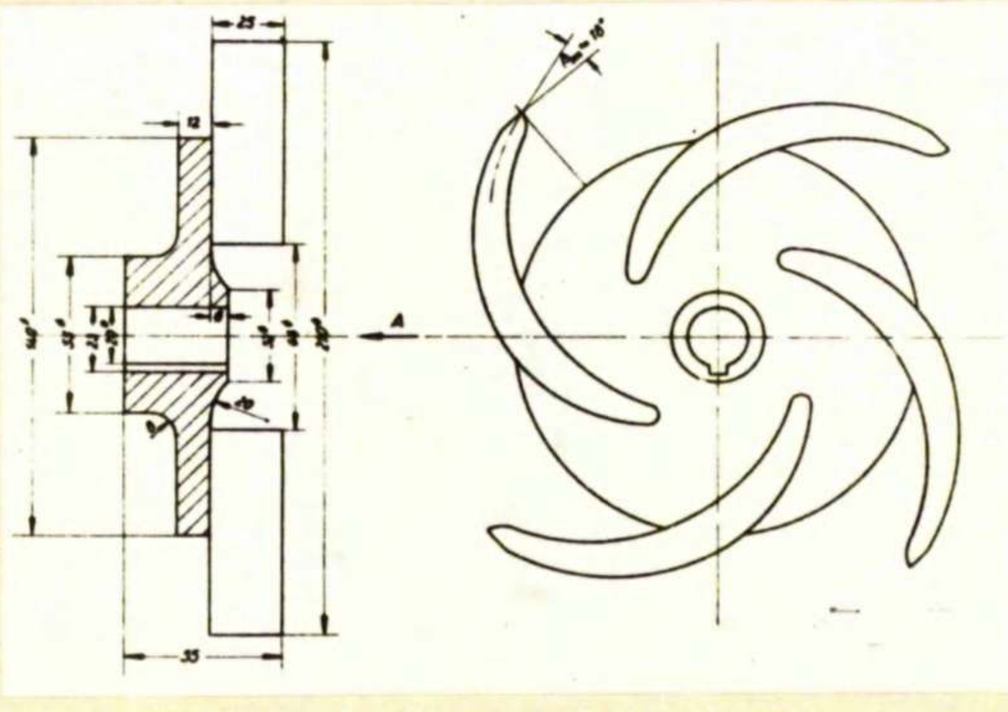
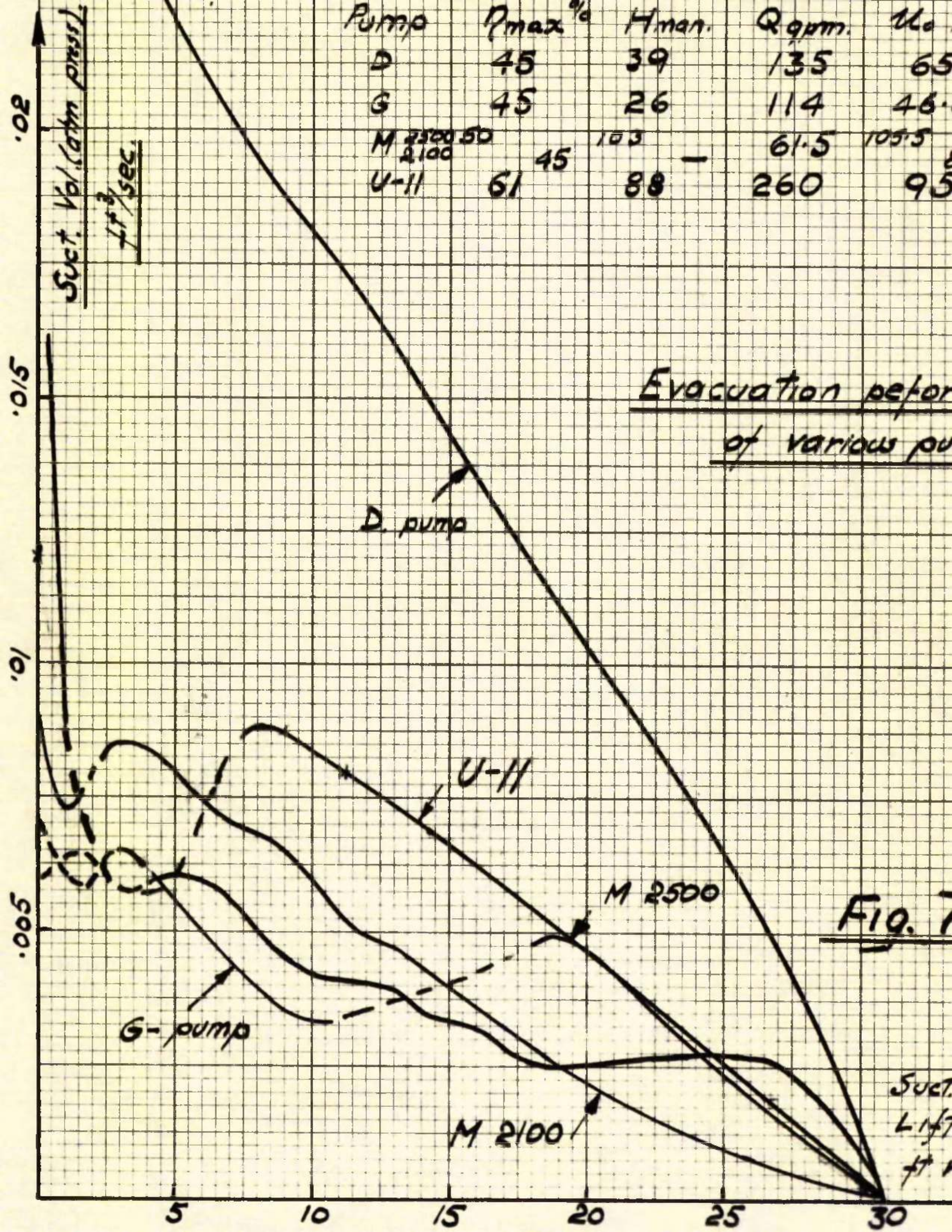
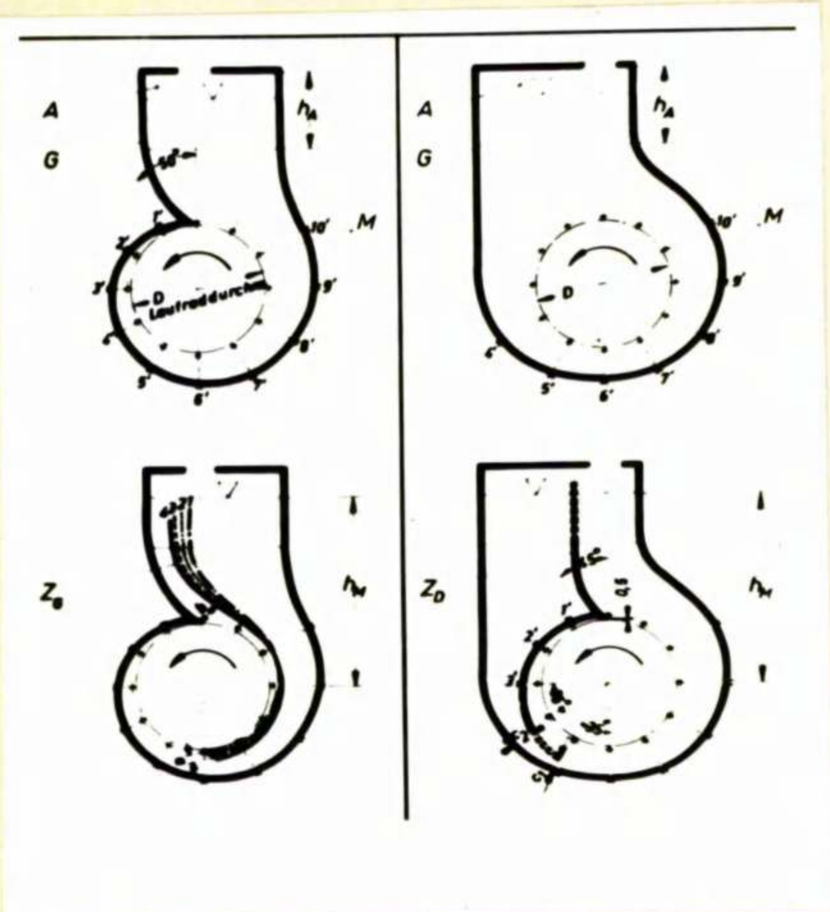


Fig.6


Pump ηmax % Hman. Qgpm. Uo
D 45 39 135 65
G 45 26 114 46.
M 2500 50 103 61.5 105.5
M 2100 45 —
U-11 61 88 260 95
Suct. Vol. (atm press.) ft³/sec.
Evacuation perfor
of various pu
D. pump
U-11
M 2500
M 2100
G- pump
Fig. 7
Suct.
Lift
ft
.02
.015
.01
.005
5
10
15
20
25
30



Pump casings and impellers of A. Welte.

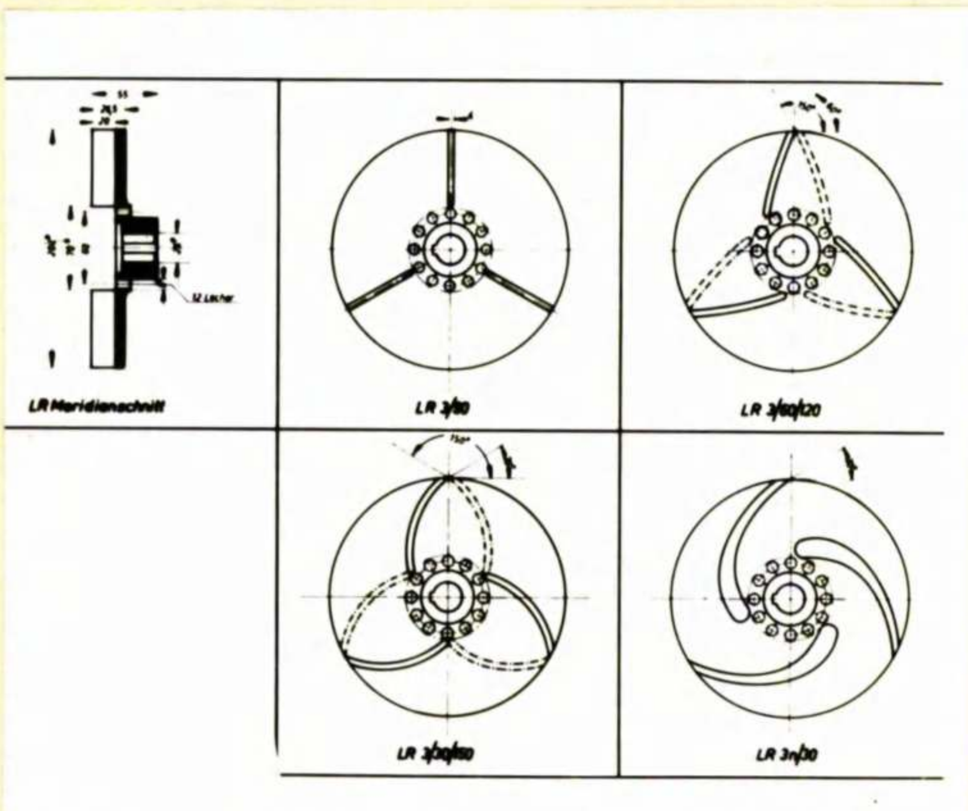
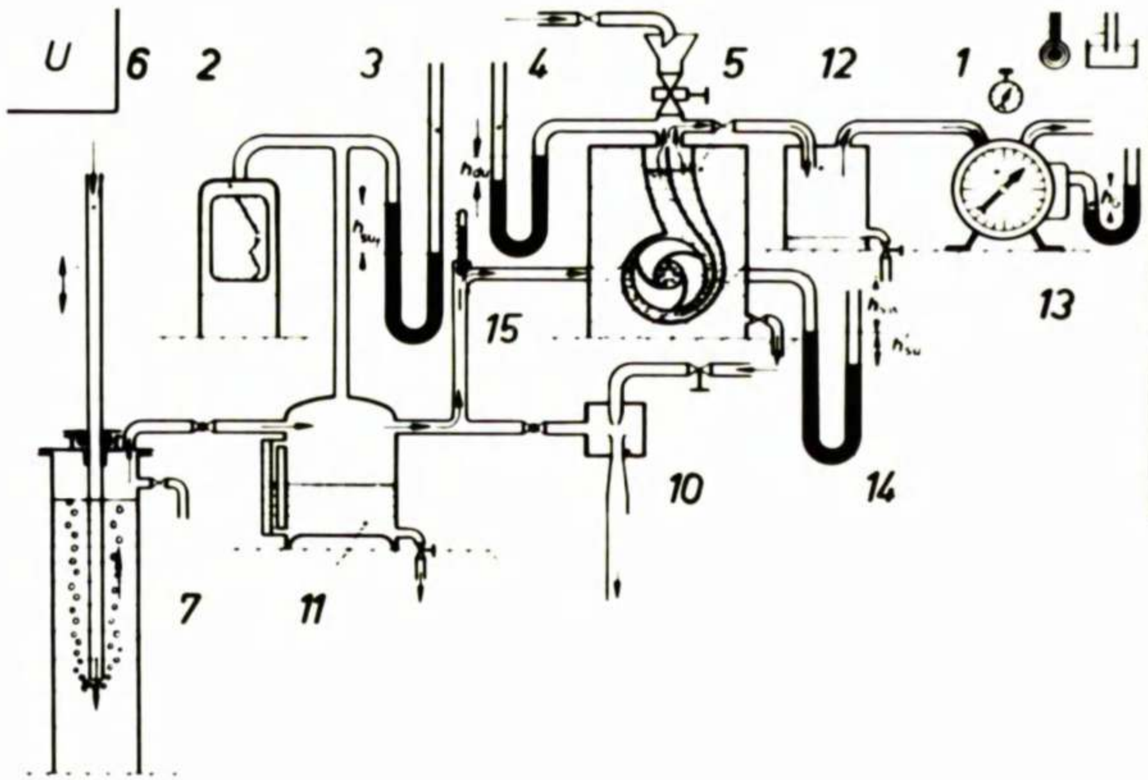
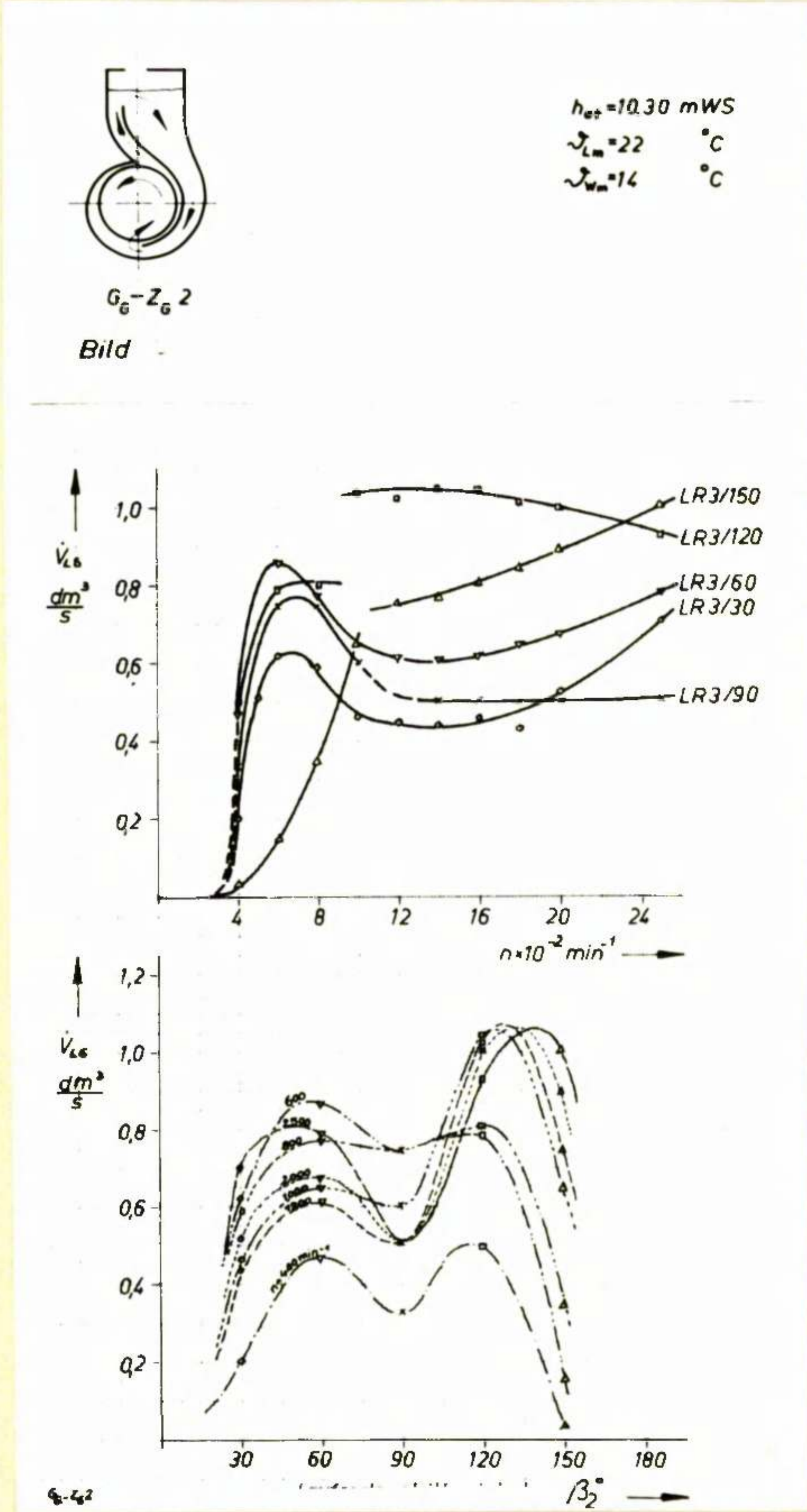


Fig. 8



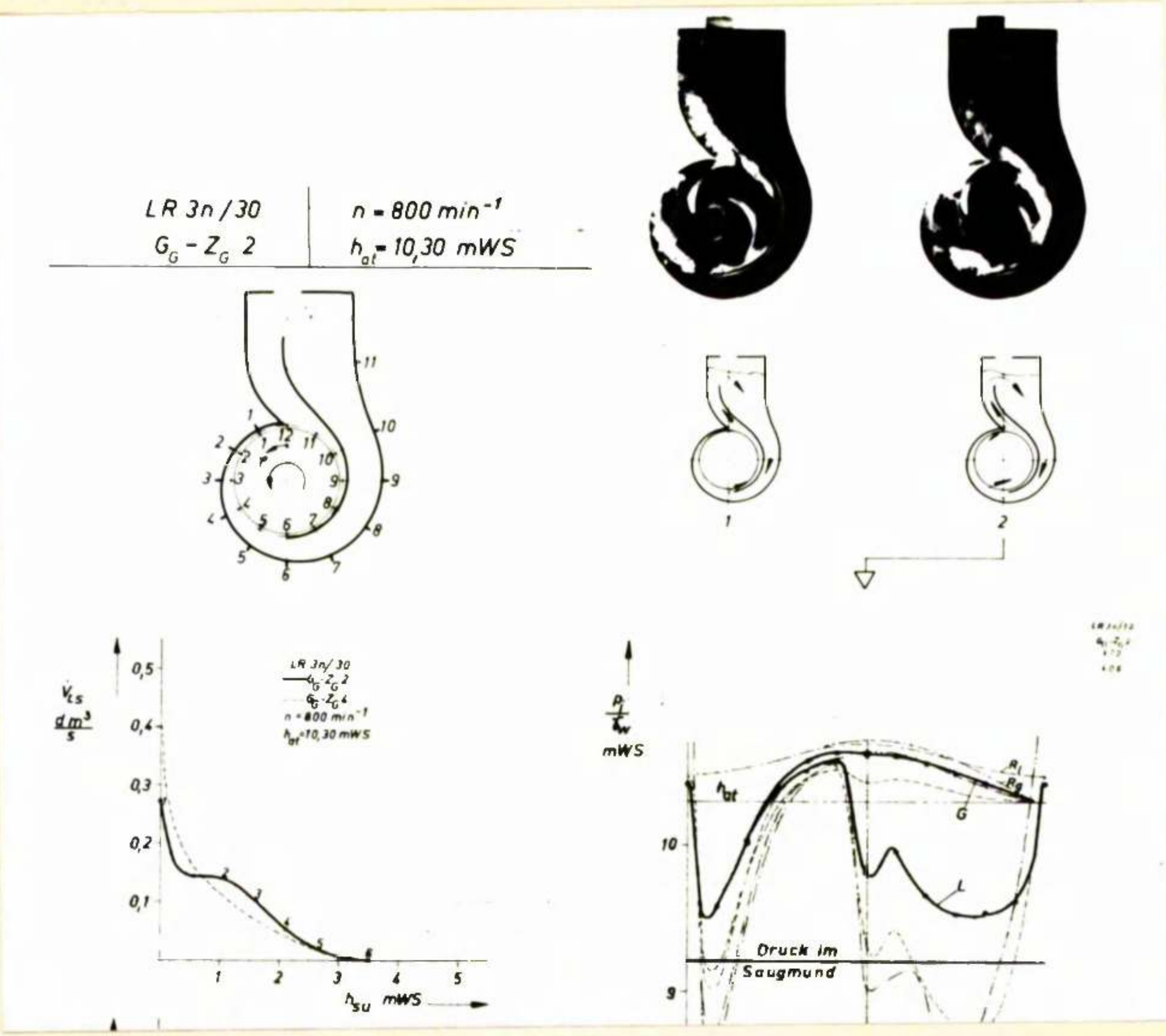
Priming performance test circuit of A. Welte.

Fig. 9



Effect of outlet an
and speed of rota
on evacuation perfo
(from ref. 2)

Fig. 10



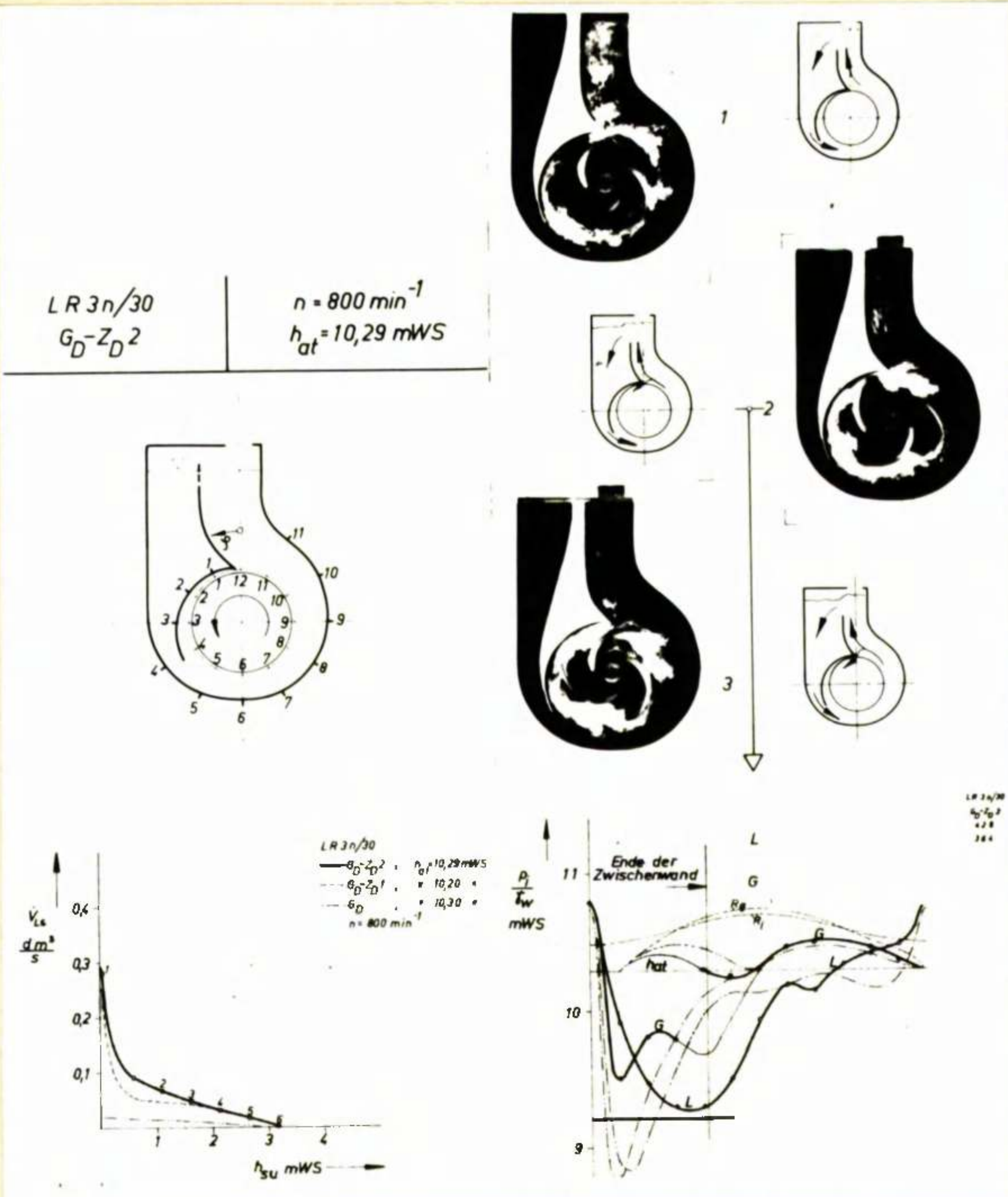
$\dot{V}_{LS}$ - Suct vol.
h su - Suct lift

hat. - atm. head
Curve G - casing wall pressure
Curve L - runner tip pressure

Pumping volume and casing and impeller tip pressures of a contra-flow pump at varying suction lifts.

(from ref. 2)

Fig.



$\dot{V}_{LS}$ - Suct. vol.

hsu - Suct. lift

hat. - atm. head

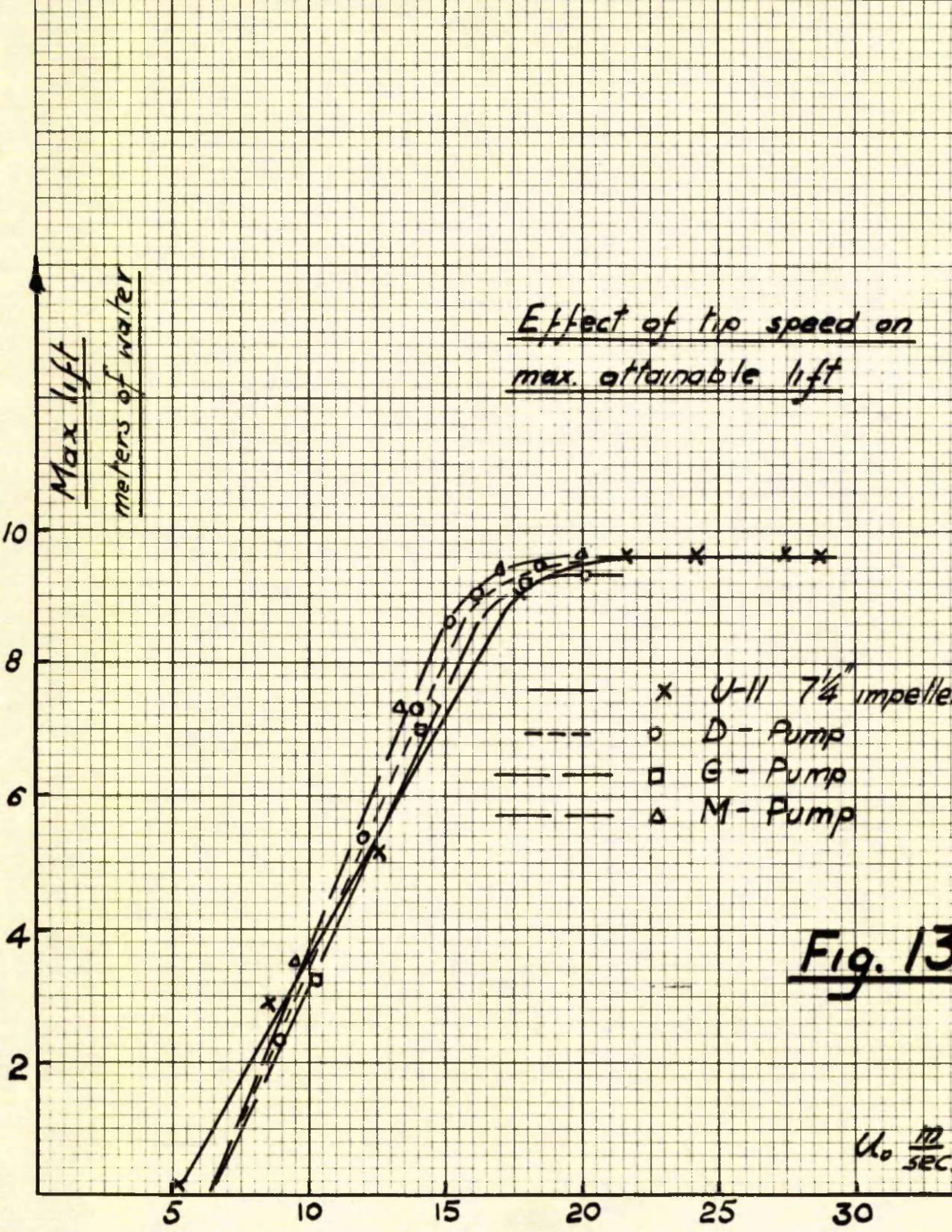
Curve G - casing wall pressure

Curve L - runner tip pressure

Pumping Volume and casing and impeller tip pressures of a parallel flow pump at varying suction lifts.

(from ref. 2)

FIG



Max lift
meters of water
Effect of tip speed on
max. attainable lift
U-II 7¼" impelle
D - Pump
G - Pump
M - Pump
10
8
6
4
2
5
10
15
20
25
30
$U_o \frac{m}{sec}$

Fig. 13

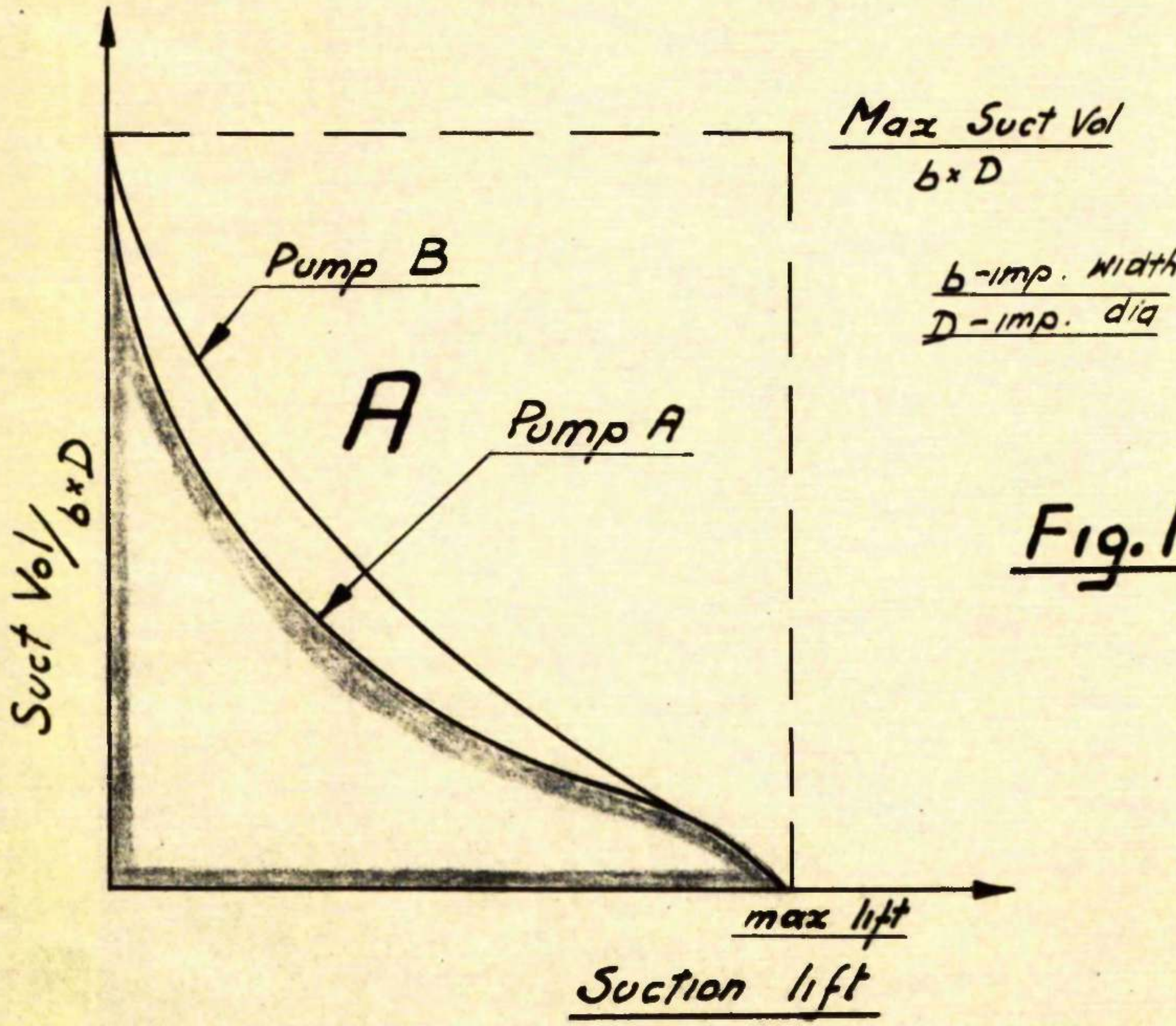
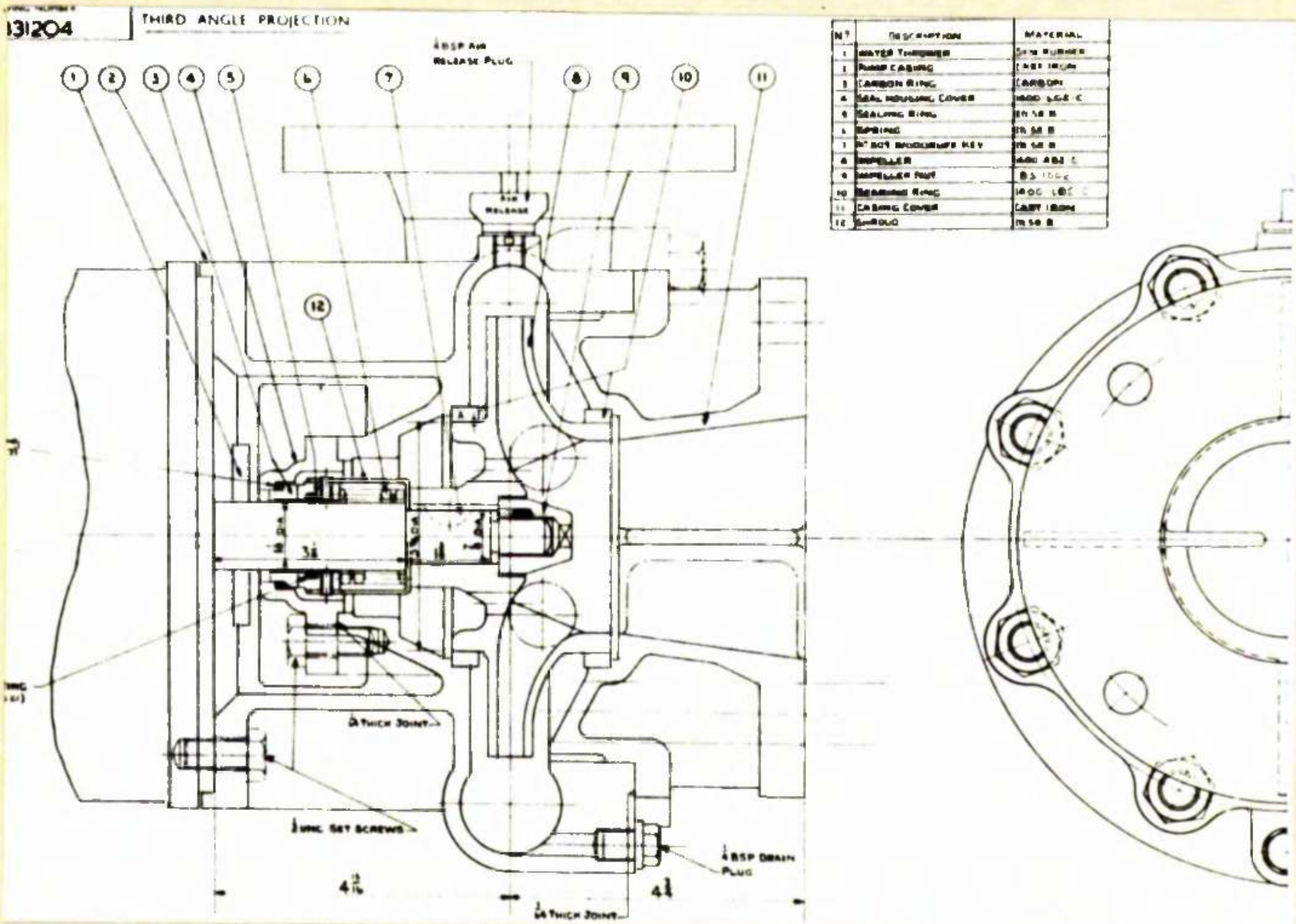


Fig. 14

Concept of Capacity Coefficient

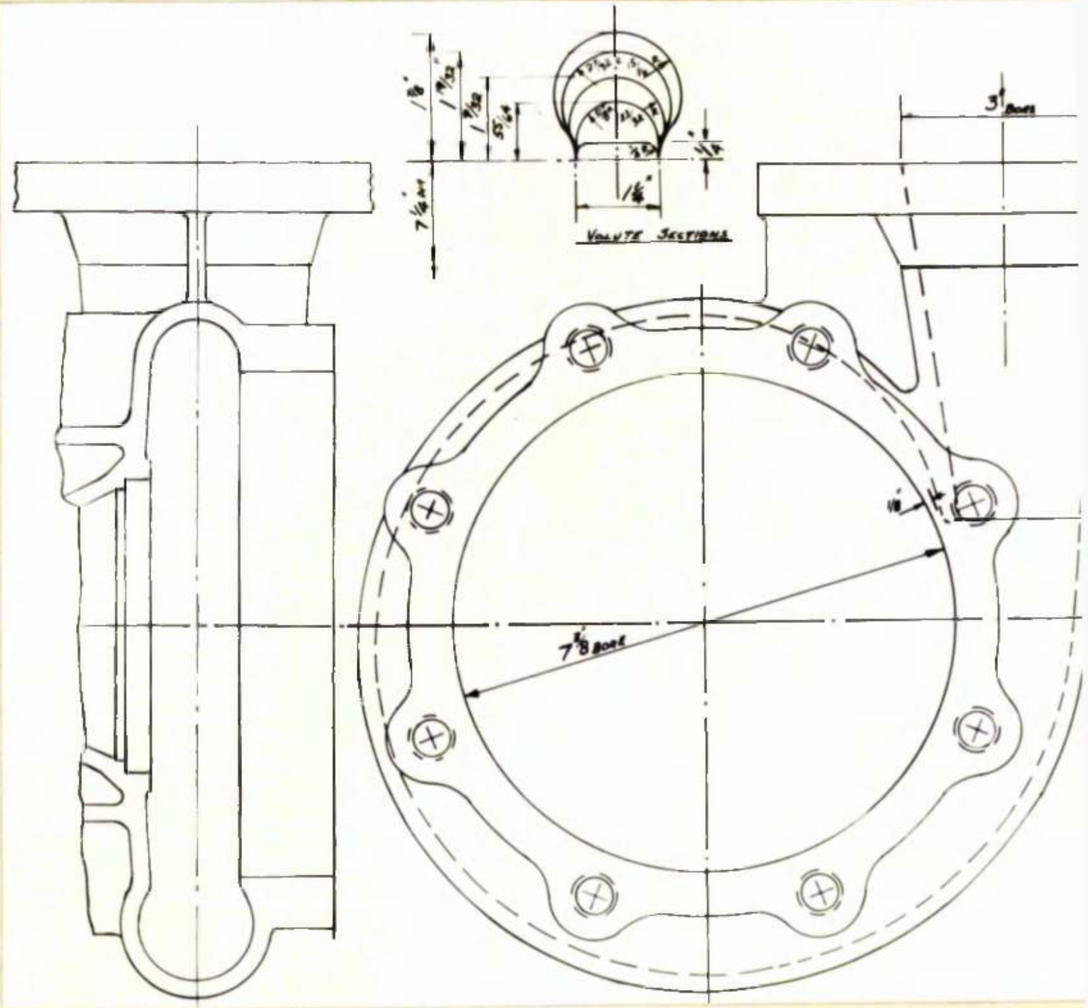
Pump B has a greater Coefficient than pump A

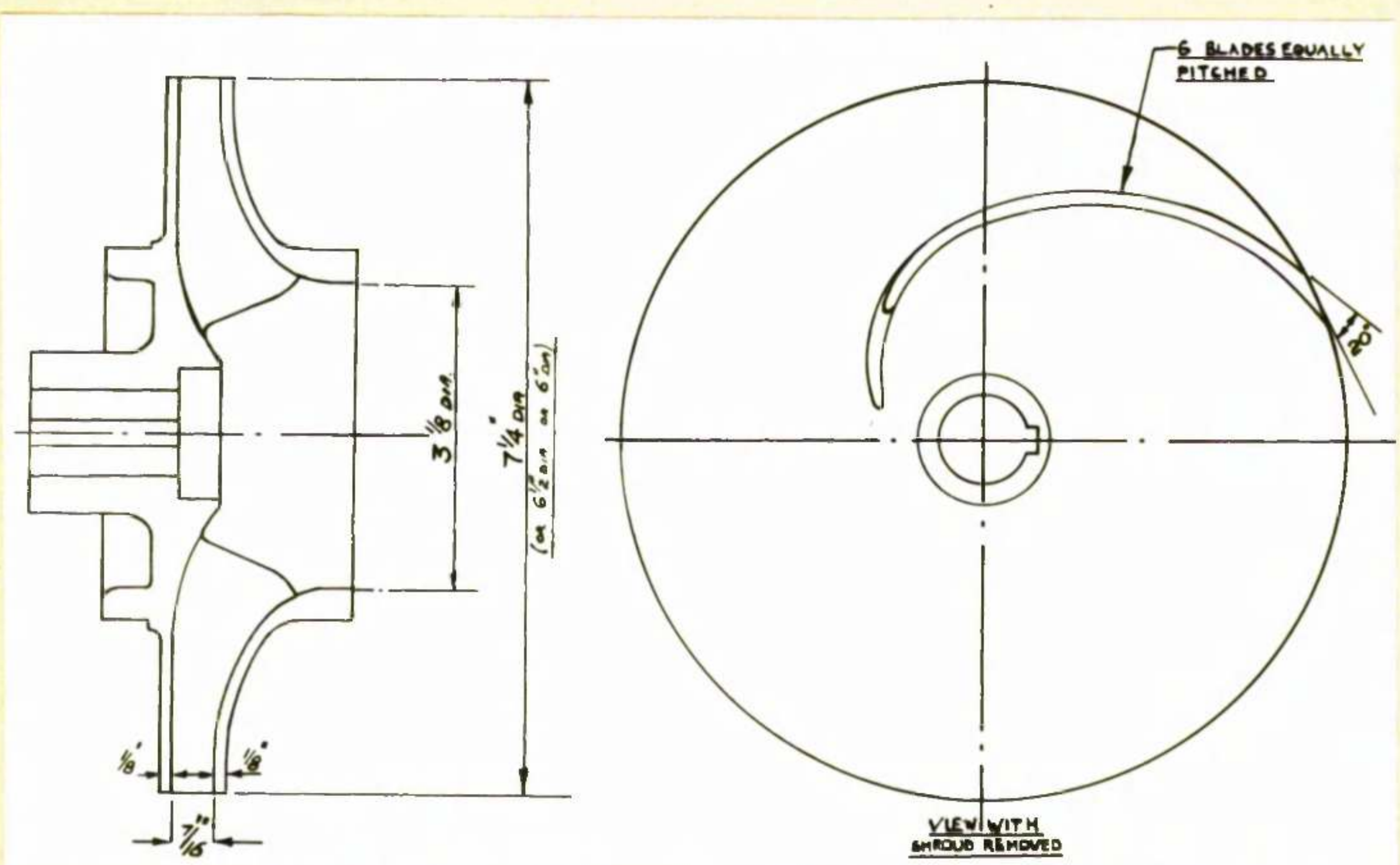


Sectional arrang
of U-11

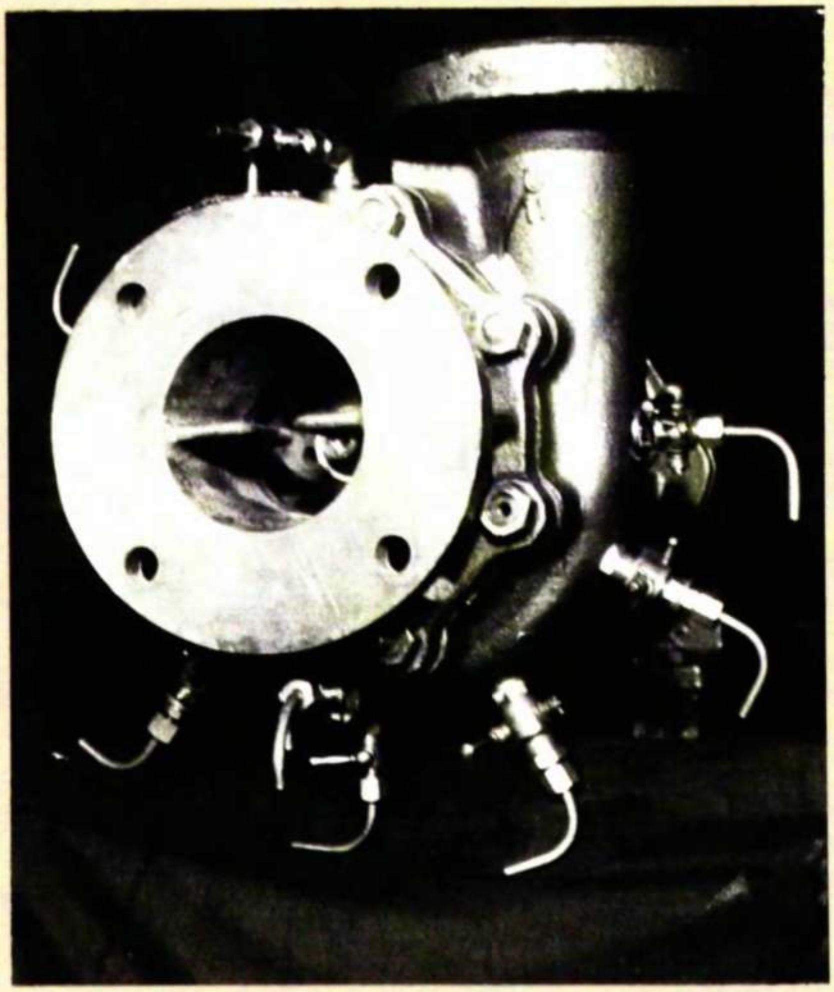
Fig.
c

U-11 impeller and volute details



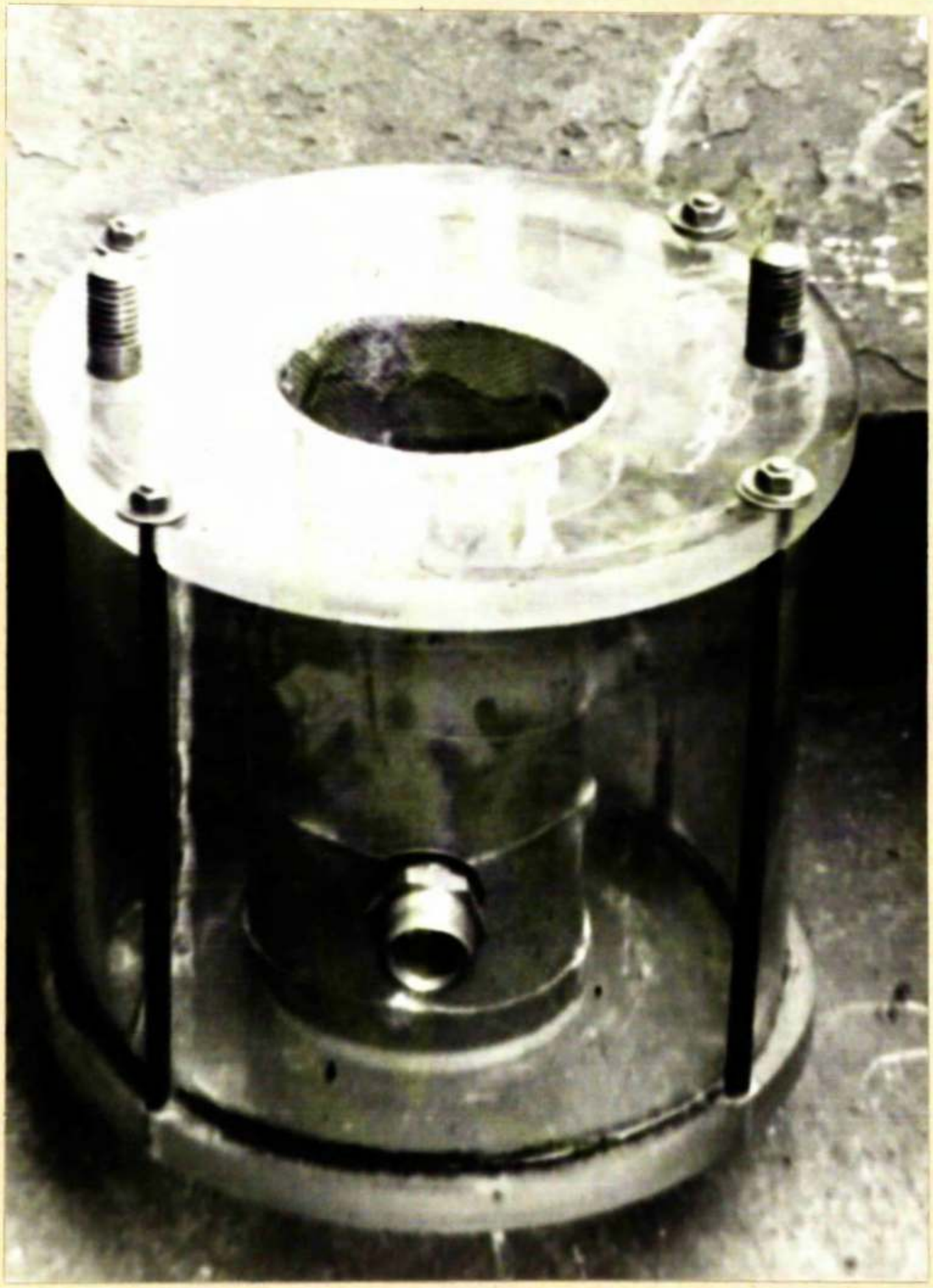


Fig



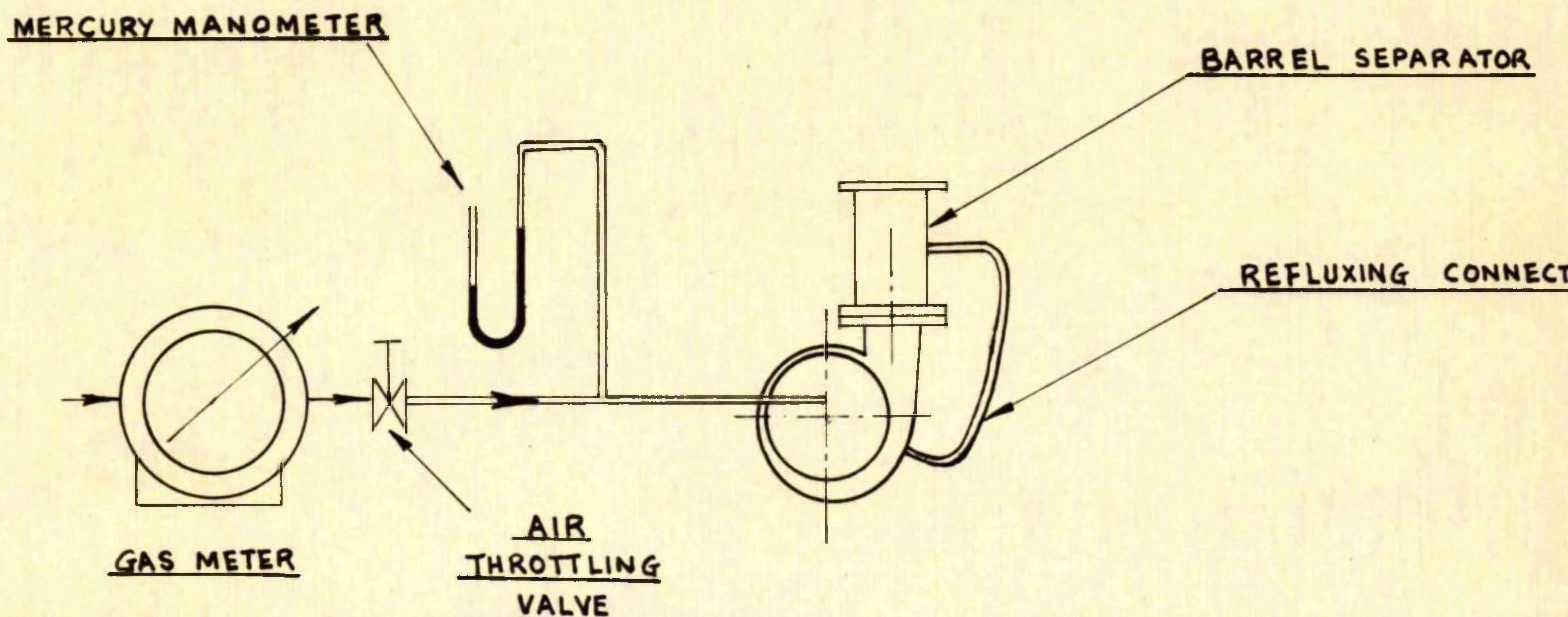
Test pump U-11 with standard end cover and casing pressure tappings in place.

Fig.



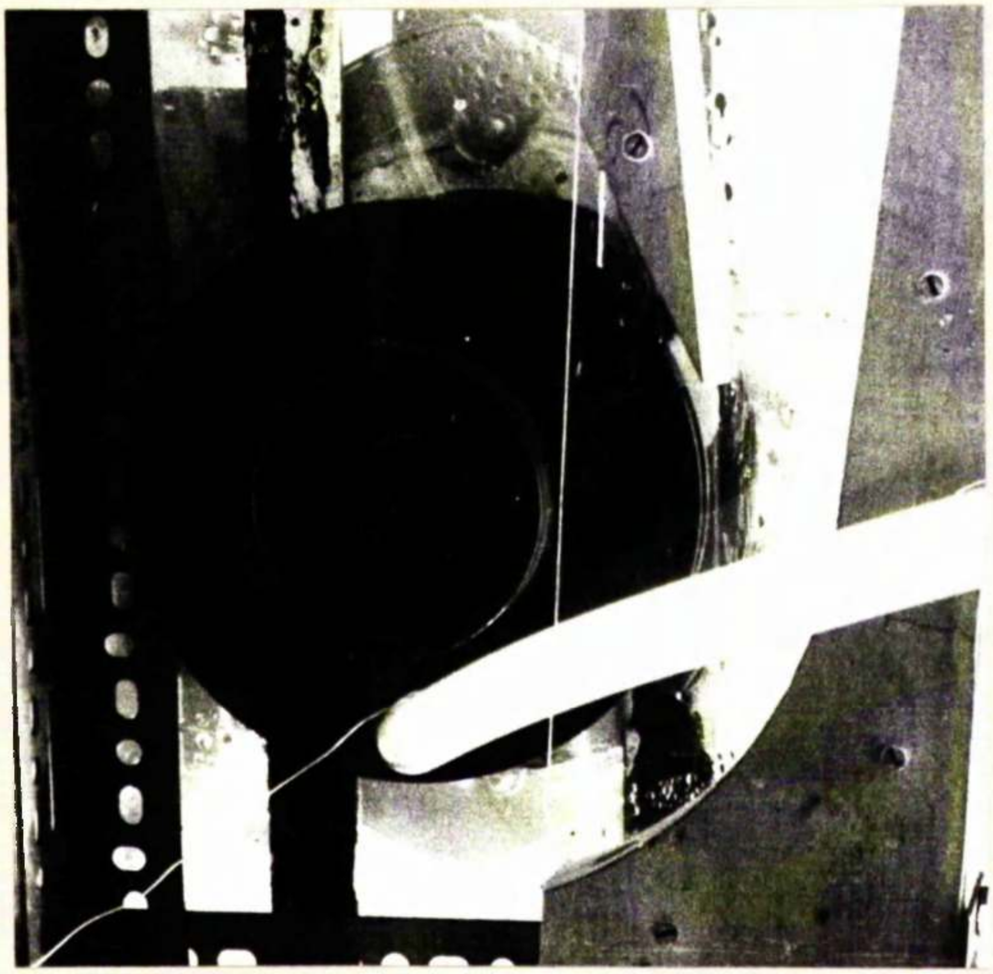
Barrel Separator

Fig.



DIAGRAMMATIC LAYOUT OF
INITIAL TEST APPARATUS

Fig



Jet and shroud interaction
with tapered collector

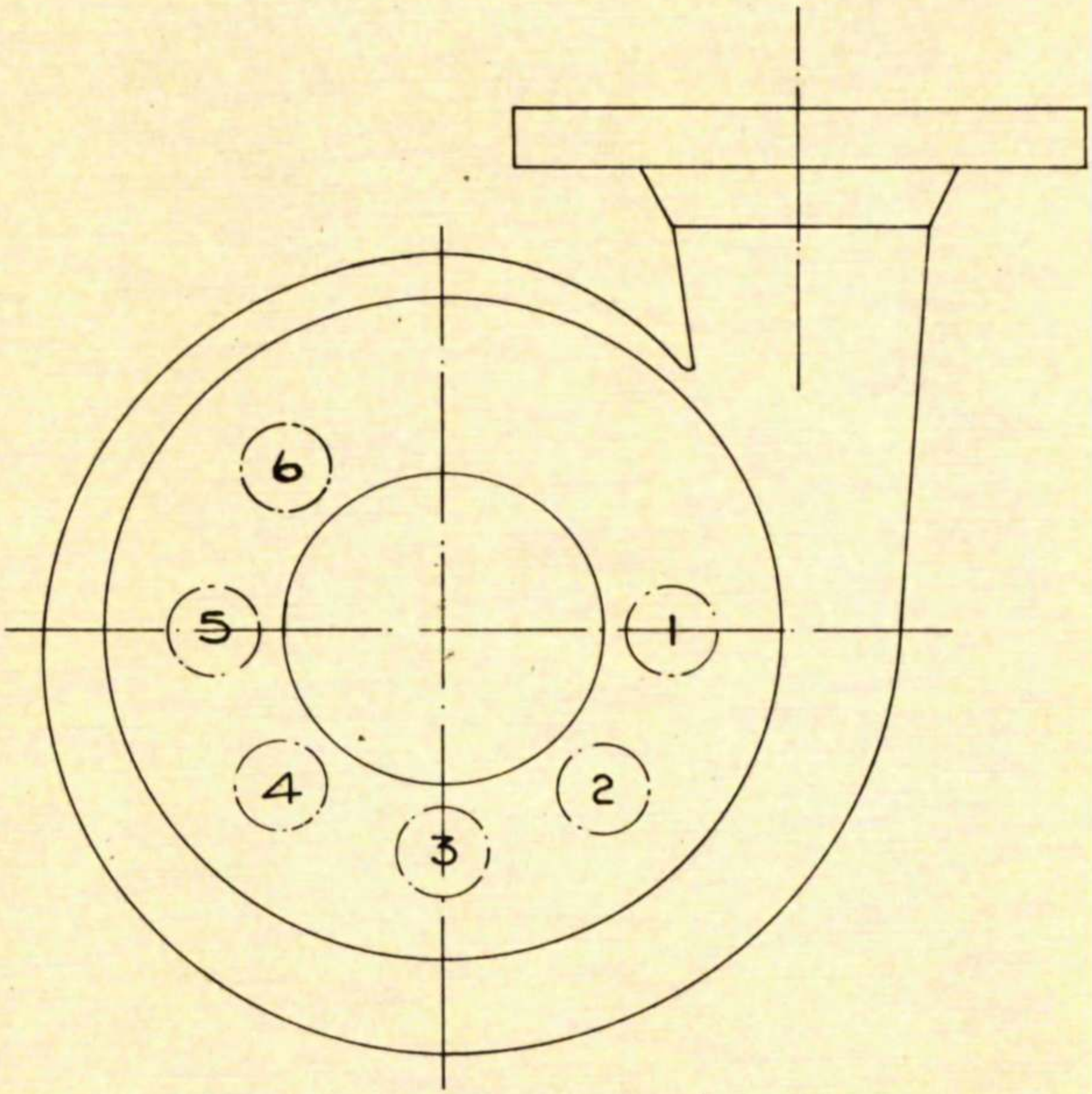
Fig. 1

Jet and shroud interaction
jet speed 2·6 ft/sec
tip speed 95 ft/sec
free impeller

Fig

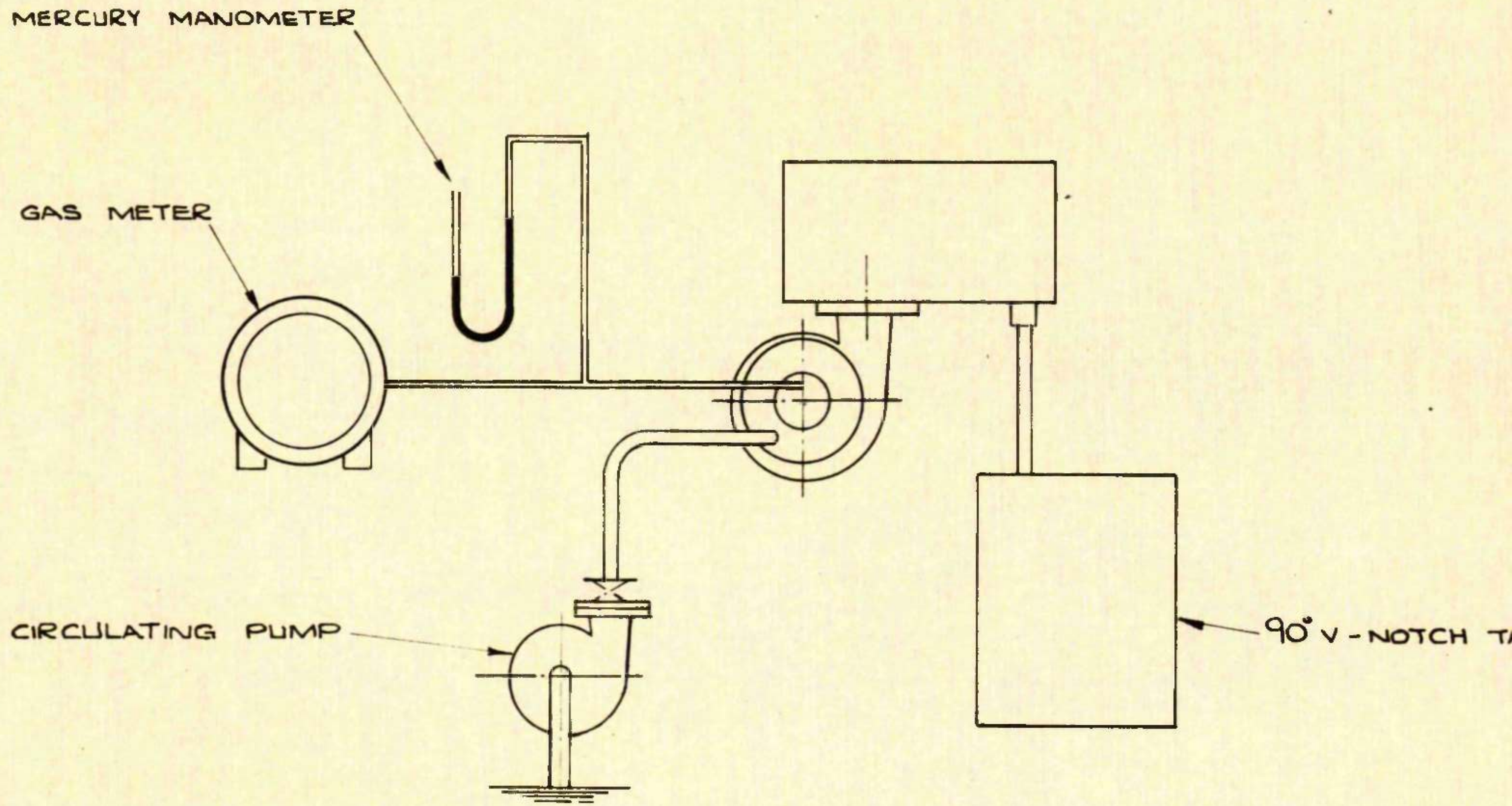
Jet and shroud interaction
jet speed 5·2 ft/sec
tip speed 95 ft/sec
free impeller

Fig



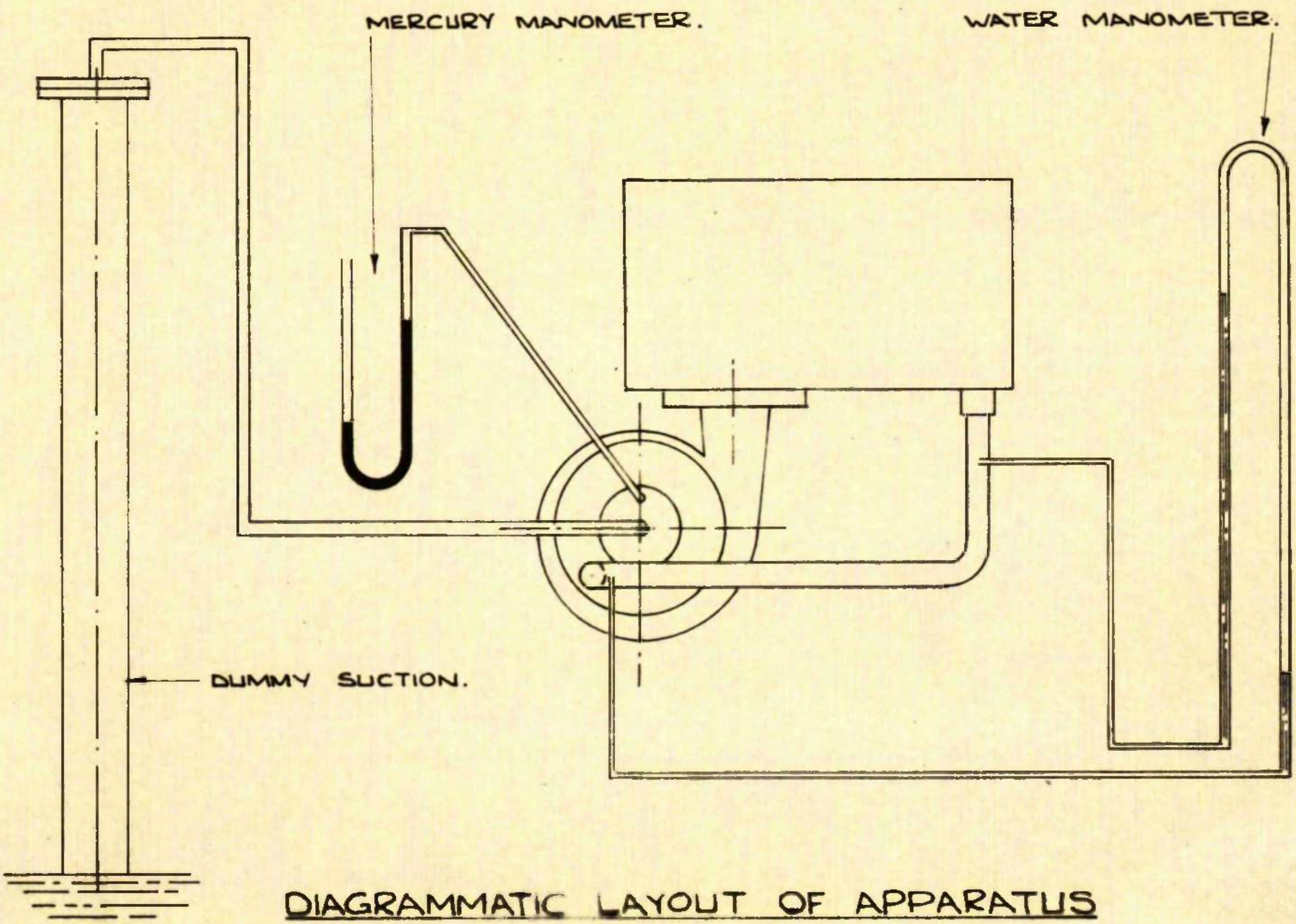
POSITION OF PUMP CIRCULATING CONNECTIONS.

Fig.



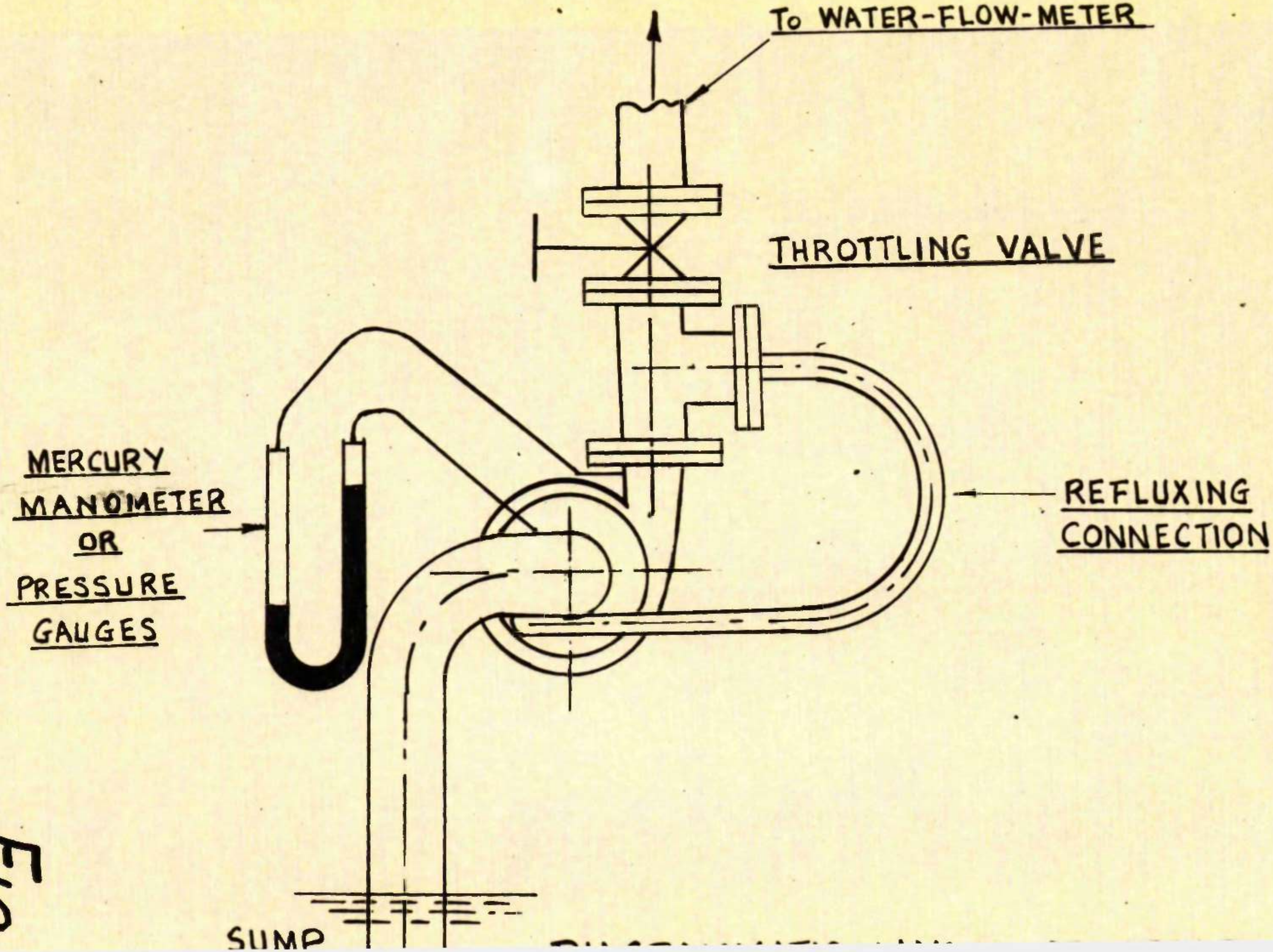
DIAGRAMMATIC LAYOUT OF APPARATUS FOR

Fig.



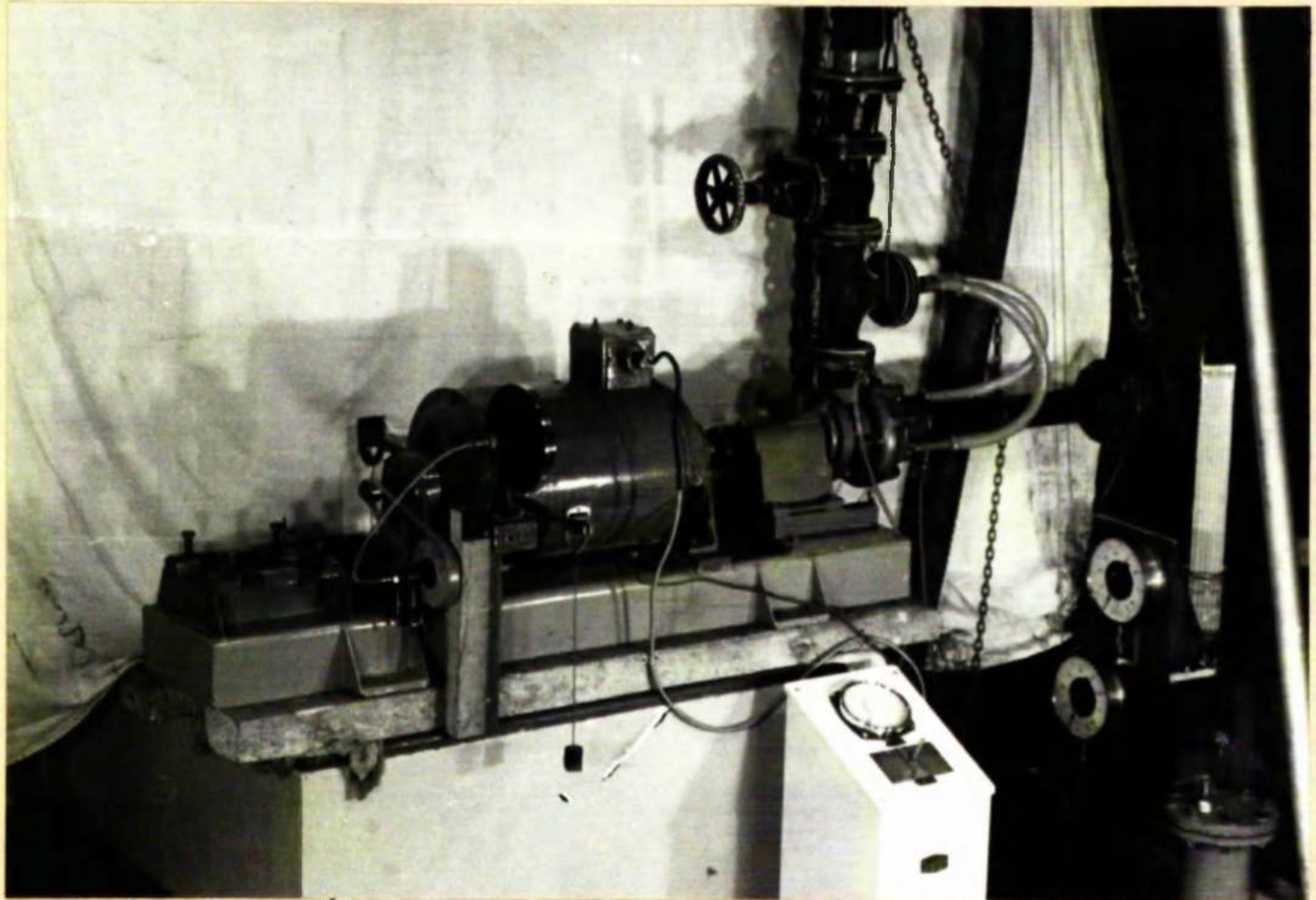
DIAGRAMMATIC LAYOUT OF APPARATUS

Fig.

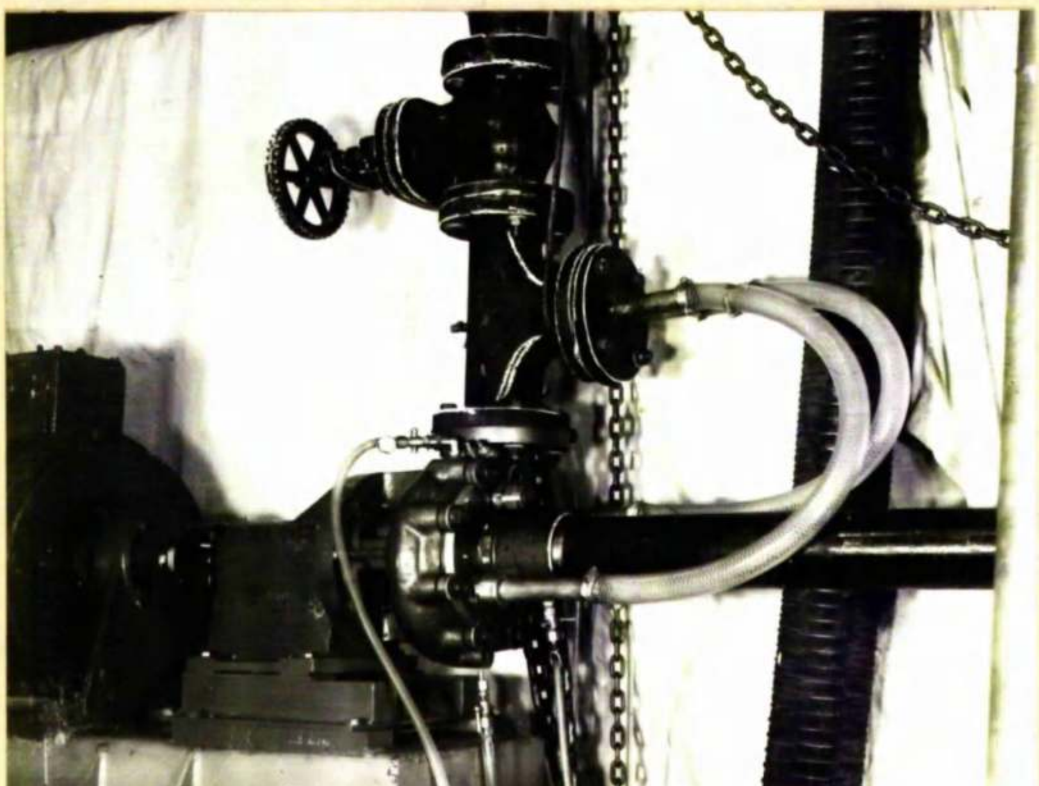


To WATER-FLOW-METER
THROTTLING VALVE
REFLUXING CONNECTION
MERCURY MANOMETER OR PRESSURE GAUGES
SUMP

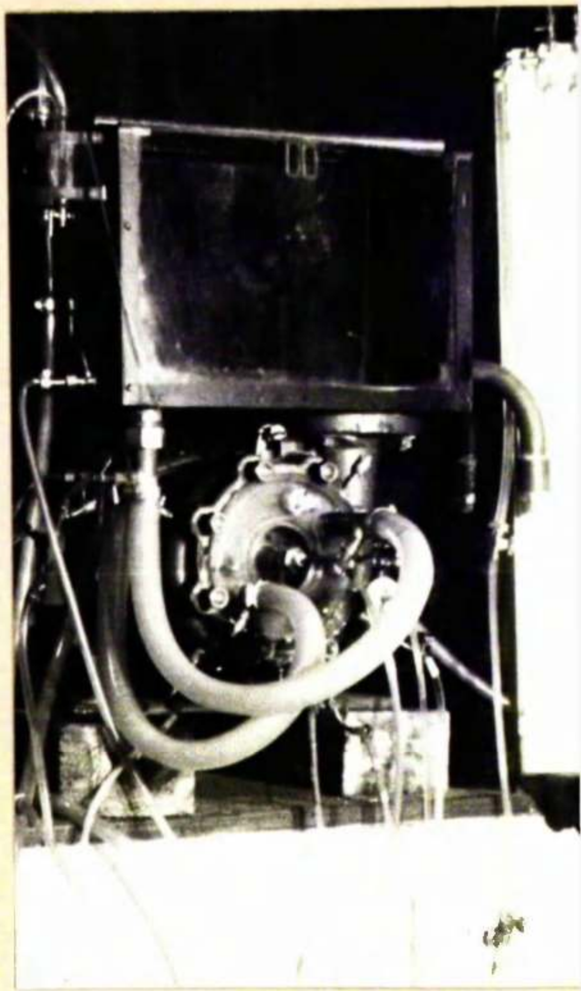
Fig.

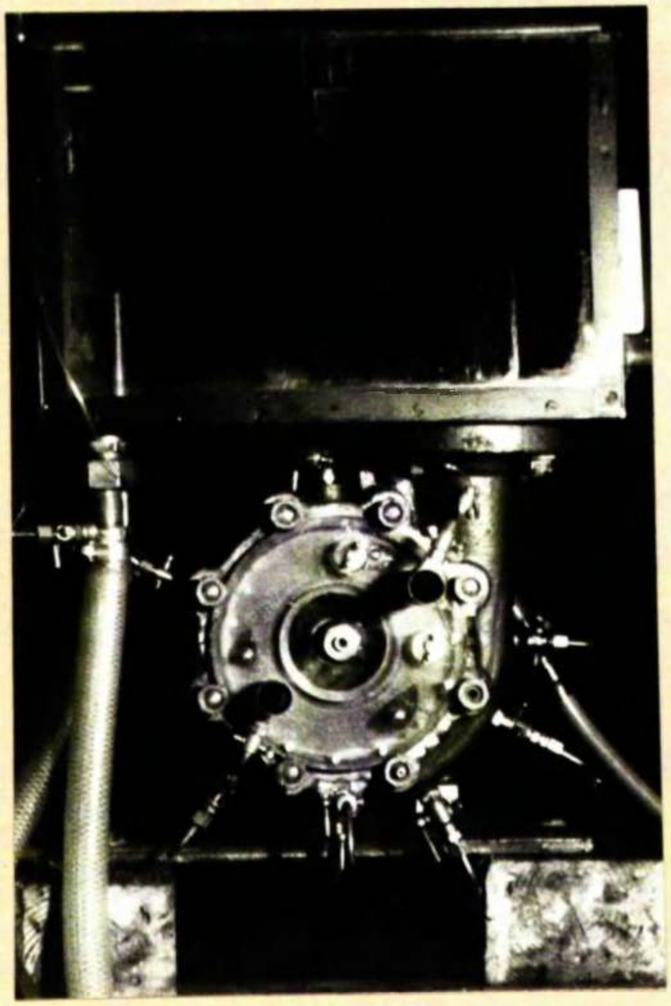


Development rig, arranged for water pump tes



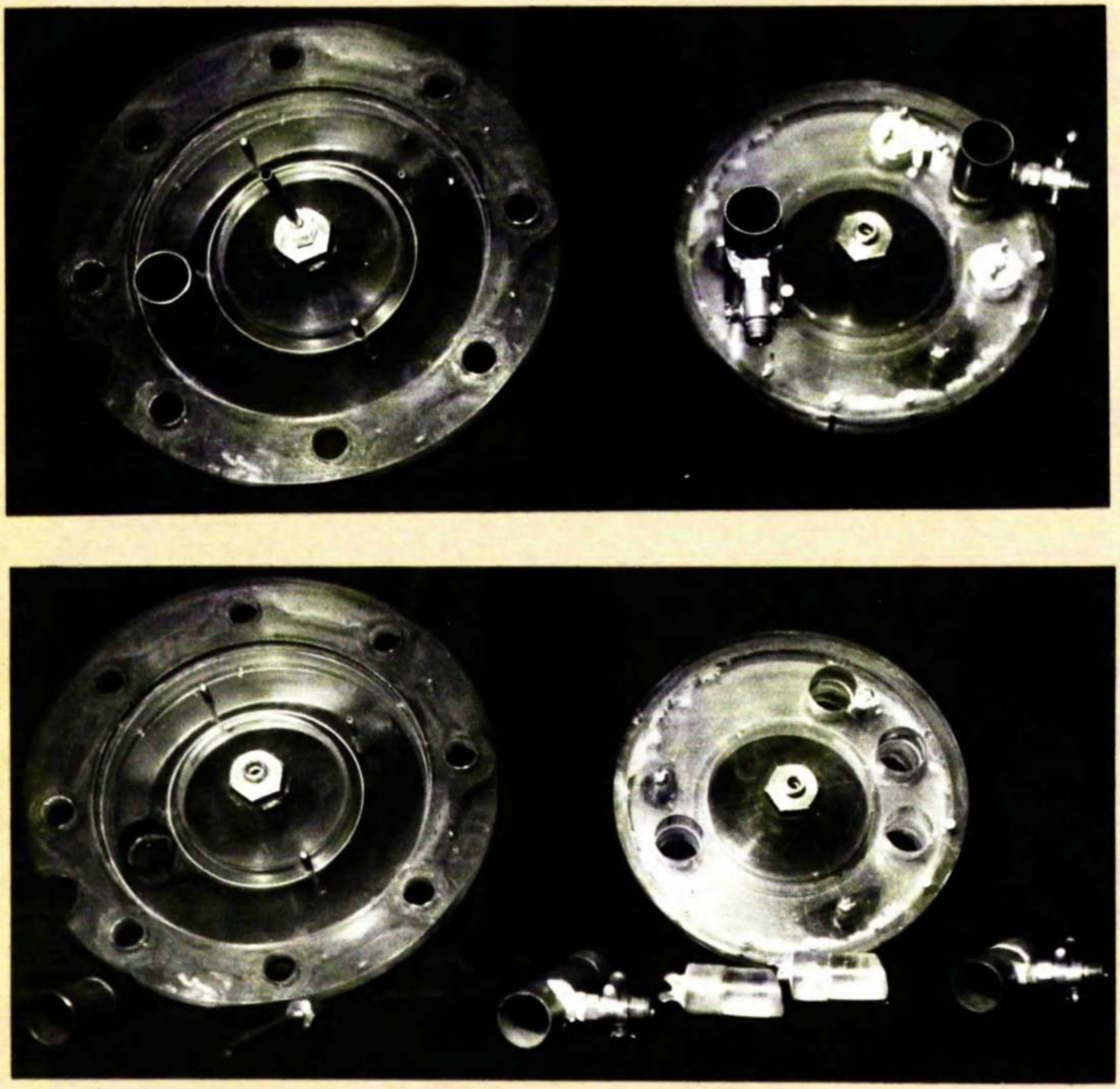
Fig





Development rig, arrang
for evacuation test.

Fig.



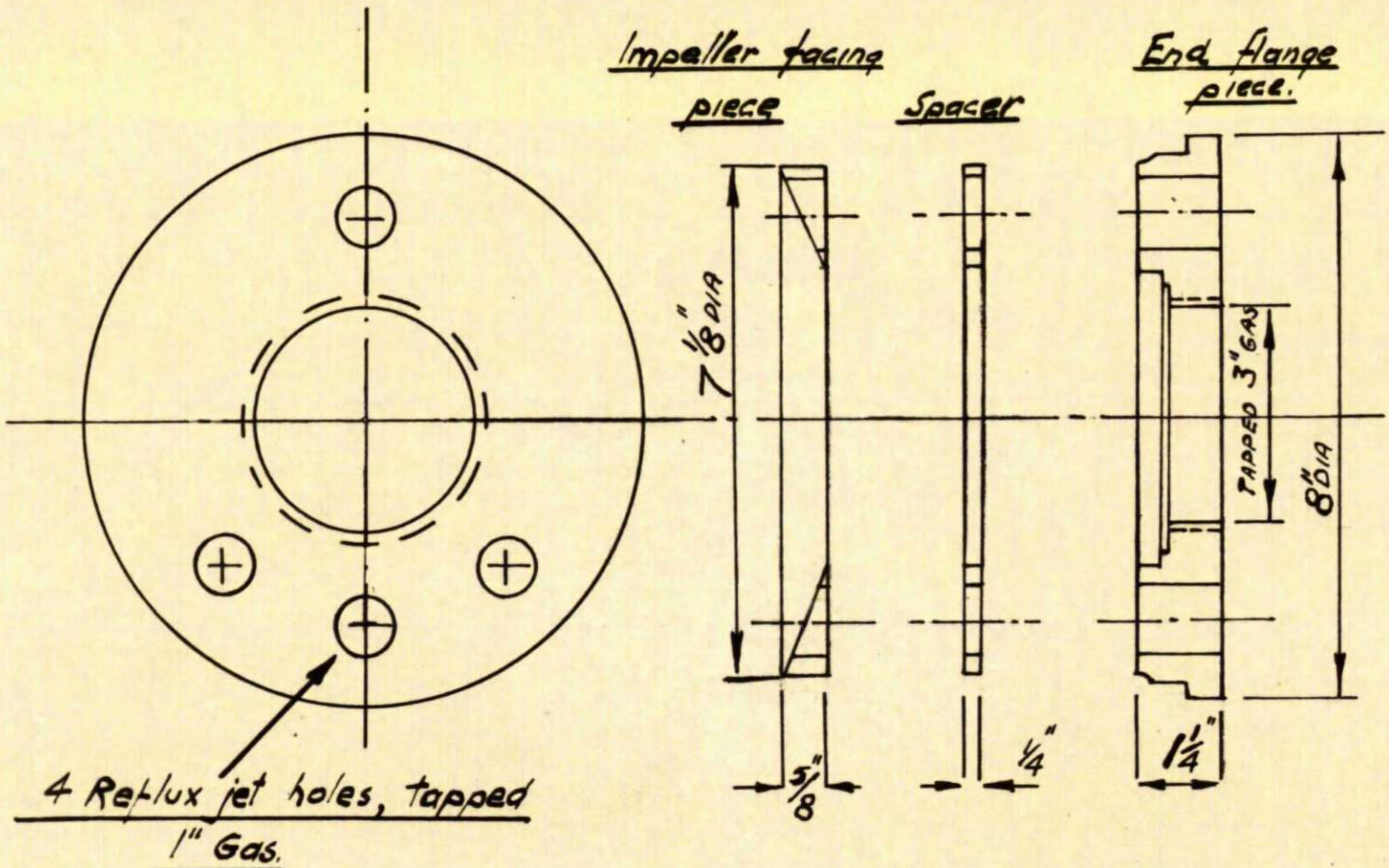
Perspex re-circulation

end covers

and

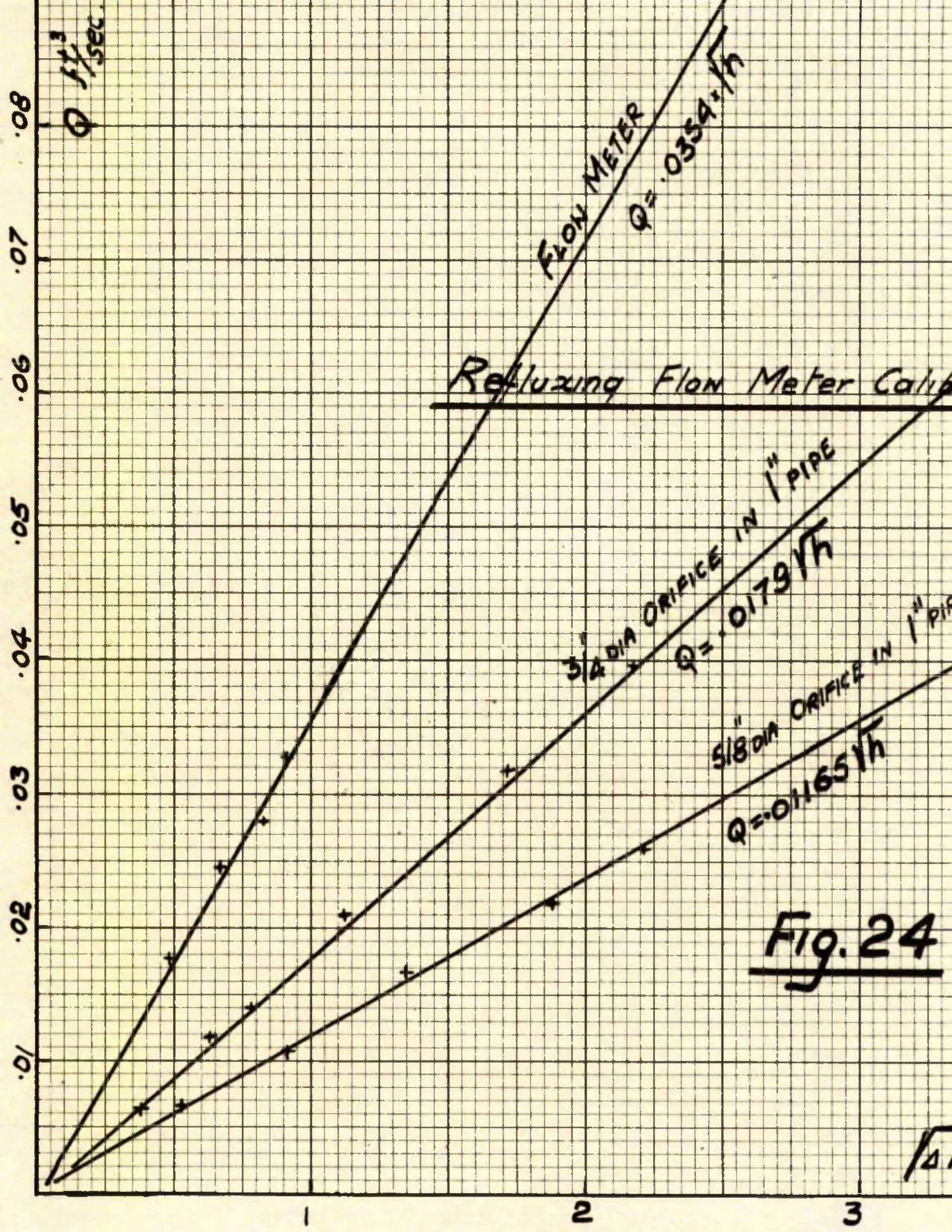
fittings

Fig.



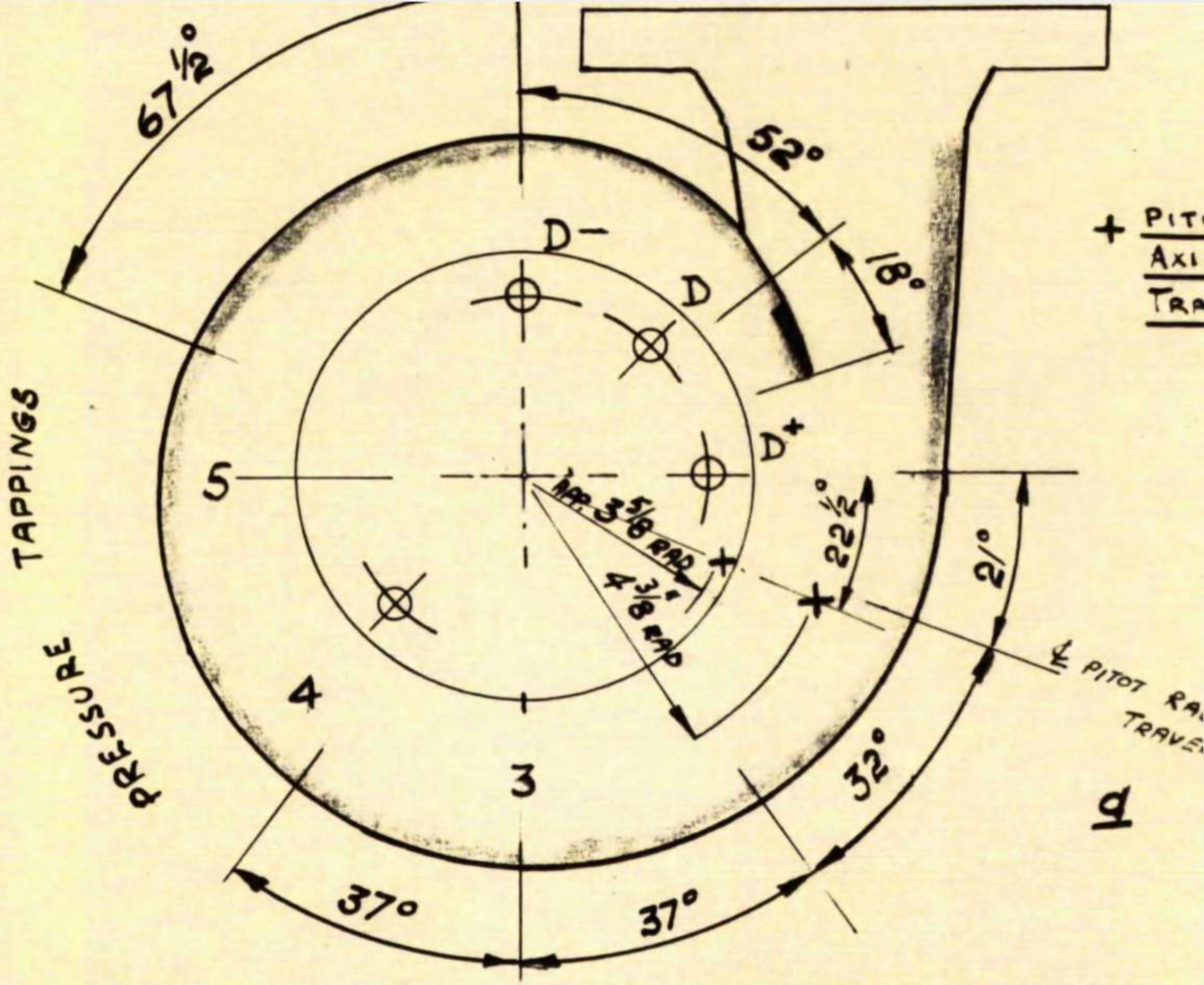
STEEL End Cover Details

Fig. 6



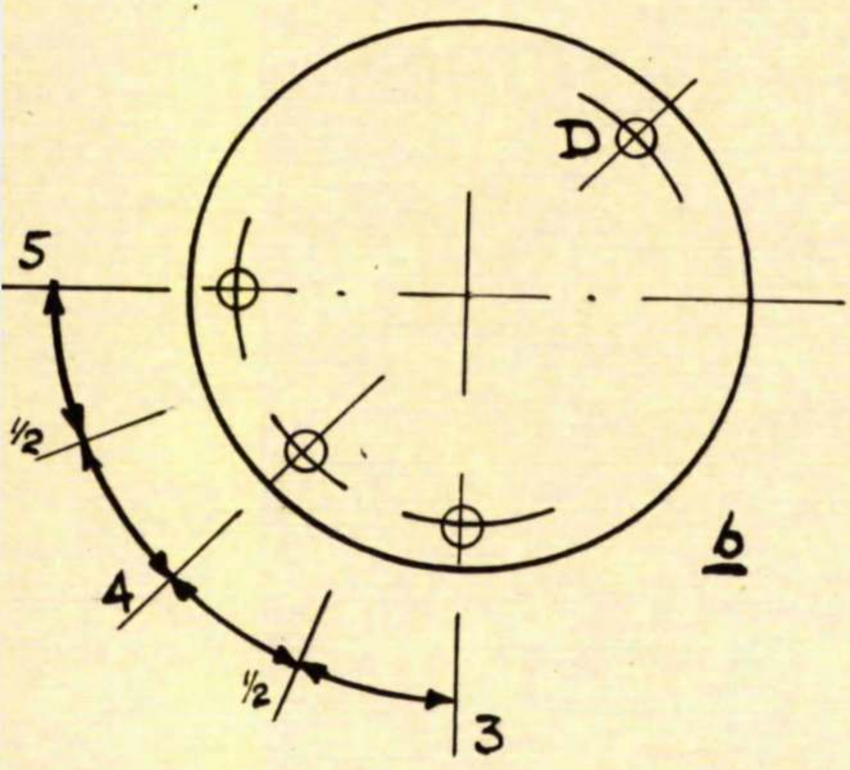
Q ft³/sec
.08
.07
.06
.05
.04
.03
.02
.01
Flow Meter
Q = .0354√h
Refluxing Flow Meter Calib
3/4" dia orifice in 1" pipe
Q = .0179√h
5/8" dia orifice in 1" pi
Q = .01165√h
1
2
3

Fig. 24



Orientation of pressure t

reflux-jets and Pitot



Fig

a

Definition of jet positioning.

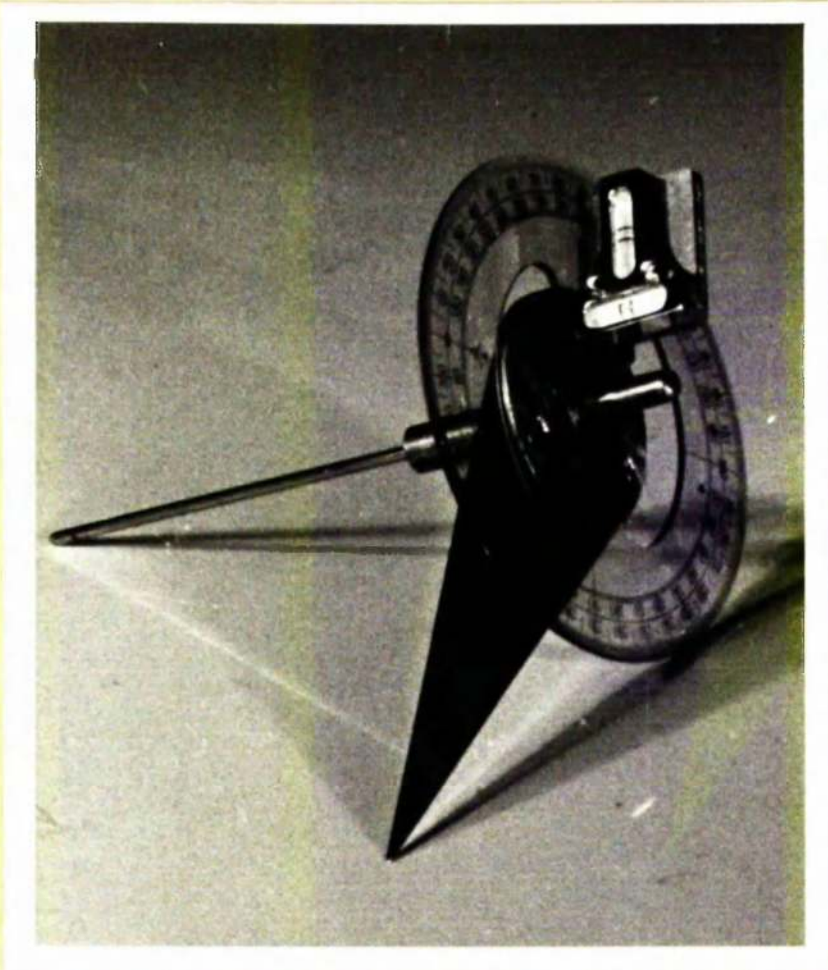
A numeral or <u>D</u> indicate a single jet only as shown
fig 25.b.

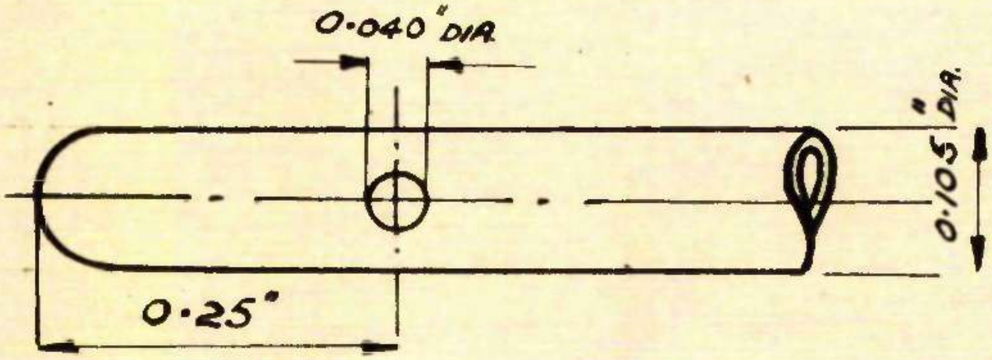
180° or D proceeded by a numeral indicate the blocki
jet diametrally opposite the main jet denoted by
the numeral.

D^+ and D^- indicate the blocking jet displaced cl
or anticlockwise from diametral position.

× If the <u>symbol</u> D preceeds the numeral, the bloc
jet is fixed in position D shown in fig. 25b. The
jet is then in a position indicated by the numer
i.e. D-3, D-4, D-5.

Fig.

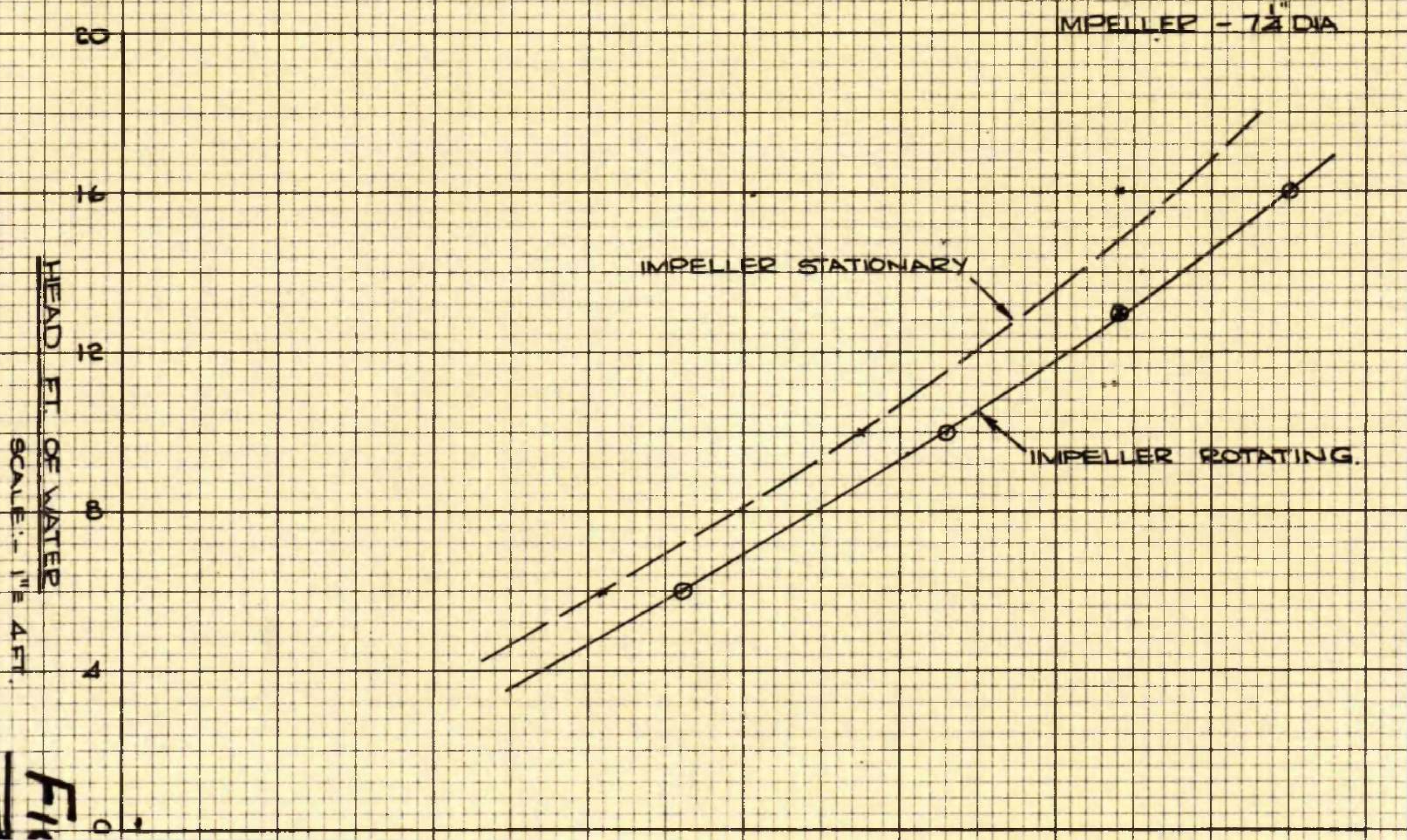


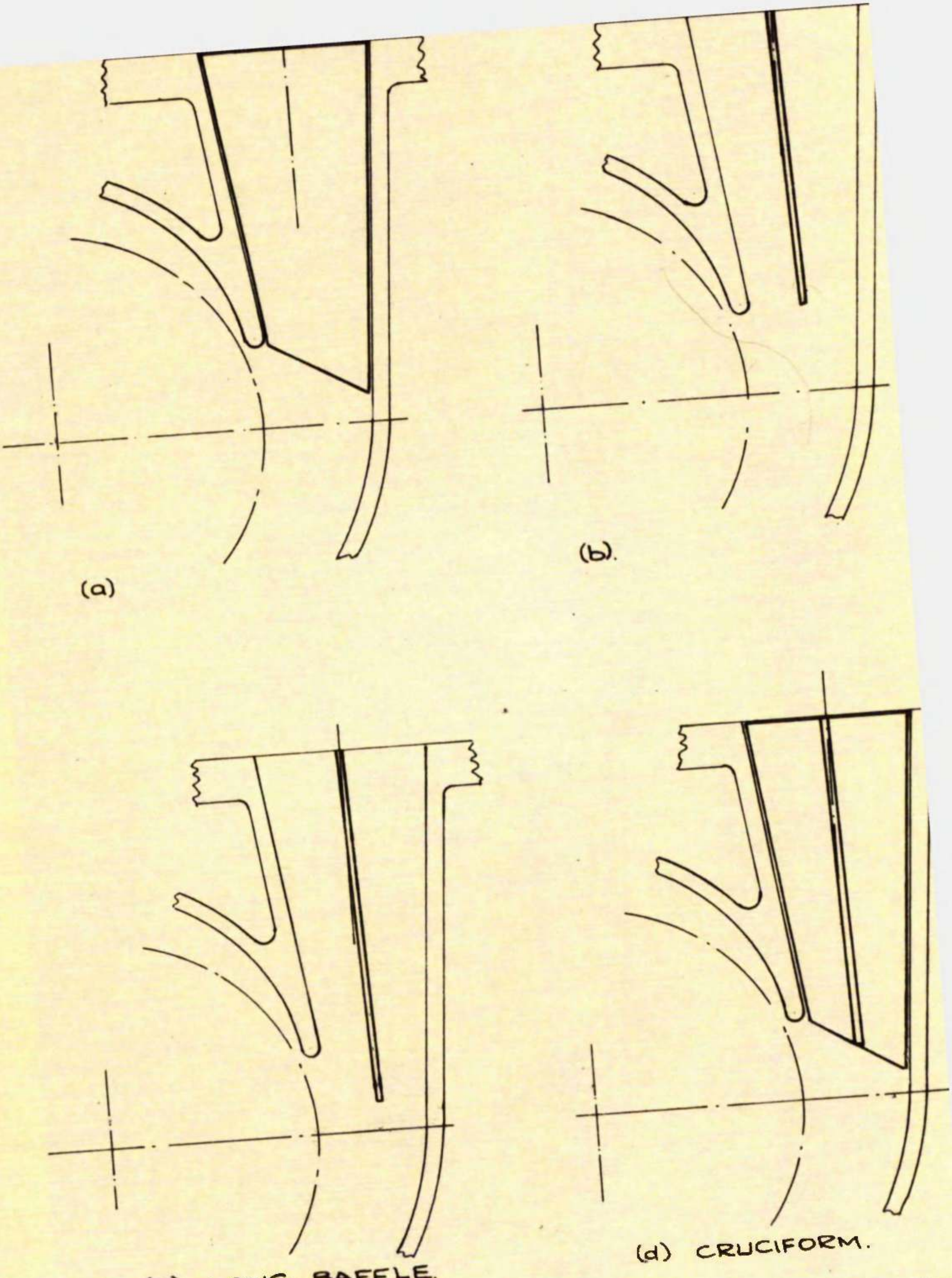


Pitot Cylinder
and
Protractor

Fig. 2

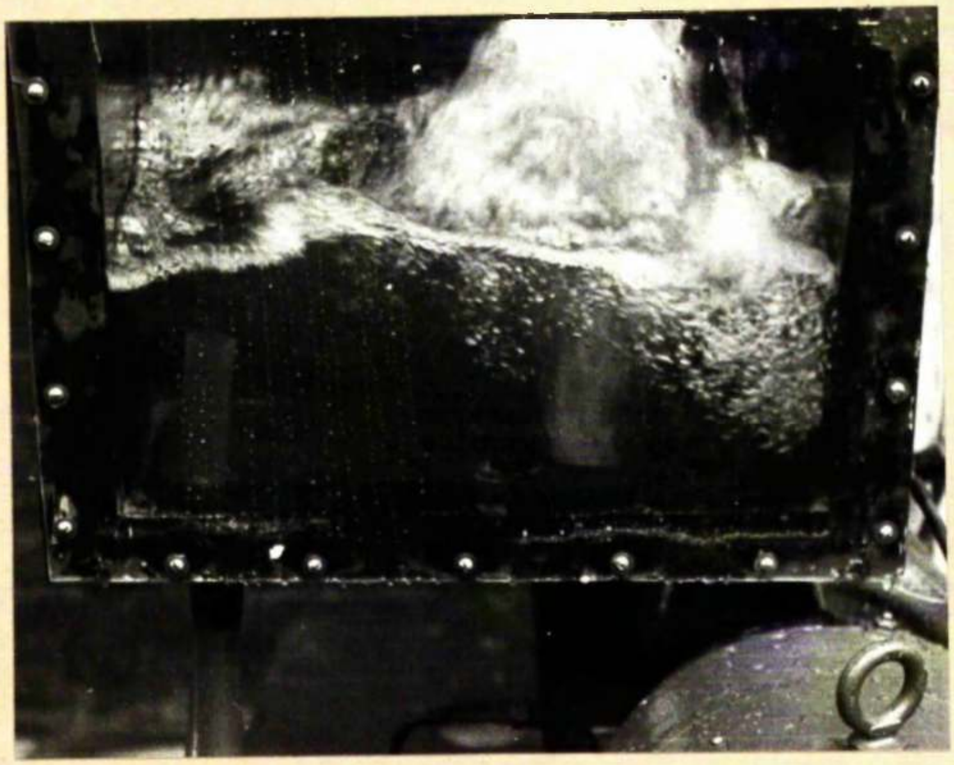
GRAPH OF SUPPLY HEAD AGAINST QUANTITY





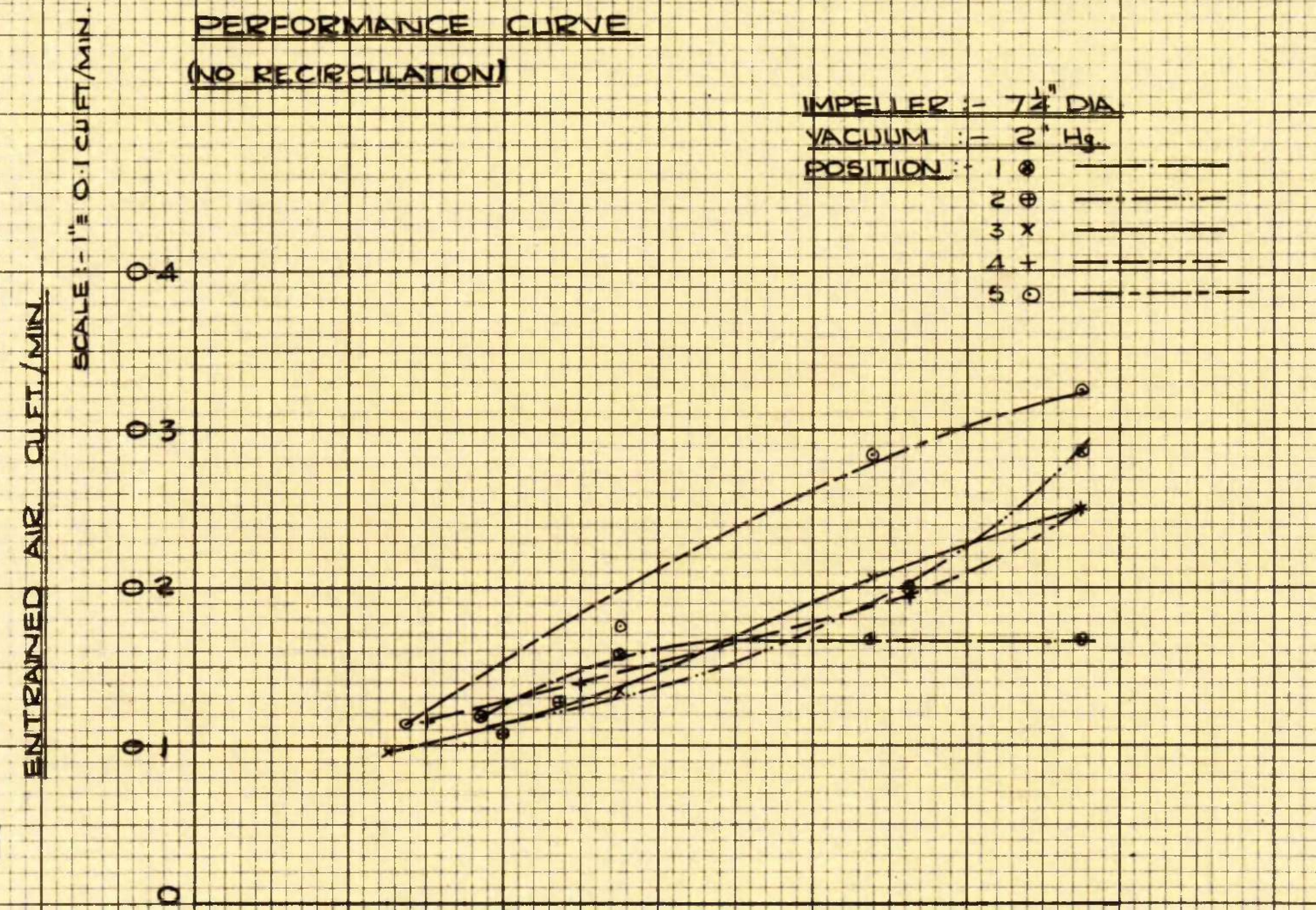
TYPICAL BAFFLES.

Fi



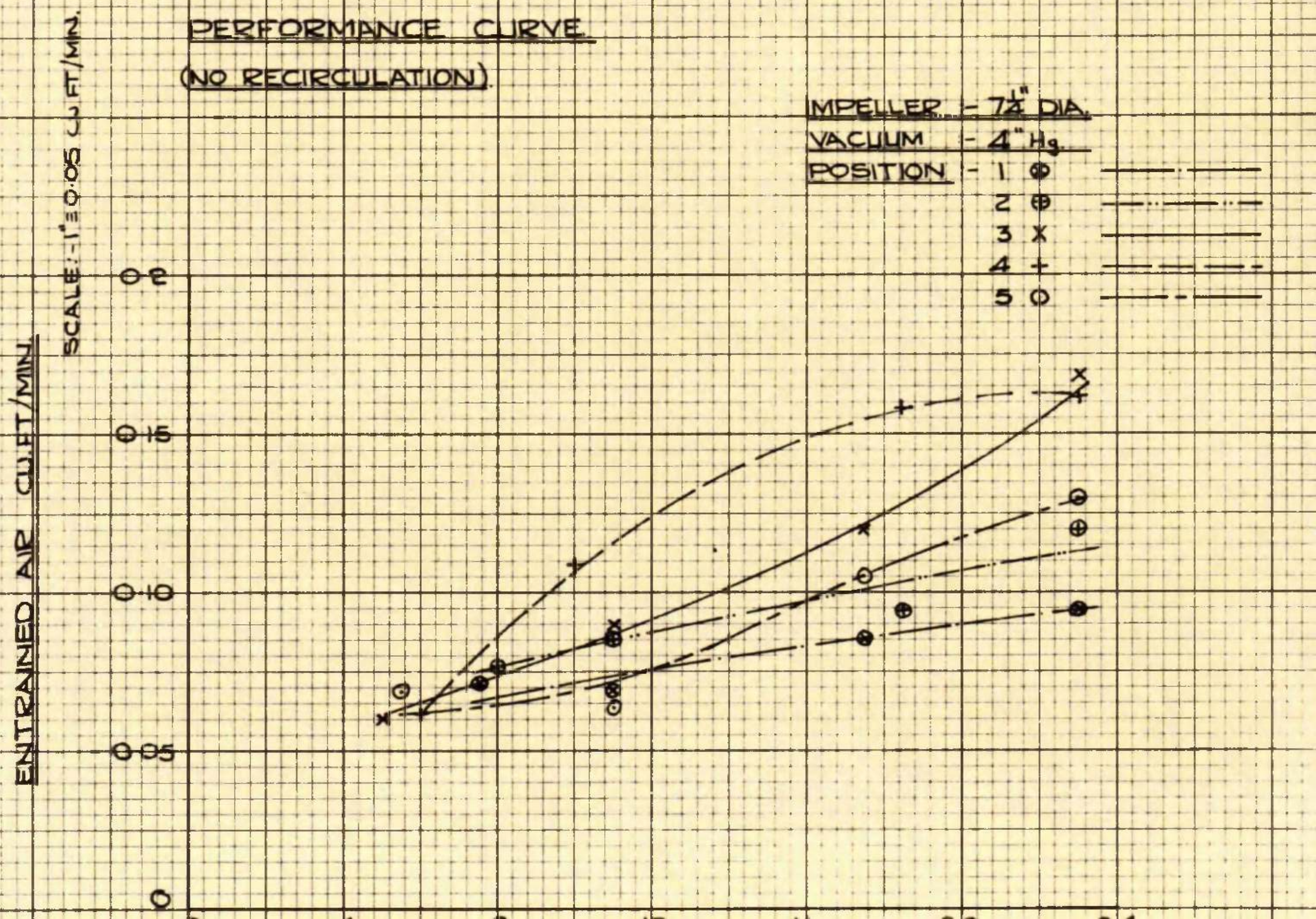
Feasibility study box separator operating at 4" Hg vacuum suction.

Fig.



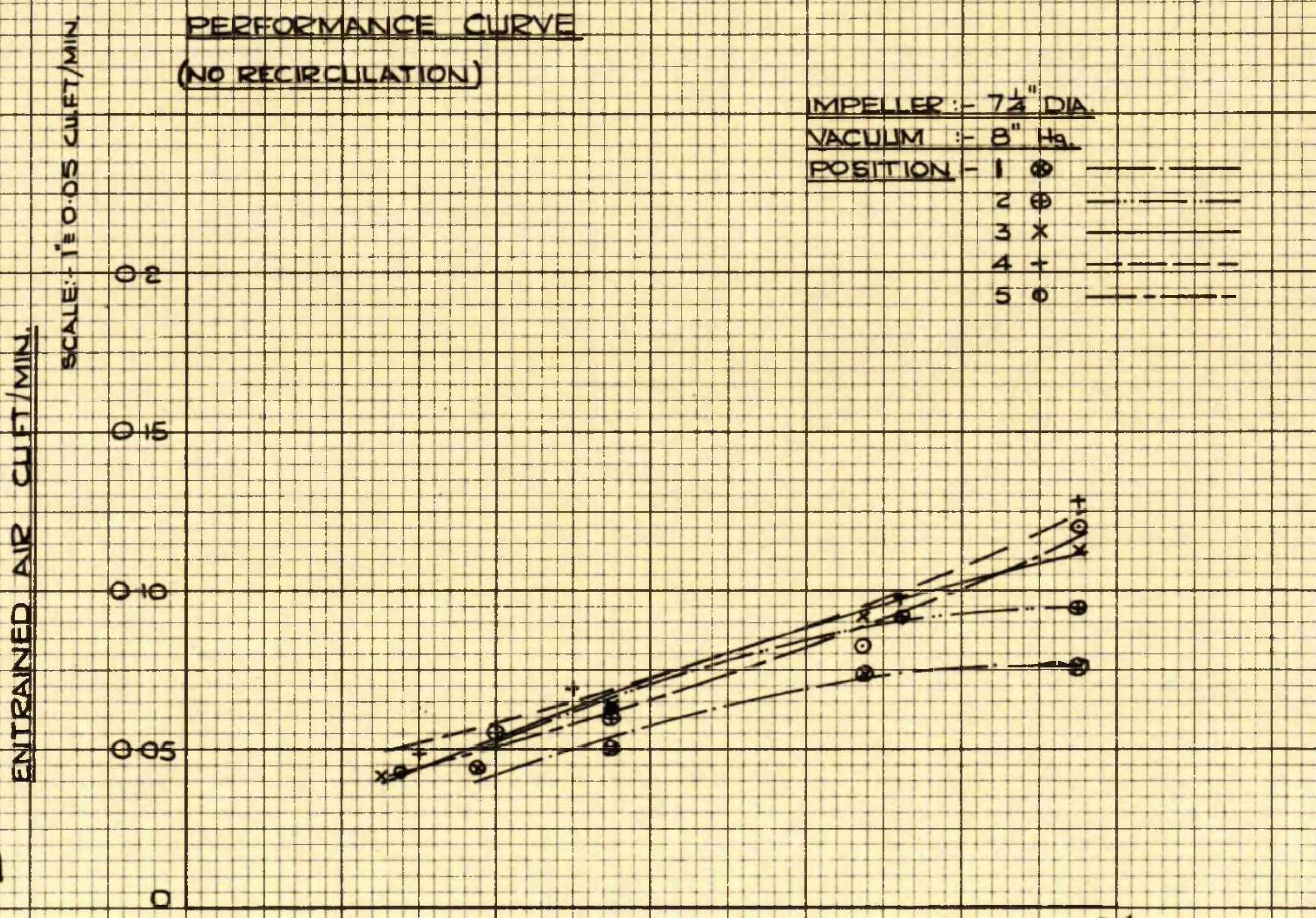
PERFORMANCE CURVE
(NO RECIRCULATION)
IMPELLER :- 7¼" DIA.
VACUUM :- 2" Hg.
POSITION :- 1 ⊗
2 ⊕
3 x
4 +
5 ⊙
ENTRAINED AIR CU.FT./MIN.
SCALE :- 1" = 0·1 CU.FT./MIN.
0·4
0·3
0·2
0·1
0

Fig.



PERFORMANCE CURVE
(NO RECIRCULATION)
IMPELLER - 7¼" DIA.
VACUUM - 4" Hg.
POSITION - 1
2
3 X
4 +
5
ENTRAINED AIR CU.FT/MIN
SCALE:-1" = 0.05 CU.FT/MIN.
0.2
0.15
0.10
0.05
0

Fig



PERFORMANCE CURVE
(NO RECIRCULATION)
IMPELLER :- 7¼" DIA.
VACUUM :- 8" Hg.
POSITION - 1 ⊗
2 ⊕
3 ×
4 +
5 ⊙
ENTRAINED AIR CU.FT/MIN.
SCALE:- 1" = 0·05 CU.FT/MIN.
0·2
0·15
0·10
0·05
0

Fig

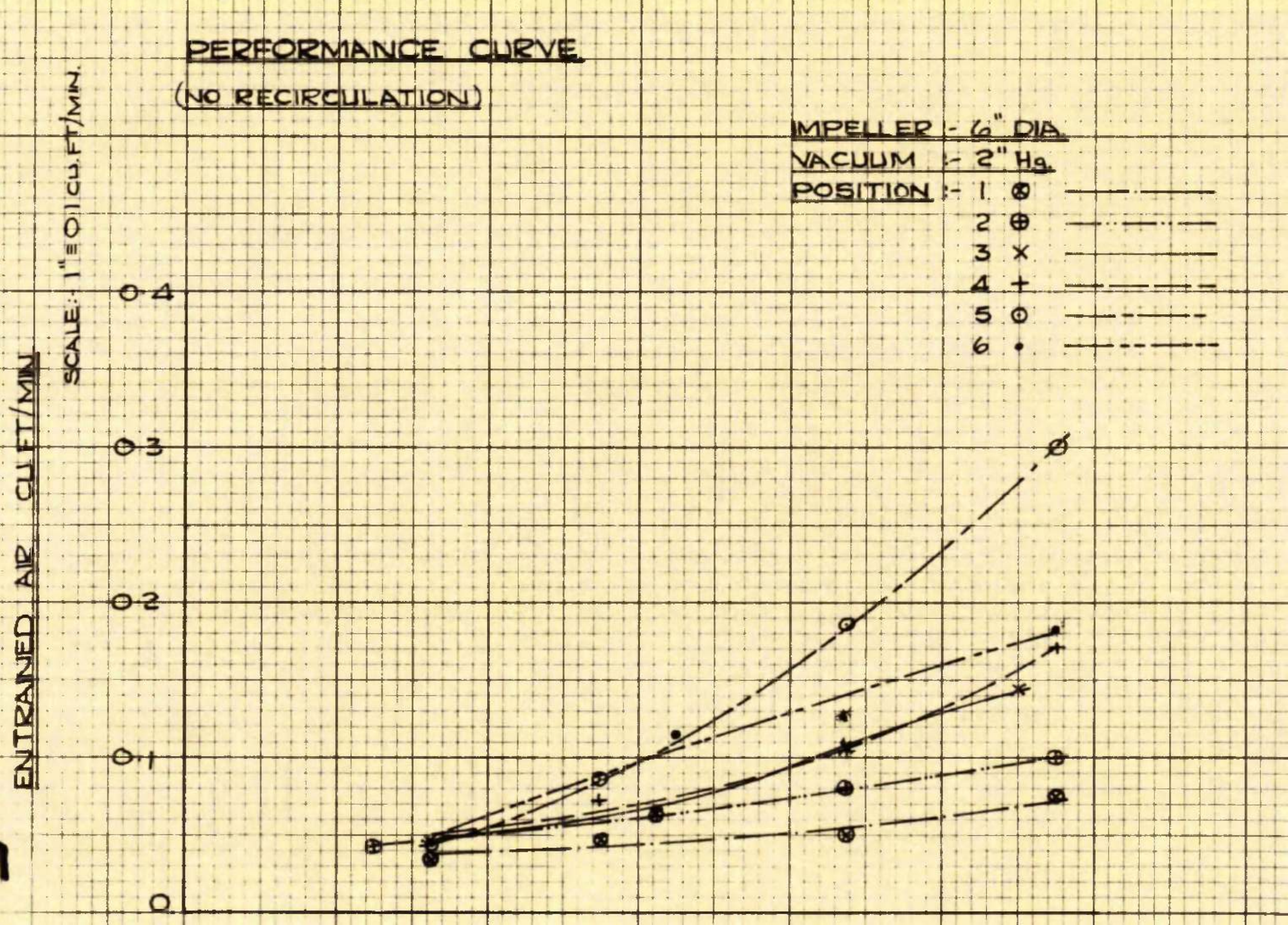
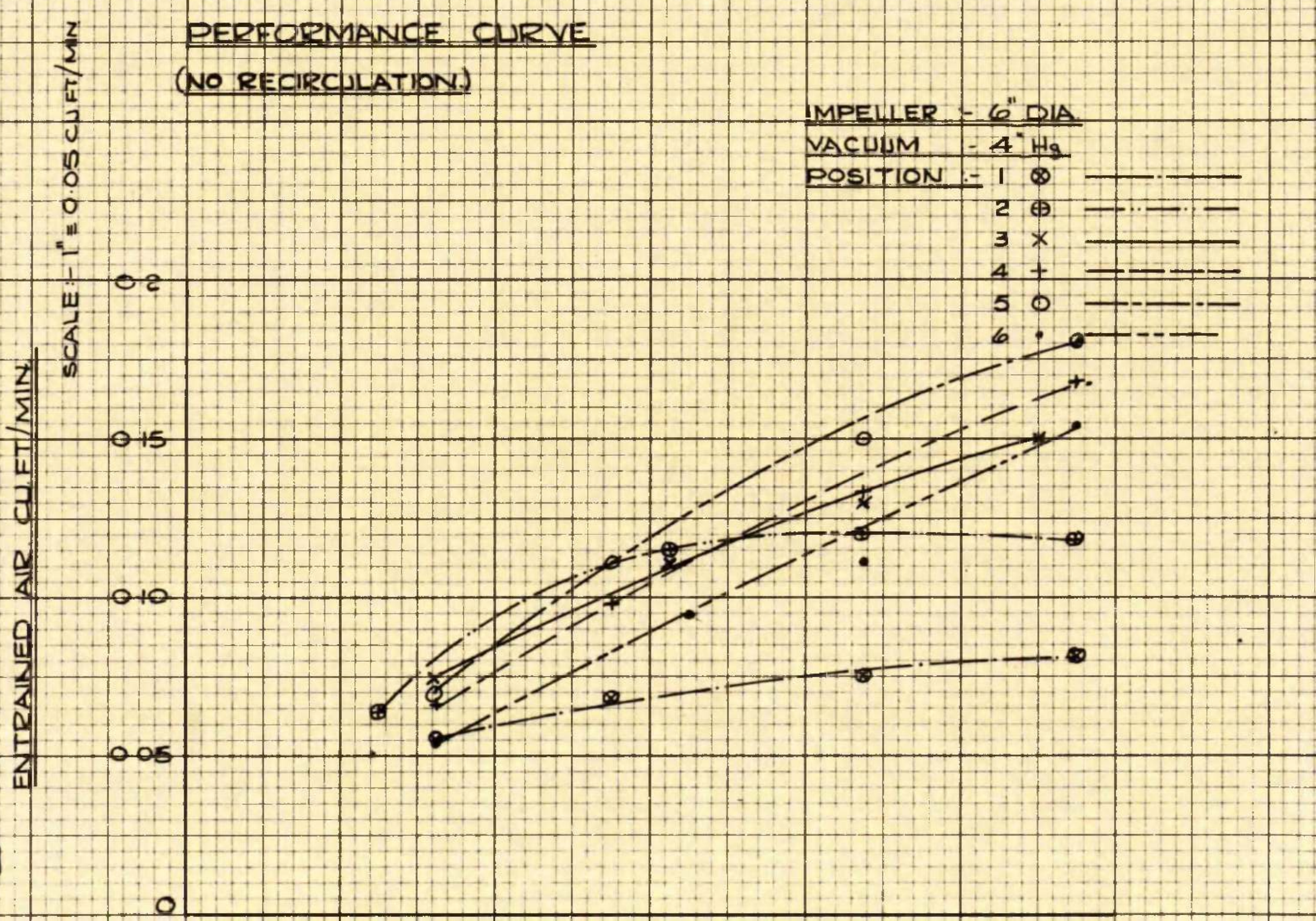
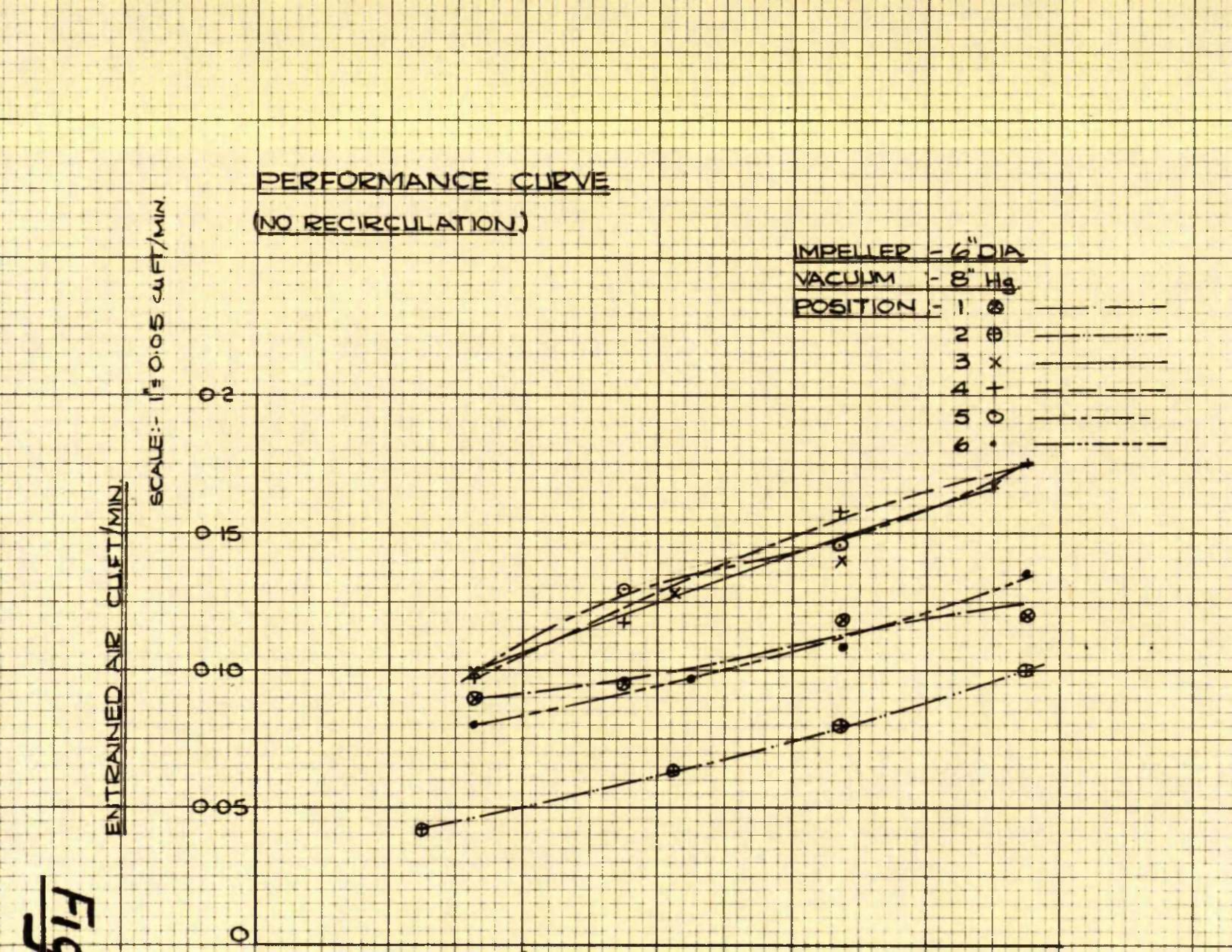
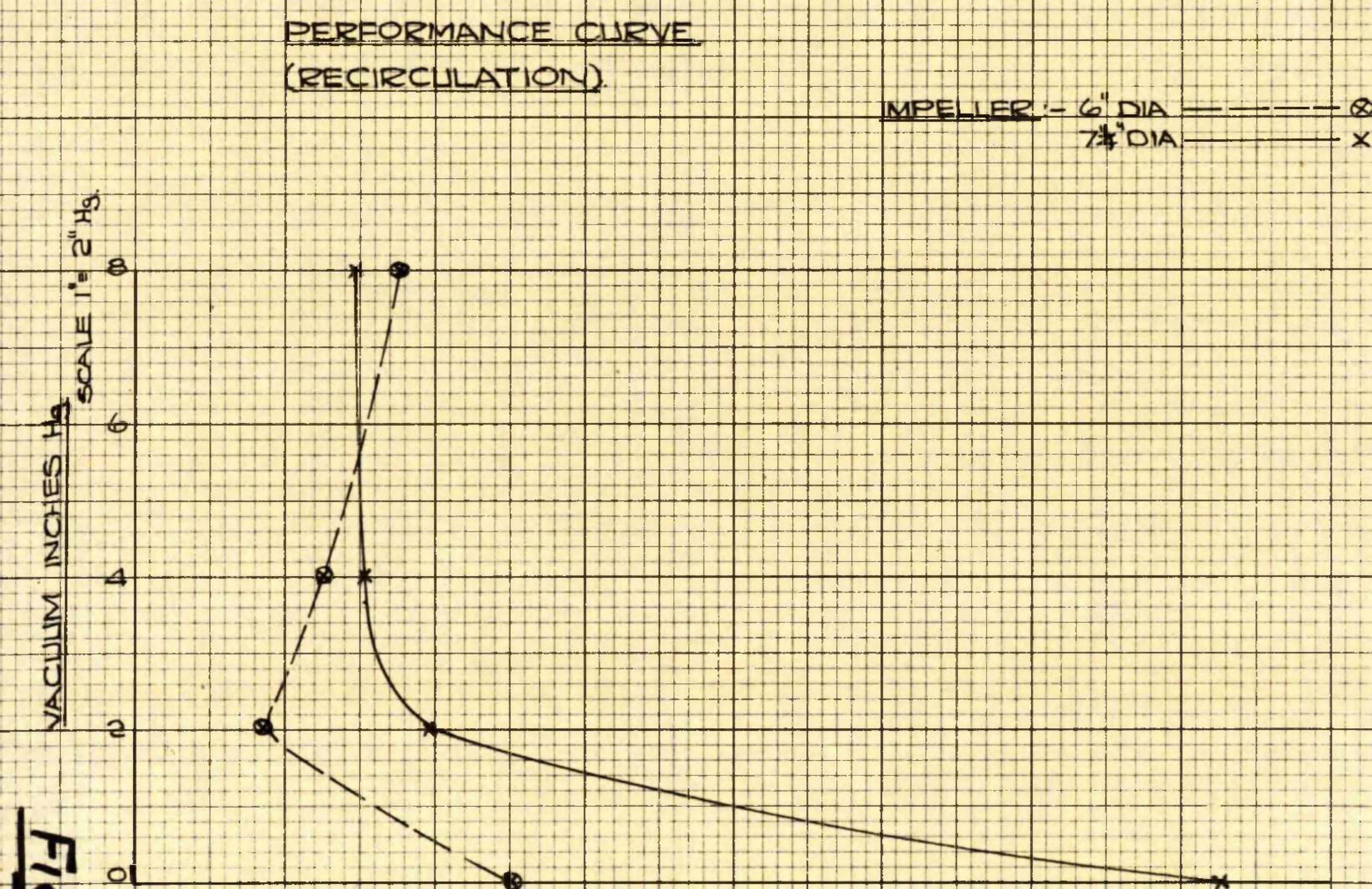

PERFORMANCE CURVE
(NO RECIRCULATION)
IMPELLER :- 6" DIA.
VACUUM :- 2" Hg.
POSITION :- 1 ⊗
2 ⊕
3 ×
4 +
5 ⊙
6 •
ENTRAINED AIR CU.FT/MIN
SCALE :- 1" ≡ 0·1 CU.FT/MIN.
0·4
0·3
0·2
0·1
0

Fig.


PERFORMANCE CURVE
(NO RECIRCULATION.)
IMPELLER - 6" DIA
VACUUM - 4" Hg
POSITION :- 1 ⊗
2 ⊕
3 ×
4 +
5 ○
6 •
ENTRAINED AIR CU FT/MIN
SCALE :- 1" = 0·05 CU FT/MIN
0·2
0·15
0·10
0·05
0
Fig.


PERFORMANCE CURVE
(NO RECIRCULATION)
IMPELLER - 6" DIA
VACUUM - 8" Hg
POSITION - 1 ⊗
2 ⊕
3 ×
4 +
5 ⊙
6 •
ENTRAINED AIR CU.FT/MIN.
SCALE:- 1" = 0·05 cu.FT/MIN.
0·2
0·15
0·10
0·05
0
FIG



PERFORMANCE CURVE
(RECIRCULATION).
IMPELLER :- 6" DIA
7¼" DIA
VACUUM INCHES Hg
SCALE 1" = 2" Hg.
8
6
4
2
0

Fi

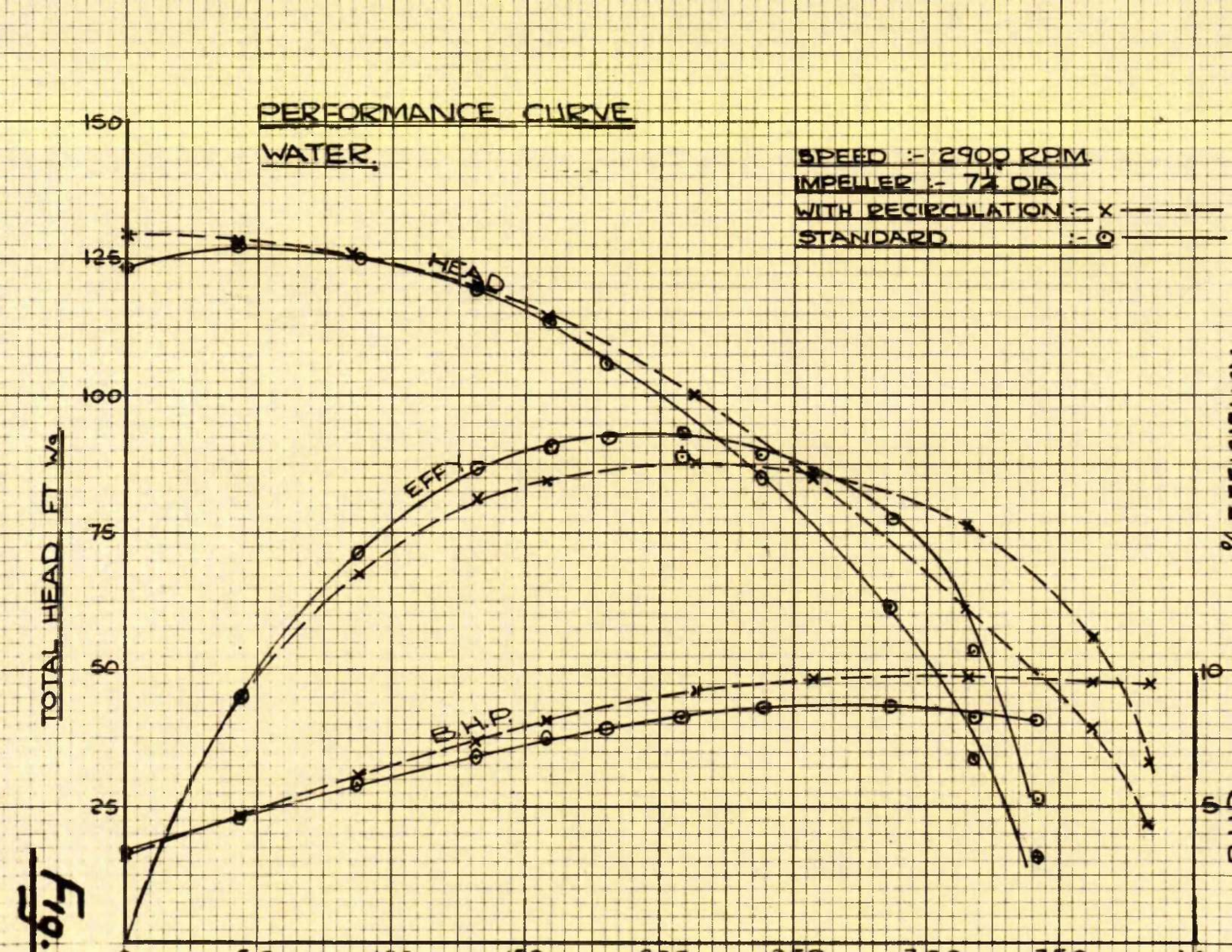
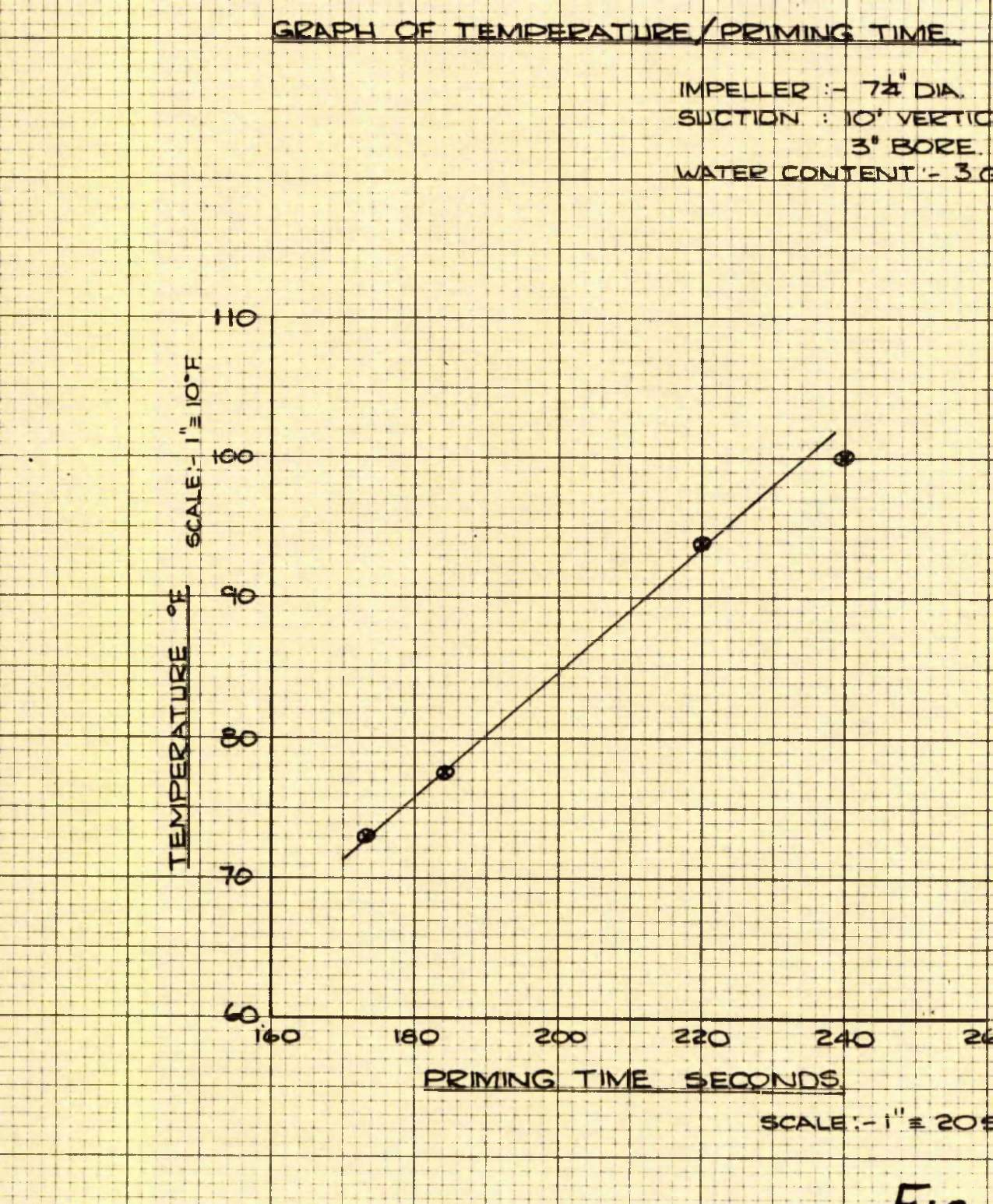

PERFORMANCE CURVE
WATER.
SPEED :- 2900 R.P.M.
IMPELLER :- 7¼ DIA.
WITH RECIRCULATION :- X
STANDARD :- ⊙
HEAD
EFF
B.H.P.
TOTAL HEAD FT Wg
150
125
100
75
50
25
10
5

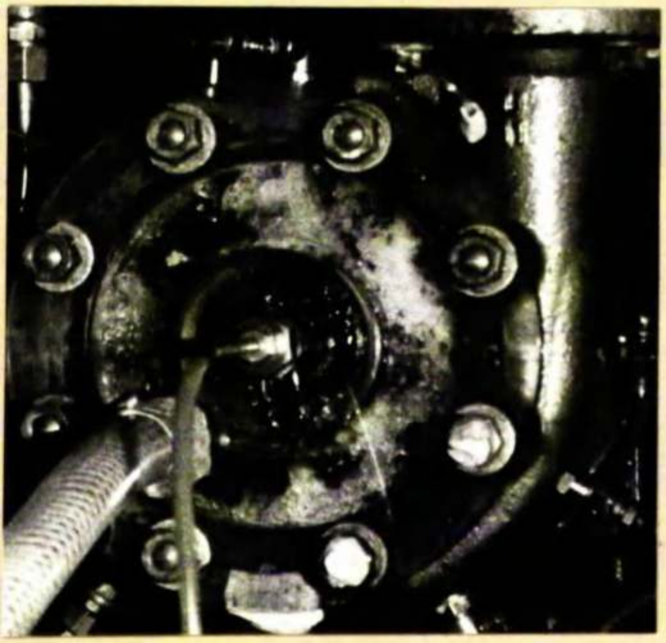
Fig.



GRAPH OF TEMPERATURE/PRIMING TIME.
IMPELLER :- 7¼" DIA.
SUCTION : 10' VERTIC
3" BORE.
WATER CONTENT :- 3
TEMPERATURE °F SCALE:- 1" ≡ 10°F
110
100
90
80
70
60
160
180
200
220
240
PRIMING TIME SECONDS.
SCALE :- 1" ≡ 20

Fig.

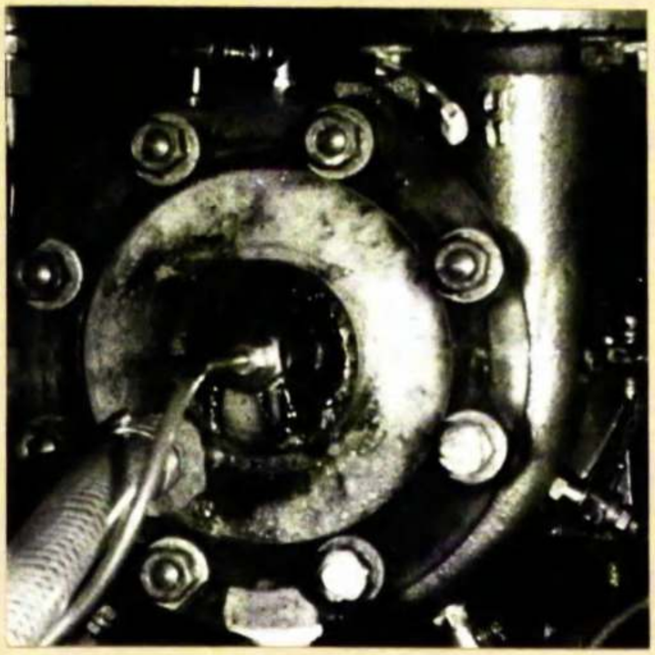
Vacuum



2" water

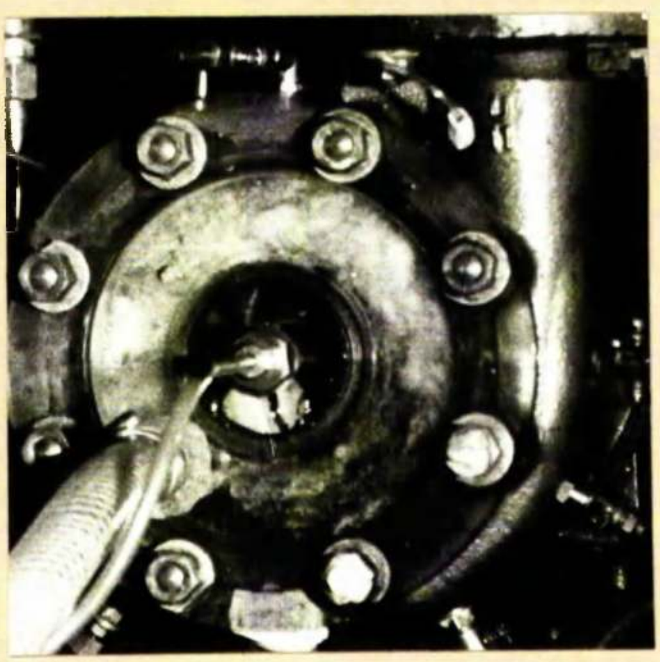
10" water

18" water



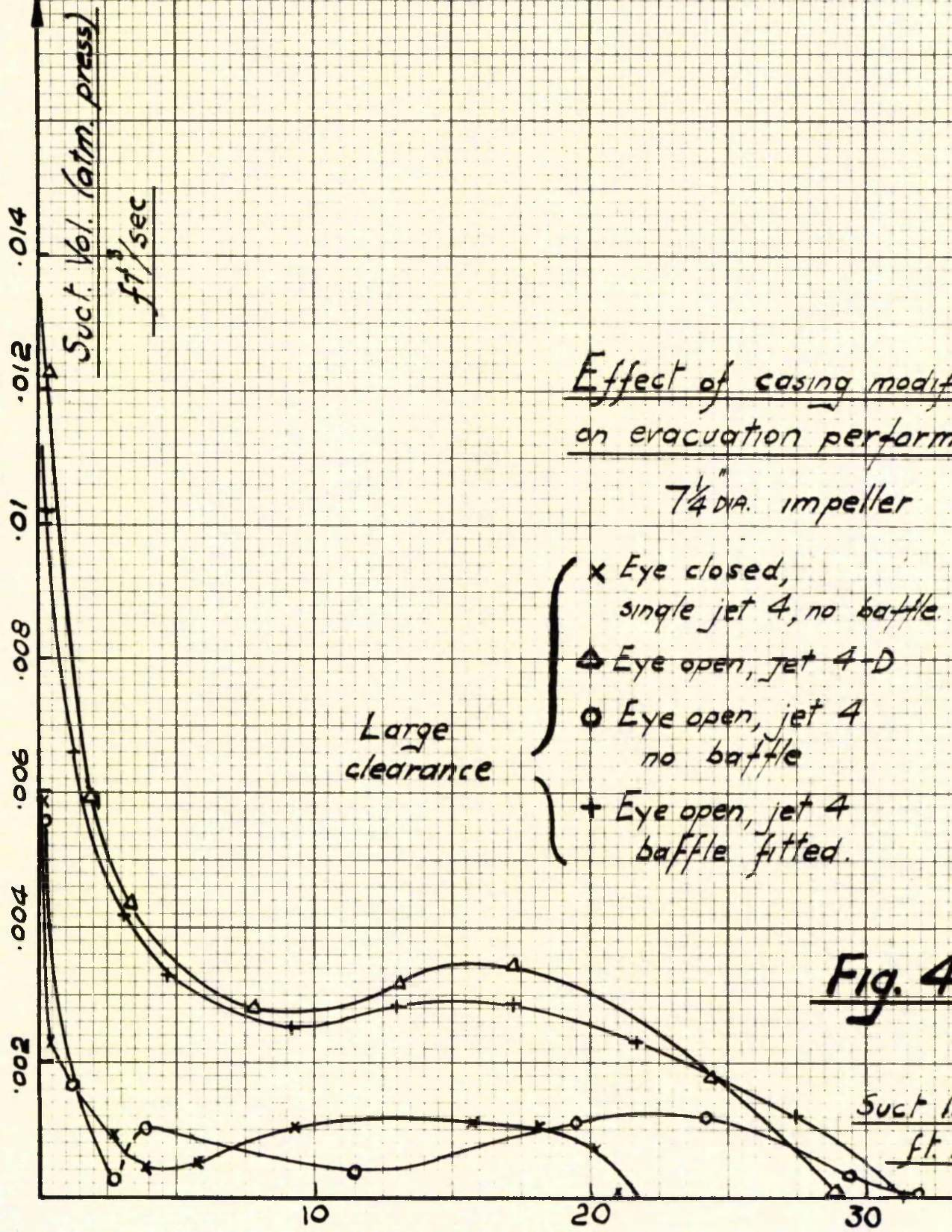
4" Mercury

14" Mercury

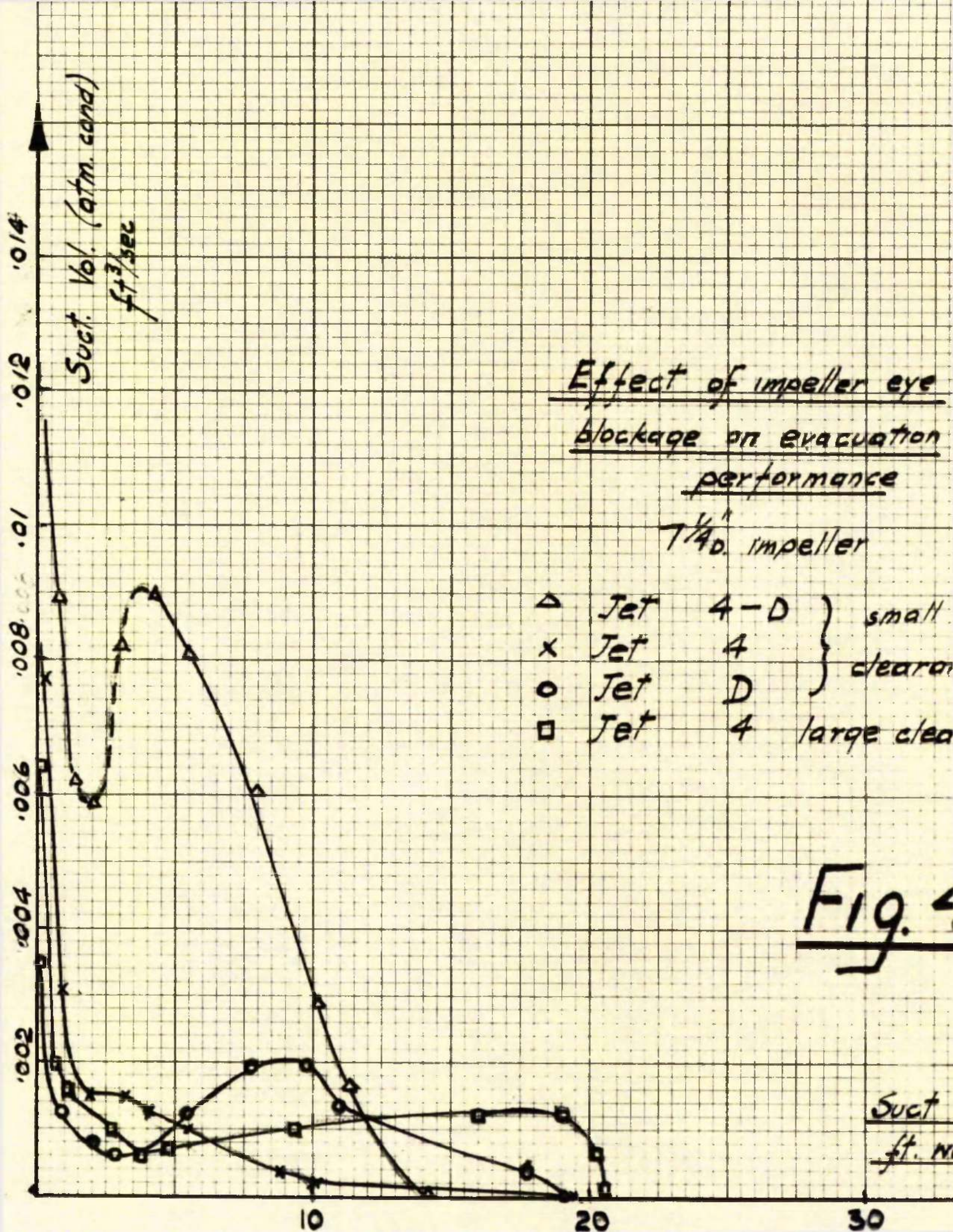


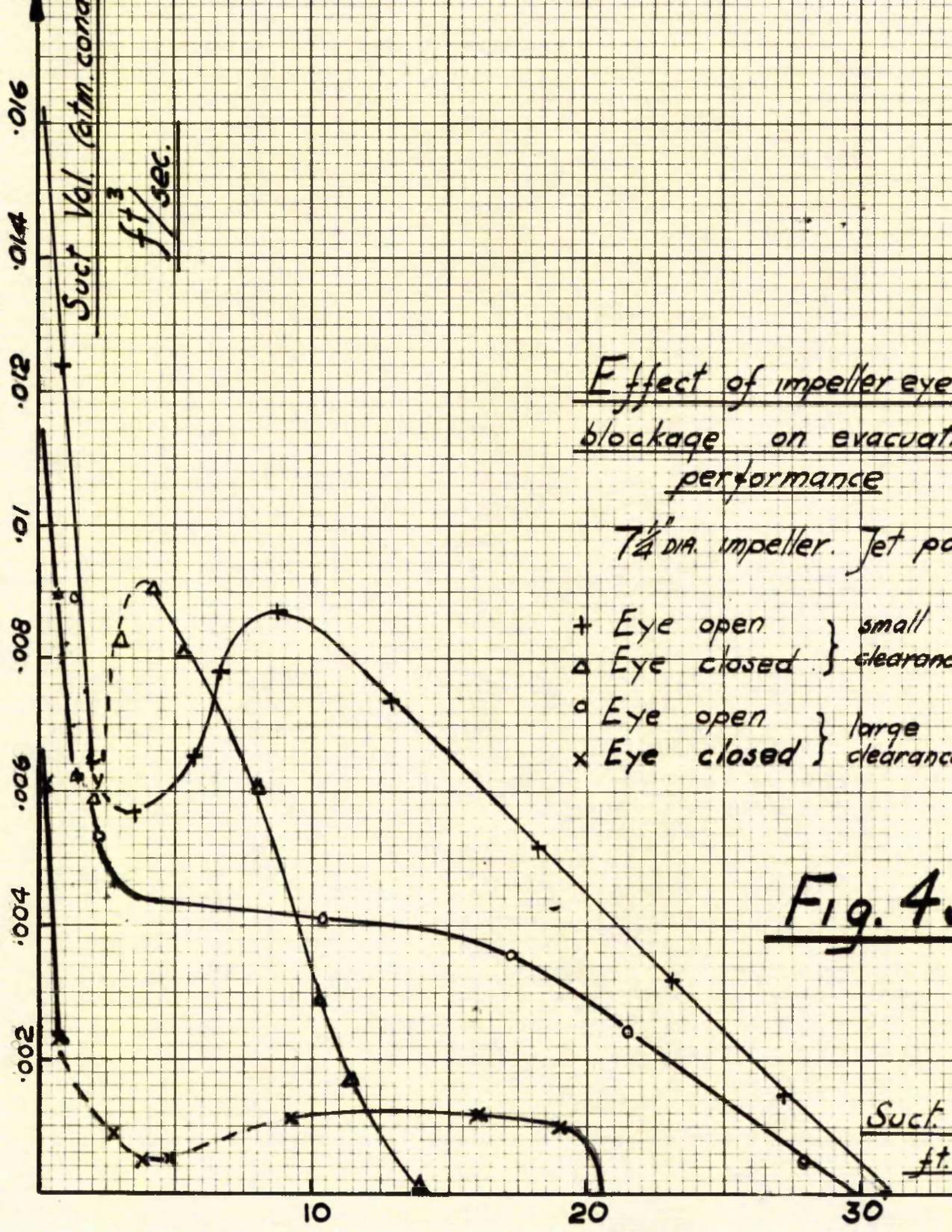
End clearance space u
evacuation conditions.
(Single jet, long baffle, la
clearance)

Fig



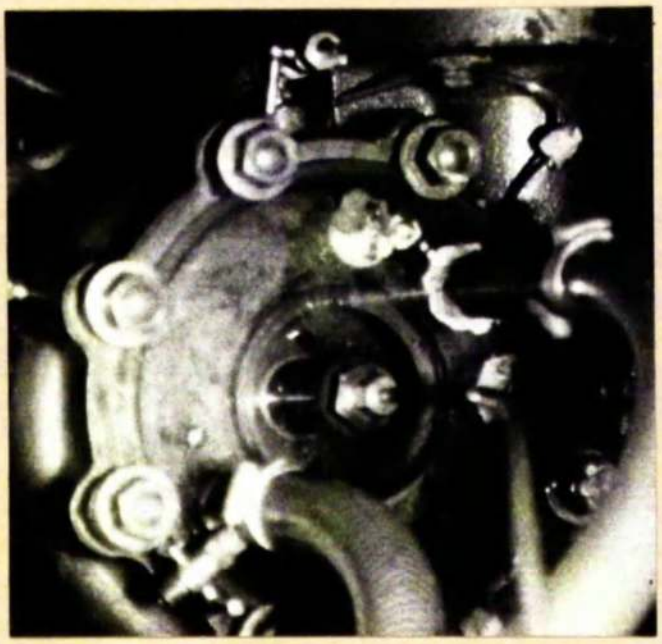
Suct. Vol. (atm. press)
ft³/sec
.014
.012
.01
.008
.006
.004
.002
10
20
30
Effect of casing modif
on evacuation perform
7¼" DIA. impeller
× Eye closed, single jet 4, no baffle
△ Eye open, jet 4-D
○ Eye open, jet 4 no baffle
Large clearance
+ Eye open, jet 4 baffle fitted.
Fig. 4
Suct
ft.


Suct. Vol. (atm. cond)
ft³/sec
.014
.012
.01
.008
.006
.004
.002
Effect of impeller eye
blockage on evacuation
performance
7¼" D. impeller
Jet 4-D
Jet 4
Jet D
small
clearan
Jet 4 large clea
Fig. 4
Suct
ft. w
10
20
30



Suct. Vol. (atm. cond
$ft^3/sec.$
Effect of impeller eye
blockage on evacuati
performance
7¼" DIA. impeller. Jet pa
+ Eye open } small
Δ Eye closed } clearanc
o Eye open } large
x Eye closed } clearanc
Suct.
ft.
·016
·014
·012
·01
·008
·006
·004
·002
10
20
30

Fig. 4.



2" water Vacuum

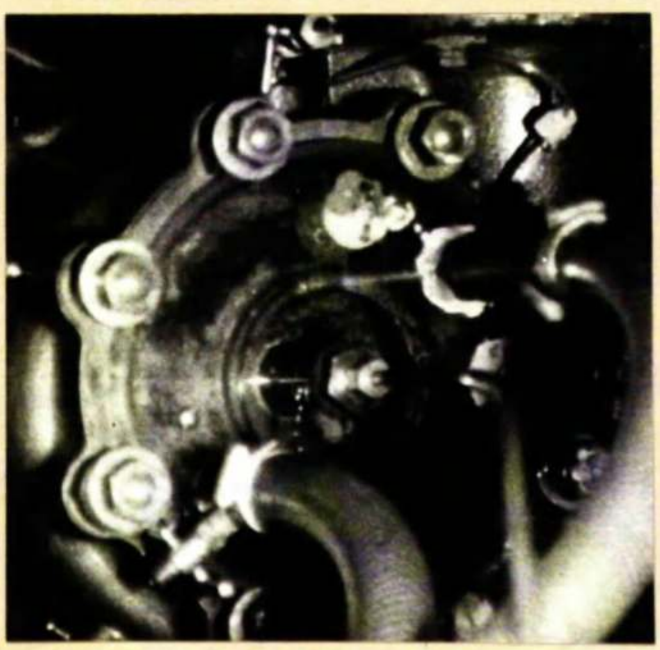
4" Mercury Vacuu

End clearance s

under air evacuat

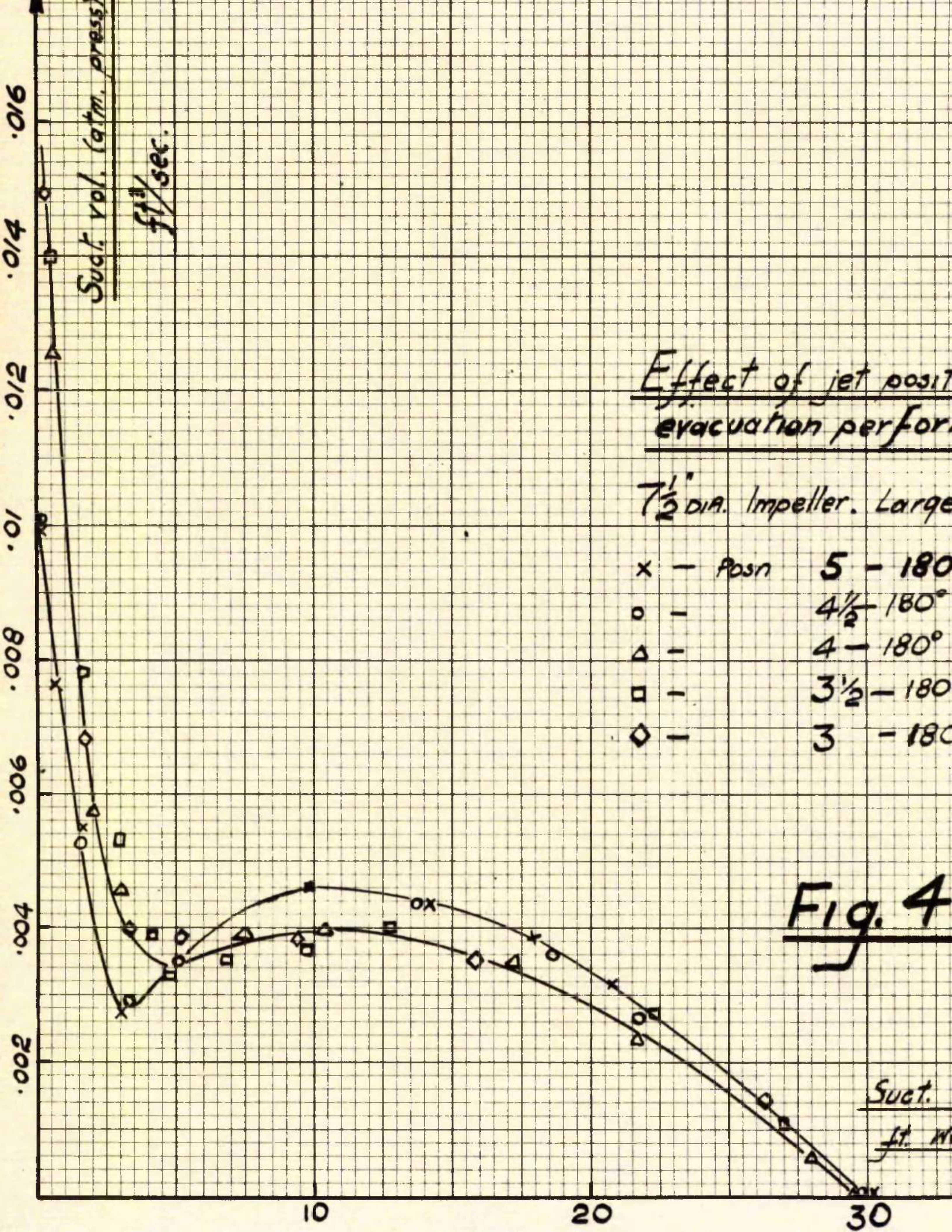
conditions.

(Two jets, reduced cleara



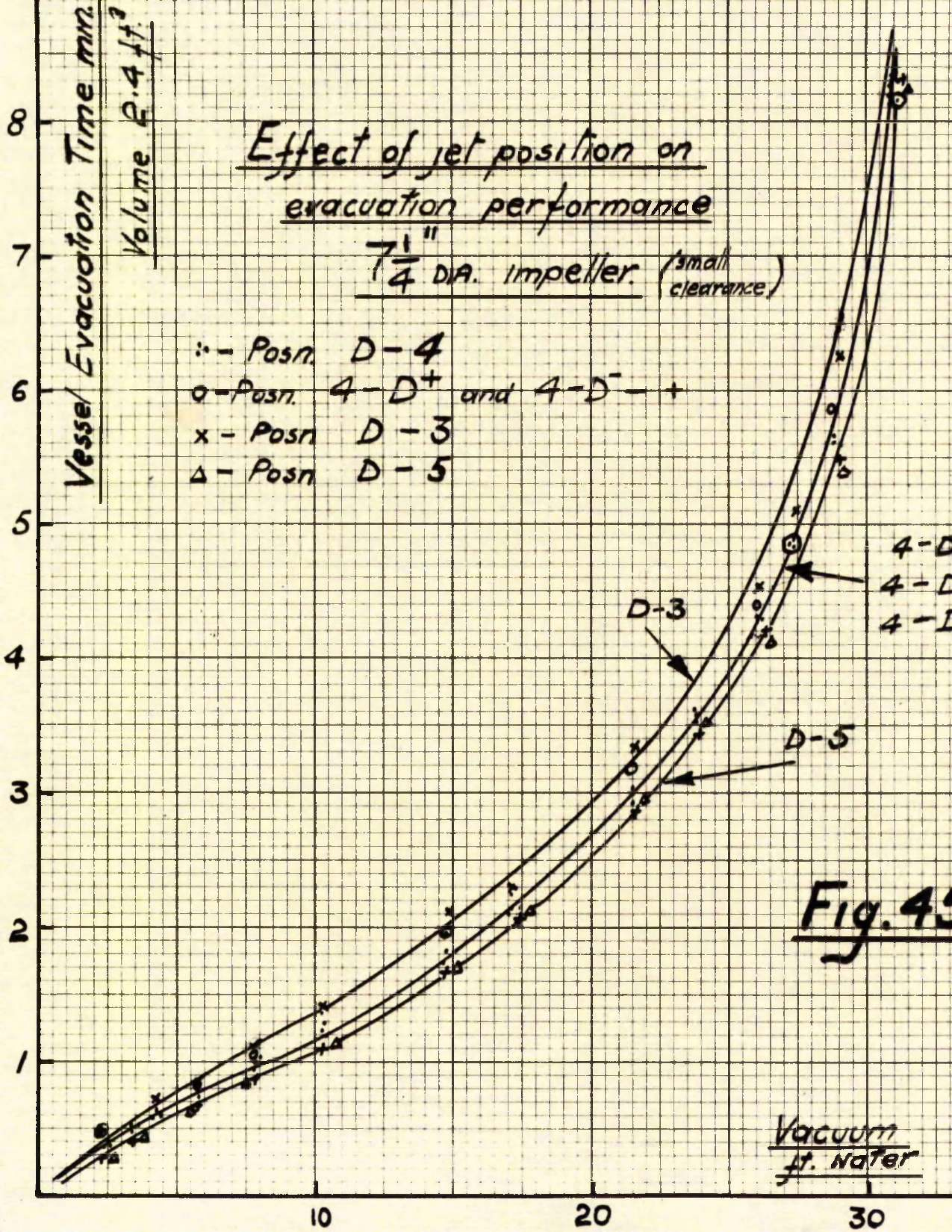
18" Mercury Vacuum.
(Increased circulation
causes a sligh reentrainment
of air bubbles from the separator)

Fi

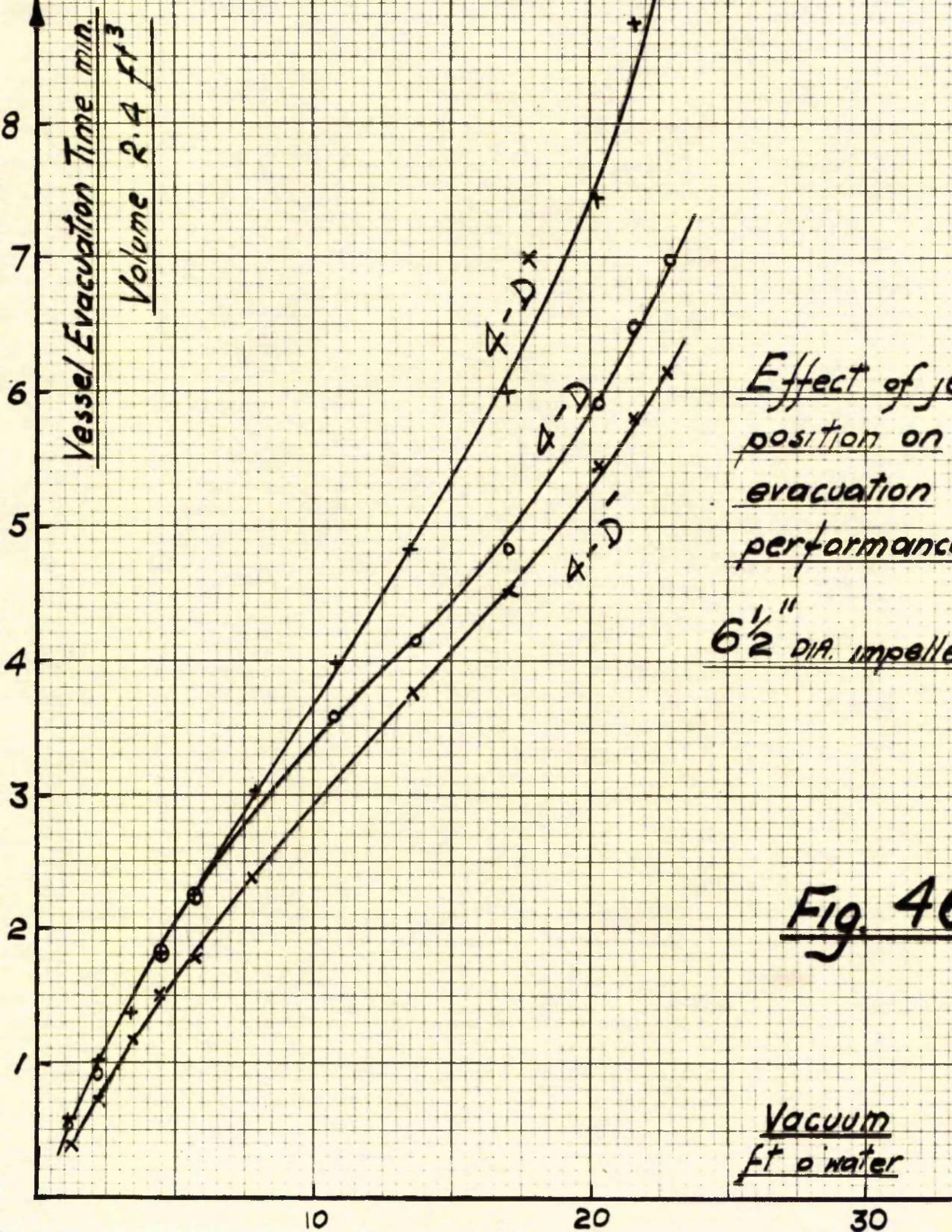


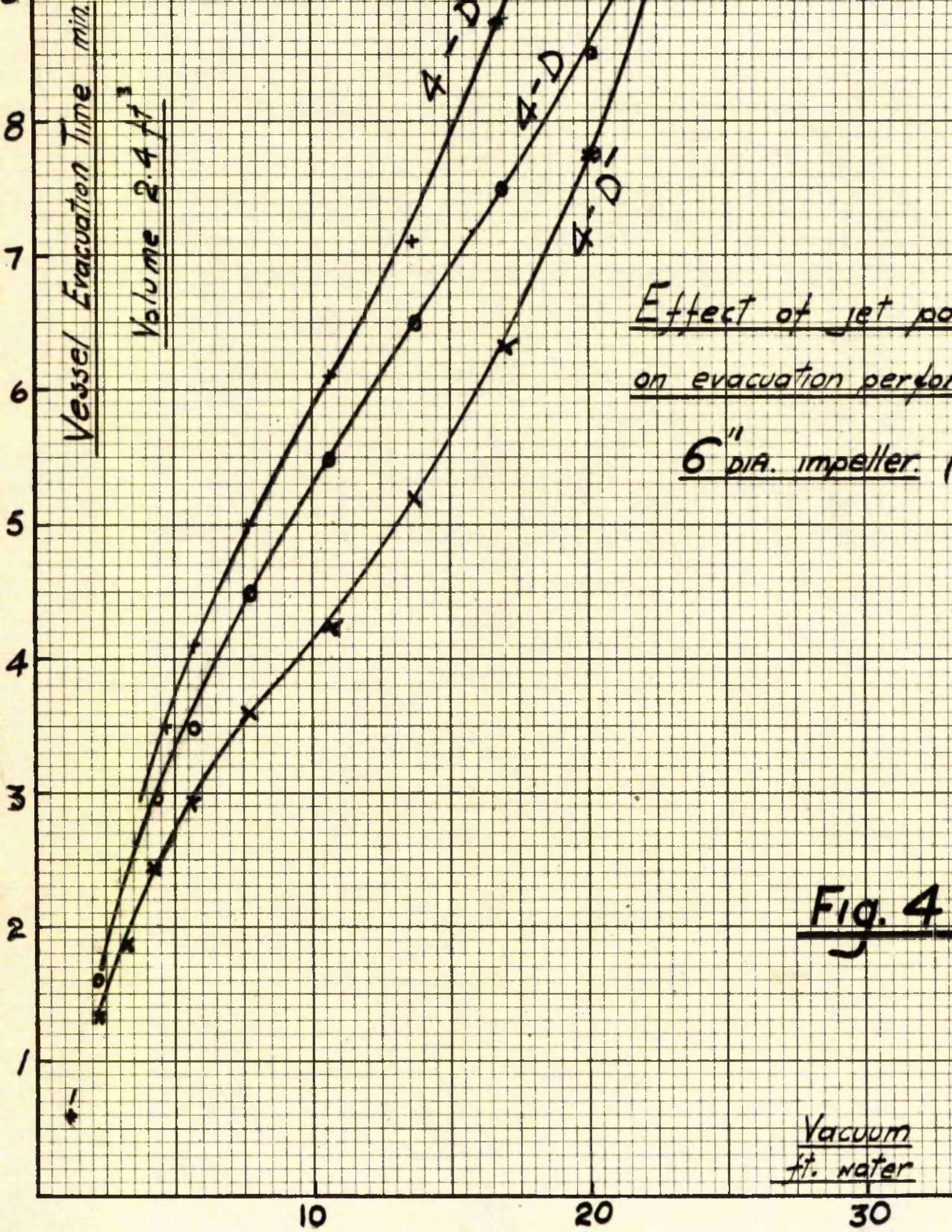
Effect of jet posit
evacuation perfor
$7\frac{1}{2}$" DIA. Impeller. Large
× — Posn 5 - 180
o — 4½ - 180°
Δ — 4 - 180°
□ — 3½ - 180
◇ — 3 - 180
Suct. vol. (atm. press)
ft³/sec.
Suct.
ft. W
.016
.014
.012
.01
.008
.006
.004
.002
10
20
30

Fig. 4



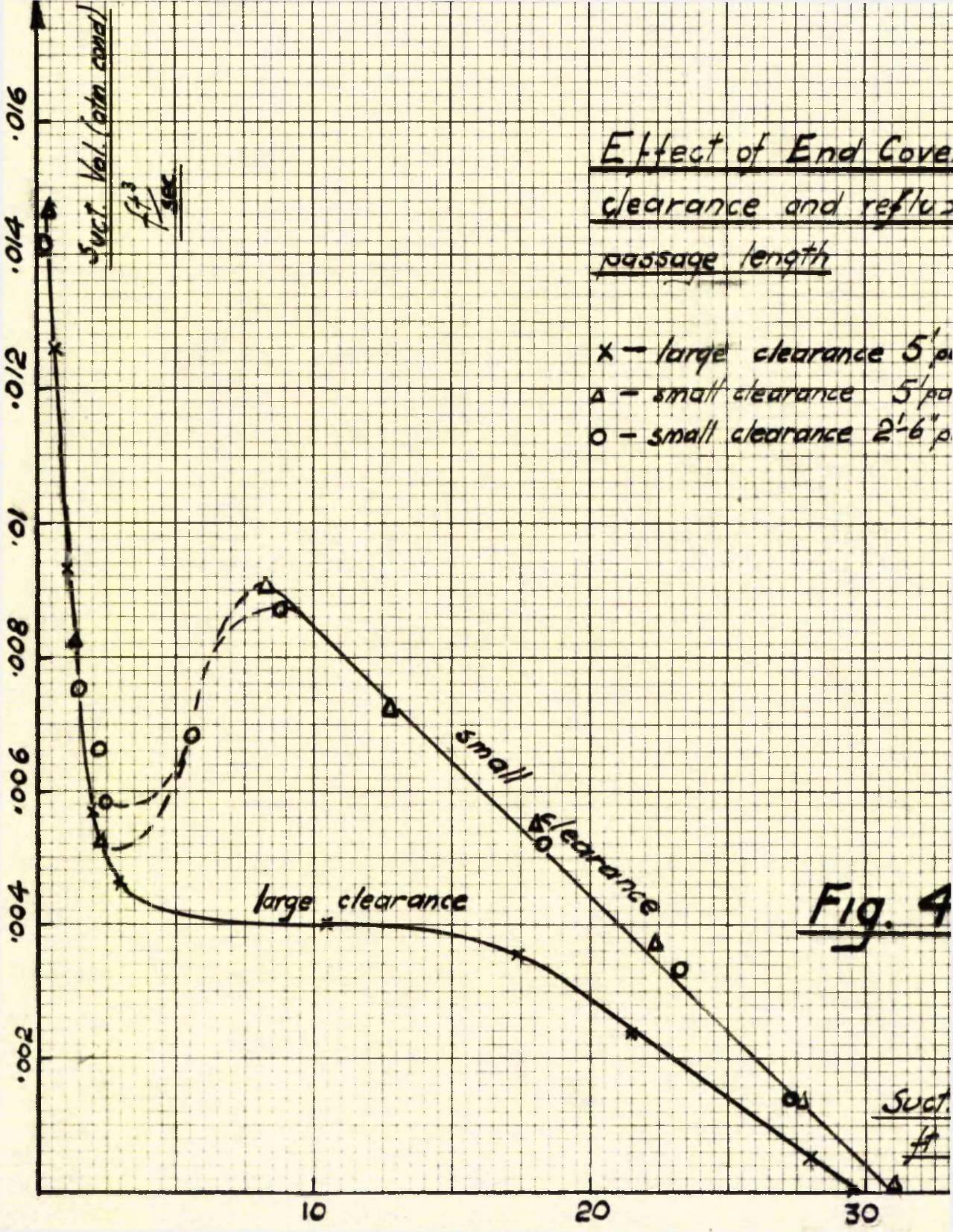
Effect of jet position on evacuation performance
$7\frac{1}{4}$" DIA. impeller. (small clearance)
∴ - Posn. D - 4
o - Posn. $4-D^{+}$ and $4-D^{-}$ — +
x - Posn D - 3
Δ - Posn D - 5
Vessel Evacuation Time min.
Volume 2.4 ft.^{3}
Vacuum ft. Water
D-3
D-5
4-D
4-D
4-D
Fig.45
1
2
3
4
5
6
7
8
10
20
30


Vessel Evacuation Time min.
Volume 2.4 ft^3
4-D×
4-D
4-D'
Effect of j
position on
evacuation
performanc
6½" DIA. impelle
Fig. 46
Vacuum
ft o'water
1
2
3
4
5
6
7
8
10
20
30



Vessel Evacuation Time min.
Volume 2.4 ft³
Effect of jet po
on evacuation perfor
6" DIA. impeller. (
4-D
4-D
4'-D'
Vacuum
ft. water
1
2
3
4
5
6
7
8
10
20
30

Fig. 4



Effect of End Cove
clearance and reflu
passage length
x – large clearance 5' pa
Δ – small clearance 5' pa
o – small clearance 2'-6" p
small clearance
large clearance
Fig. 4
Suct. Vol. (atm cond) ft³/sec
.016
.014
.012
.01
.008
.006
.004
.002
10
20
30
Suct
ft

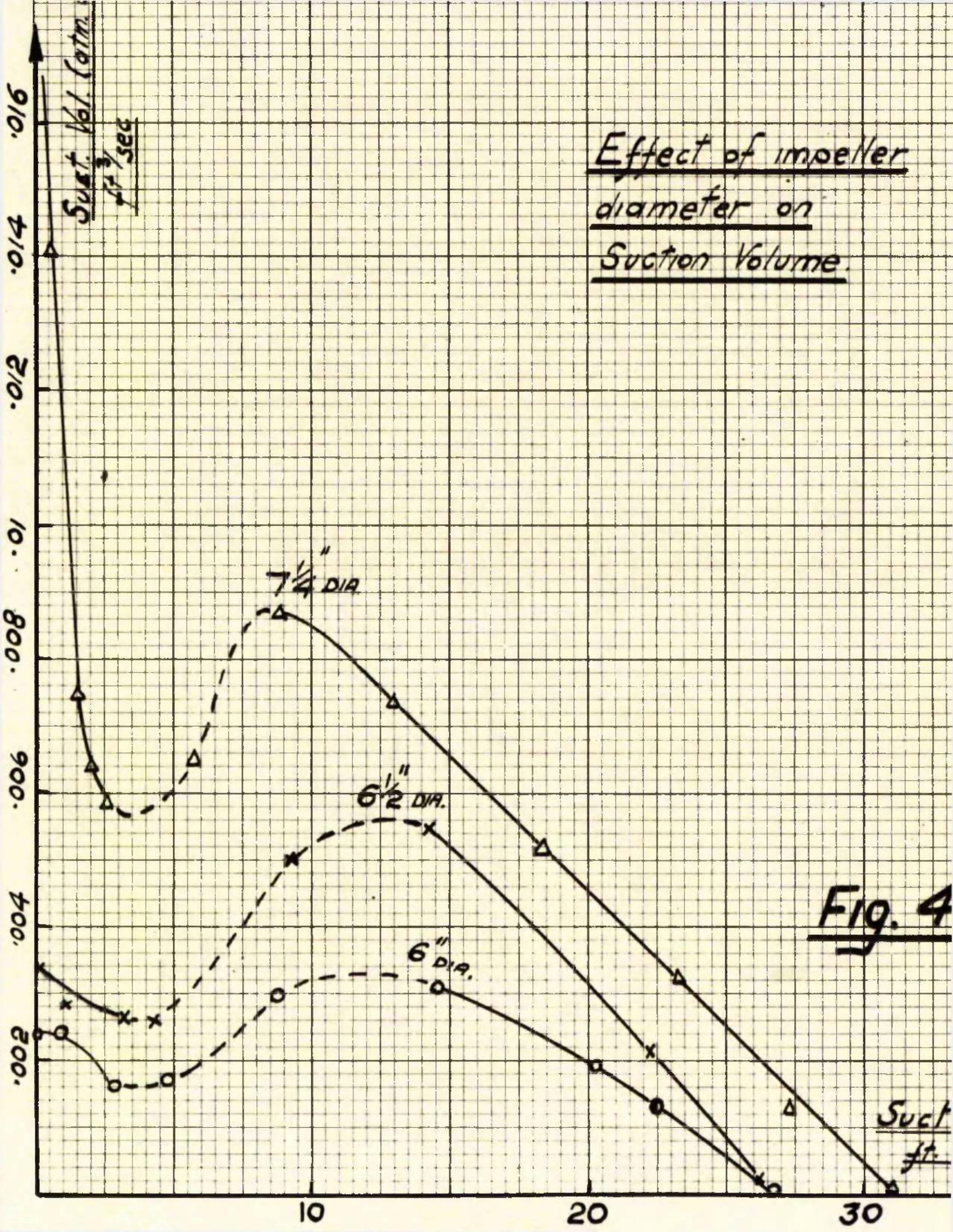
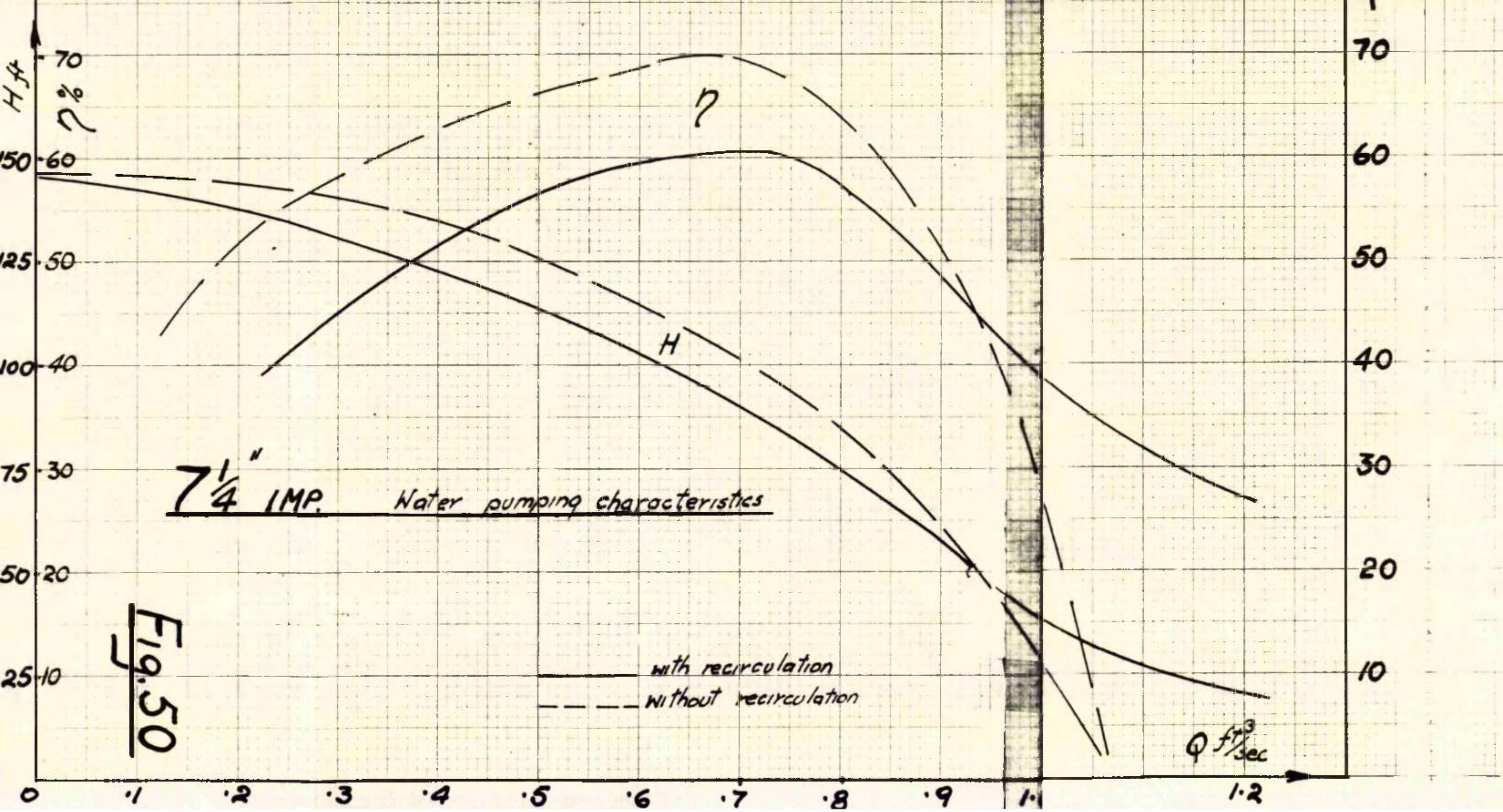
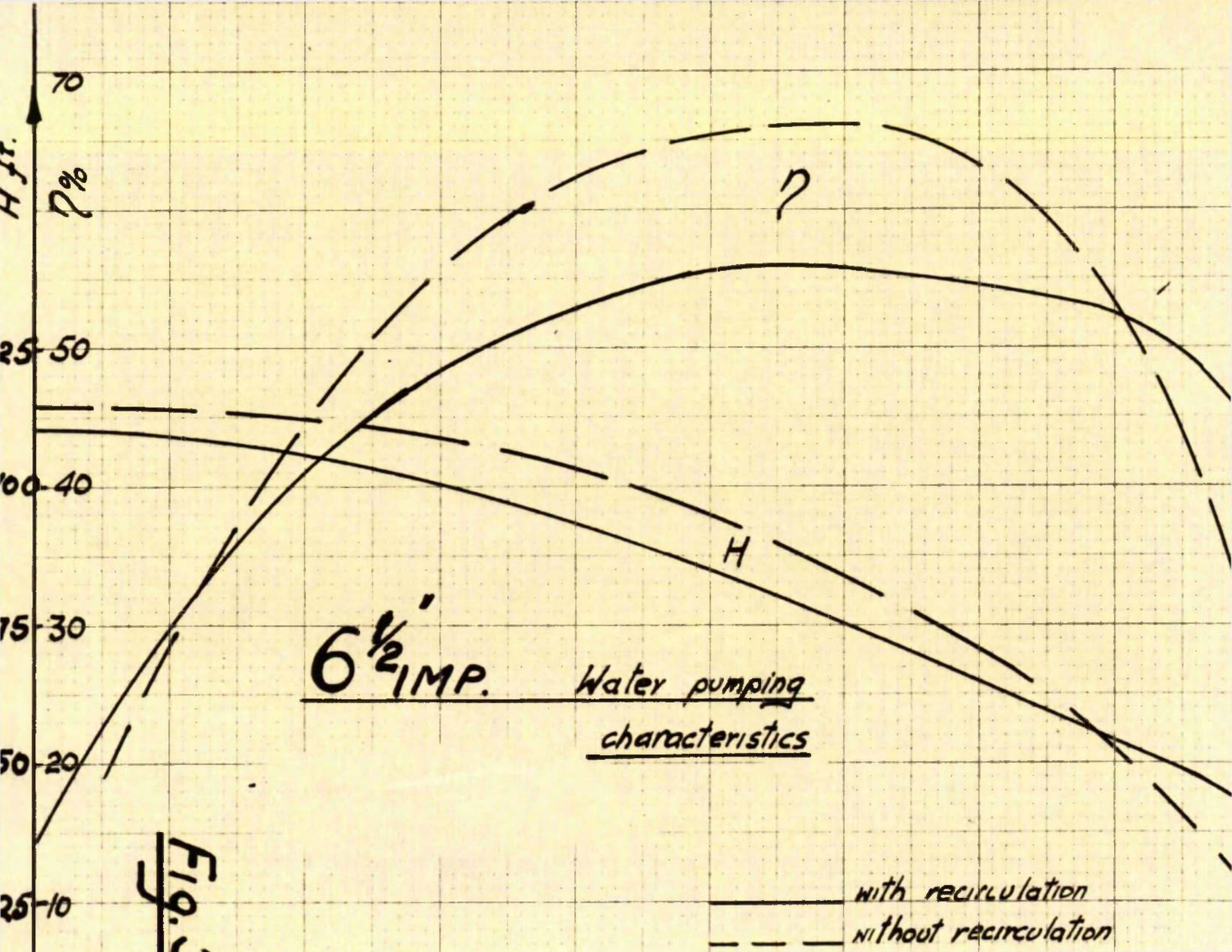

Effect of impeller diameter on Suction Volume.
Suct. Vol (atm.) ft³/sec
·016
·014
·012
·01
·008
·006
·004
·002
7¼" DIA.
6½" DIA.
6" DIA.
10
20
30

Fig. 4



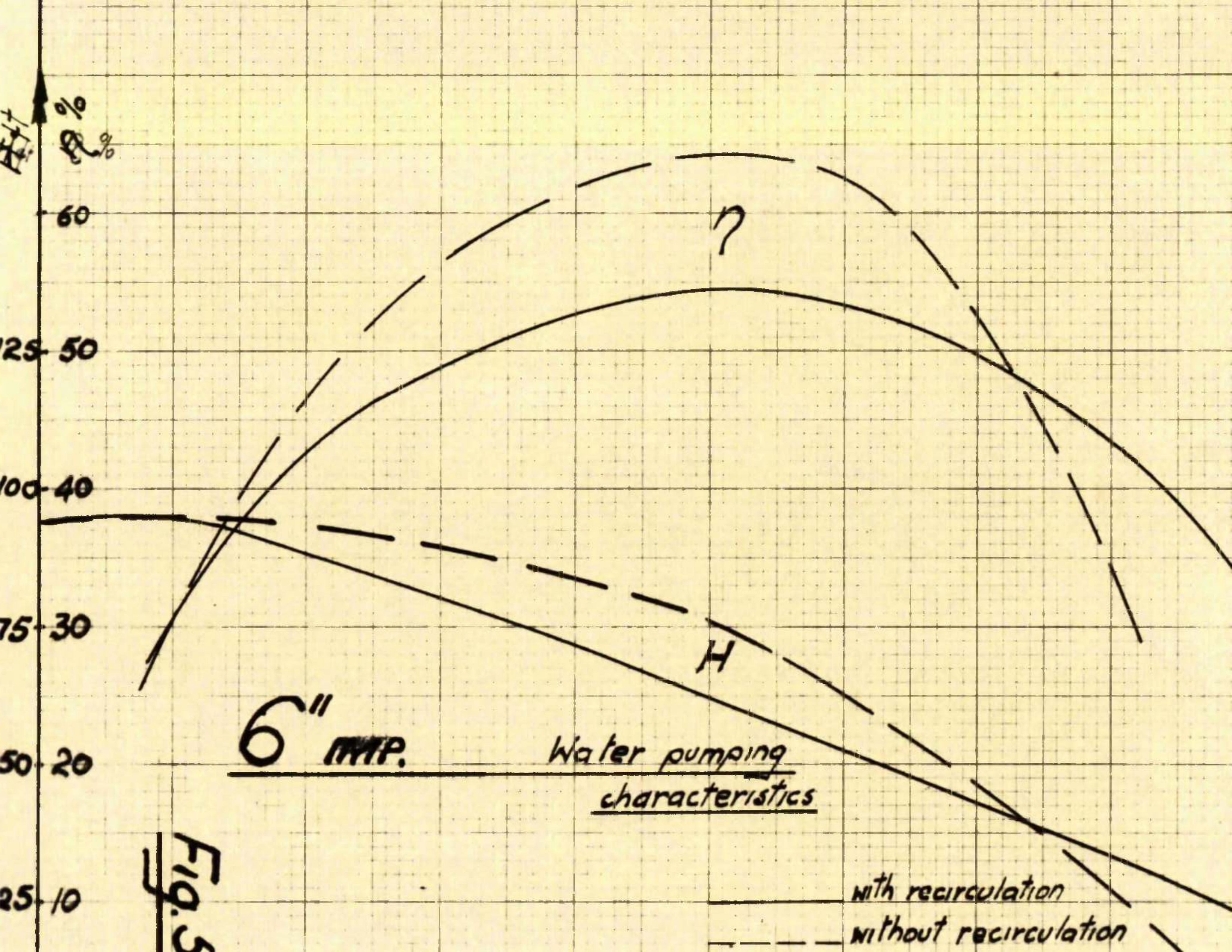
H ft
η %
150 60
125 50
100 40
75 30
50 20
25 10
0
70
η
H
7 1/4" IMP. Water pumping characteristics
with recirculation
without recirculation
Q ft³/sec
·1 ·2 ·3 ·4 ·5 ·6 ·7 ·8 ·9 1·1 1·2

Fig.50



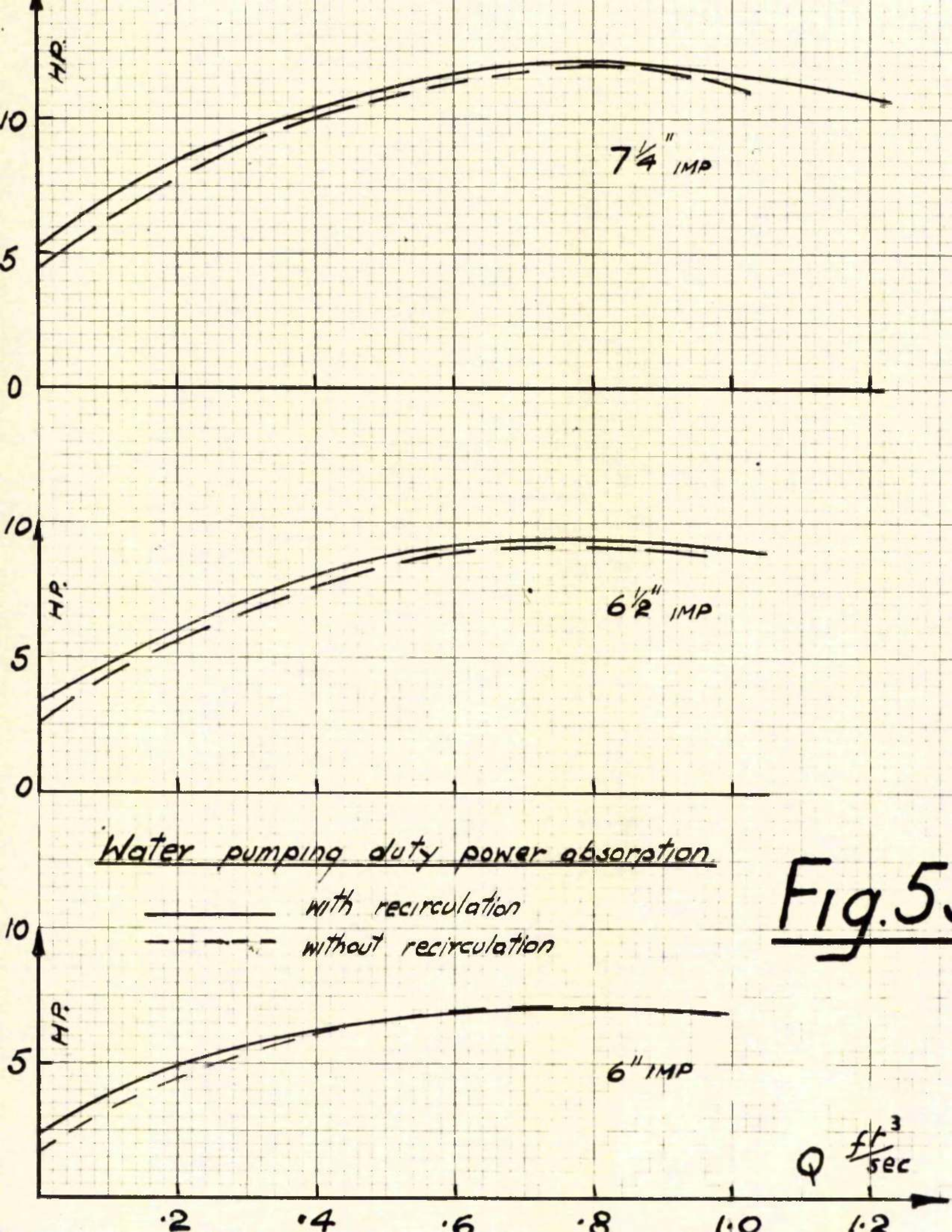
H ft.
η%
70
25-50
100-40
75-30
50-20
25-10
η
H
6½" IMP. Water pumping characteristics
with recirculation
without recirculation

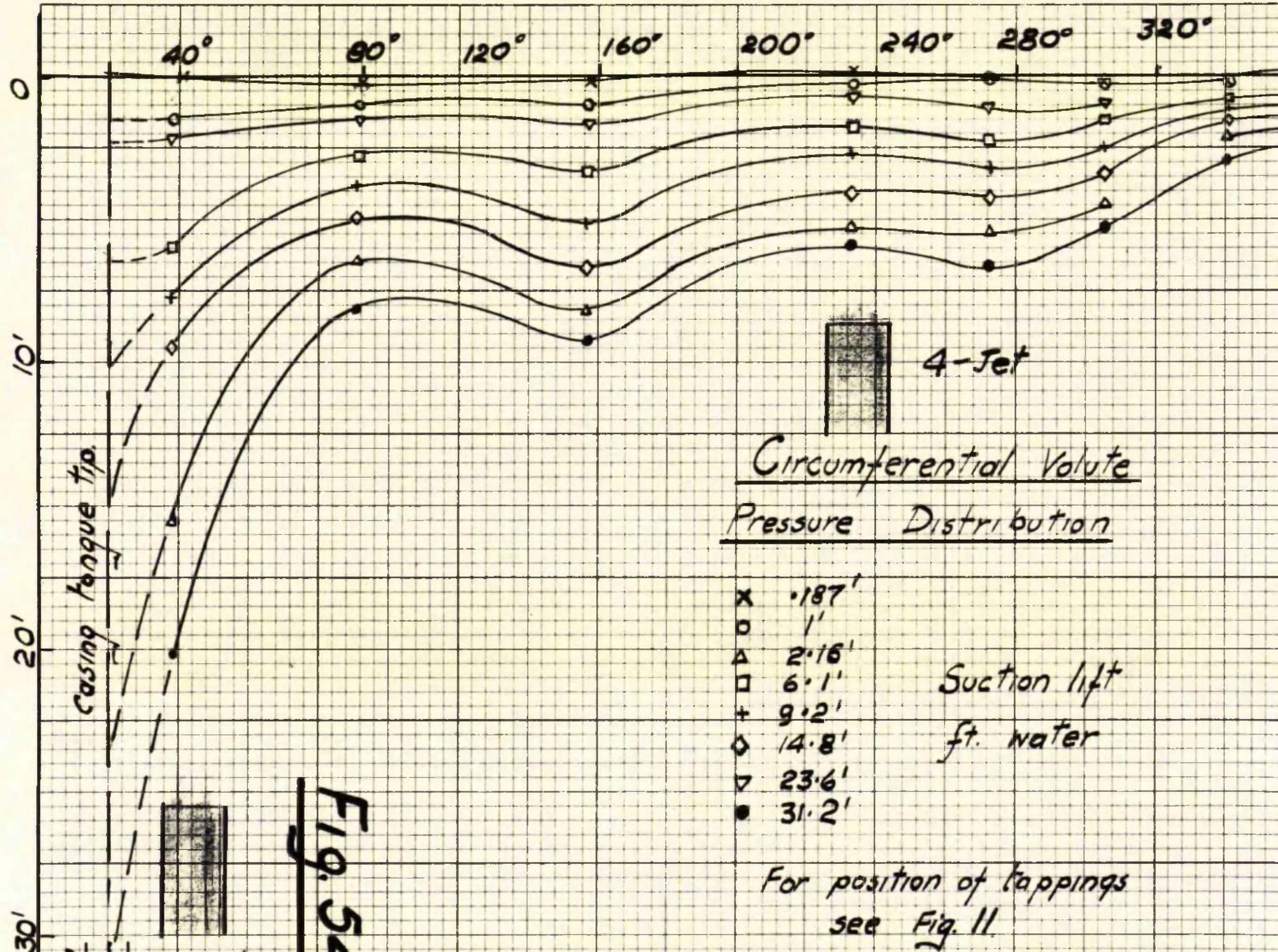
Fig. 3



H ft
η %
60
125 50
100 40
75 30
50 20
25 10
η
H
6" MMP.
Water pumping characteristics
with recirculation
without recirculation

Fig.5


Water pumping duty power absorption.
with recirculation
without recirculation
Fig.5
7¼" IMP
6½" IMP
6" IMP
H.P.
H.P.
H.P.
Q ft³/sec
0
5
10
0
5
10
5
10
·2
·4
·6
·8
1·0
1·2



40°
80°
120°
160°
200°
240°
280°
320°
0
10'
20'
30'
Casing tongue tip.
4-Jet
Circumferential Volute
Pressure Distribution
× ·187'
o 1'
△ 2·16'
□ 6·1'
+ 9·2'
◇ 14·8'
▽ 23·6'
● 31·2'
Suction lift
ft. water
For position of tappings
see Fig. 11

Fig. 54

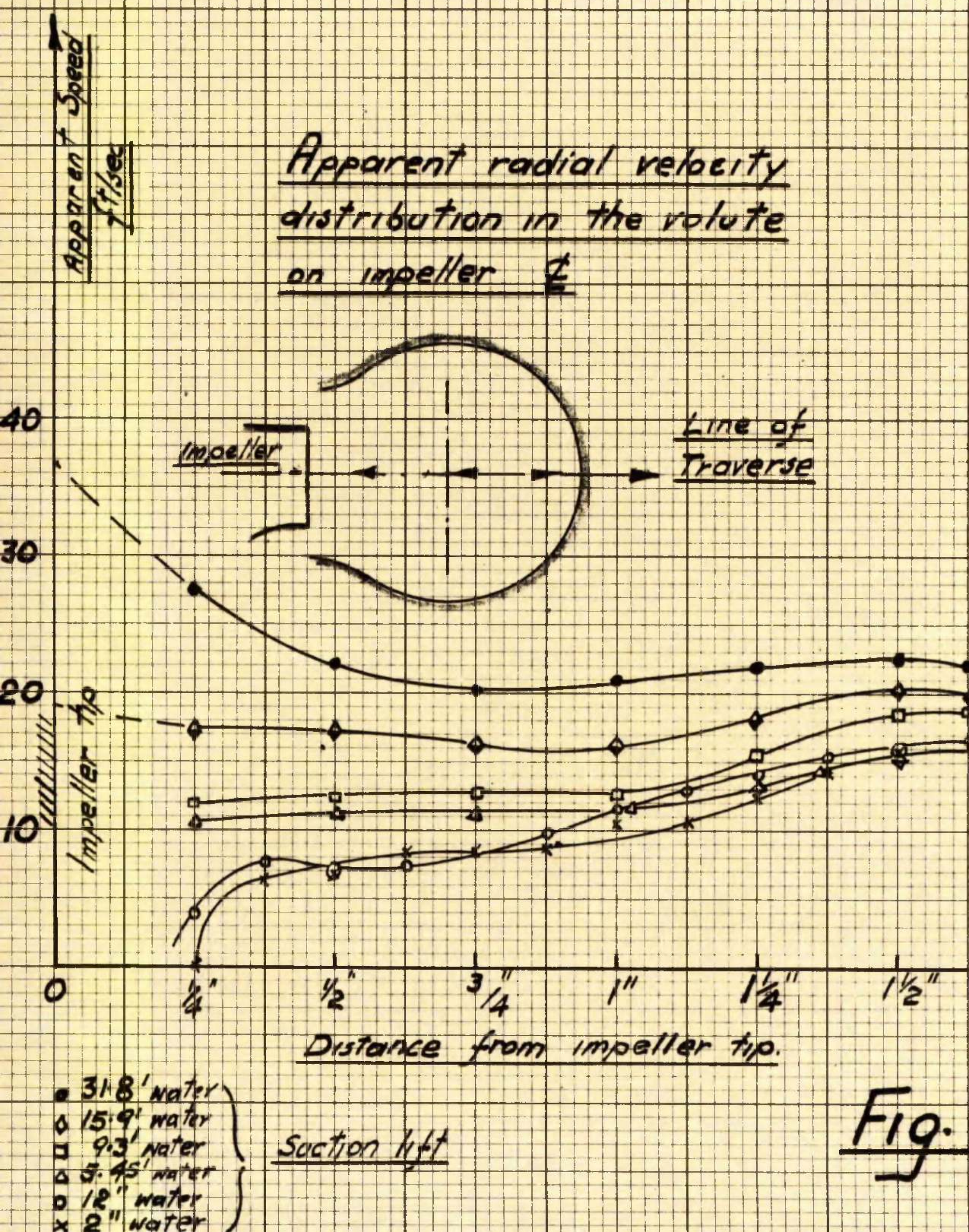
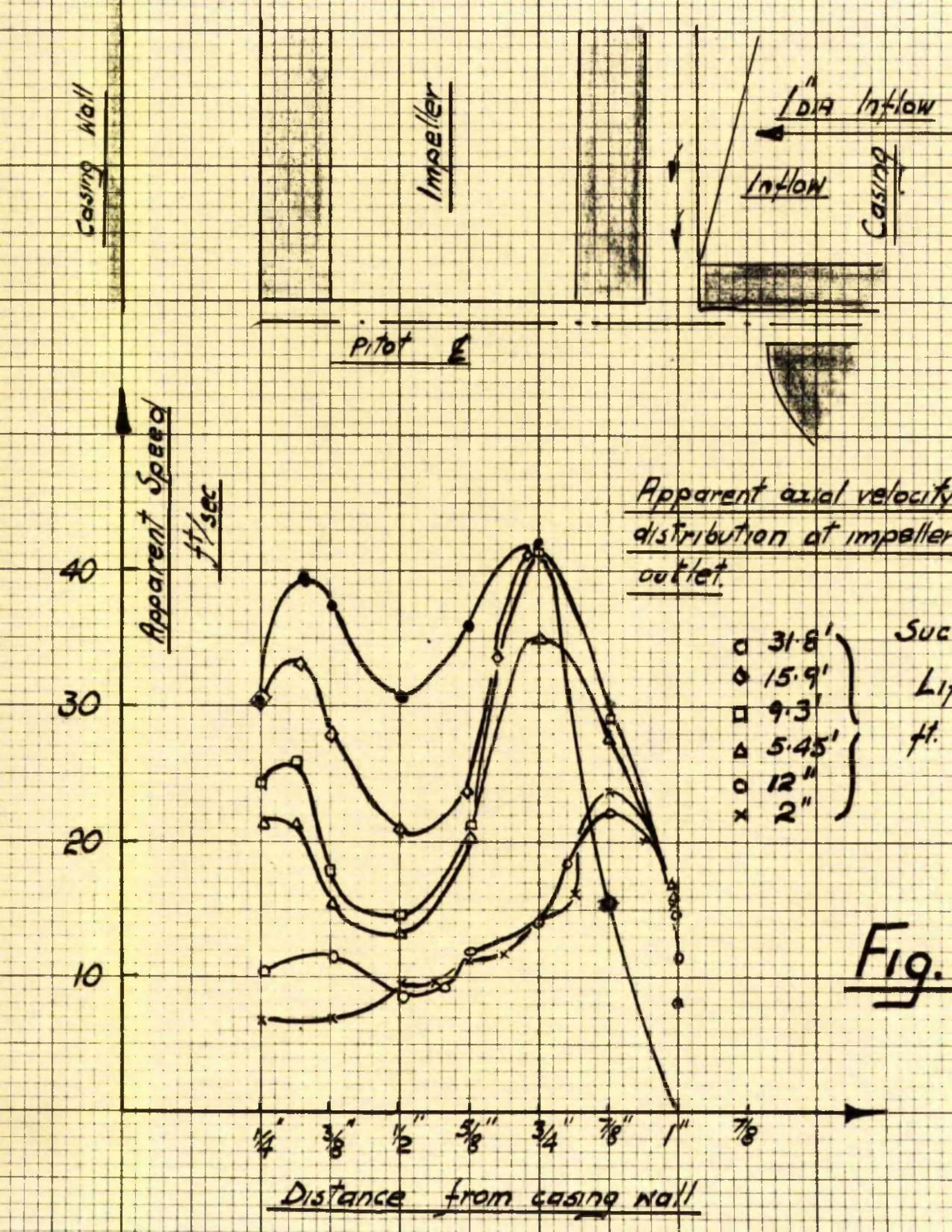
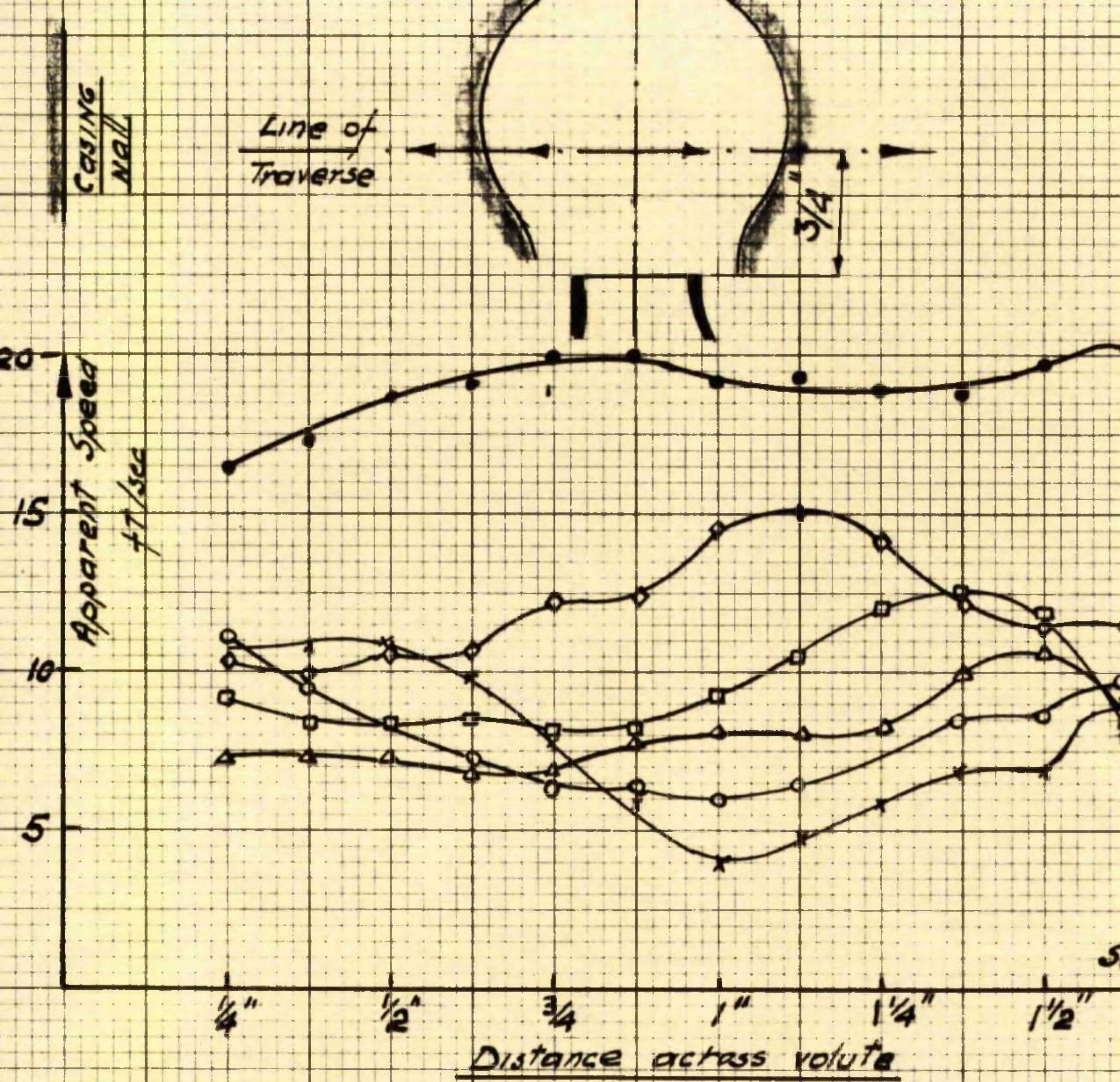

Apparent radial velocity distribution in the volute on impeller ℄
Apparent Speed ft/sec
40
30
20
10
0
Impeller
Line of Traverse
Impeller tip
1/4"
1/2"
3/4"
1"
1 1/4"
1 1/2"
Distance from impeller tip.
● 31·8' water
◇ 15·9' water
□ 9·3' water
△ 5·45' water
○ 12" water
× 2" water
Suction lift

Fig.


Casing Wall
Impeller
1" DIA Inflow
Inflow
Casing
Pitot
Apparent Speed ft/sec
Apparent axial velocity
distribution at impeller
outlet.
○ 31·8'
◇ 15·9'
□ 9·3'
△ 5·45'
○ 12"
× 2"
Suc
Li
ft.
40
30
20
10
1/4" 3/8" 1/2" 5/8" 3/4" 7/8" 1" 7/8
Distance from casing wall
Fig.

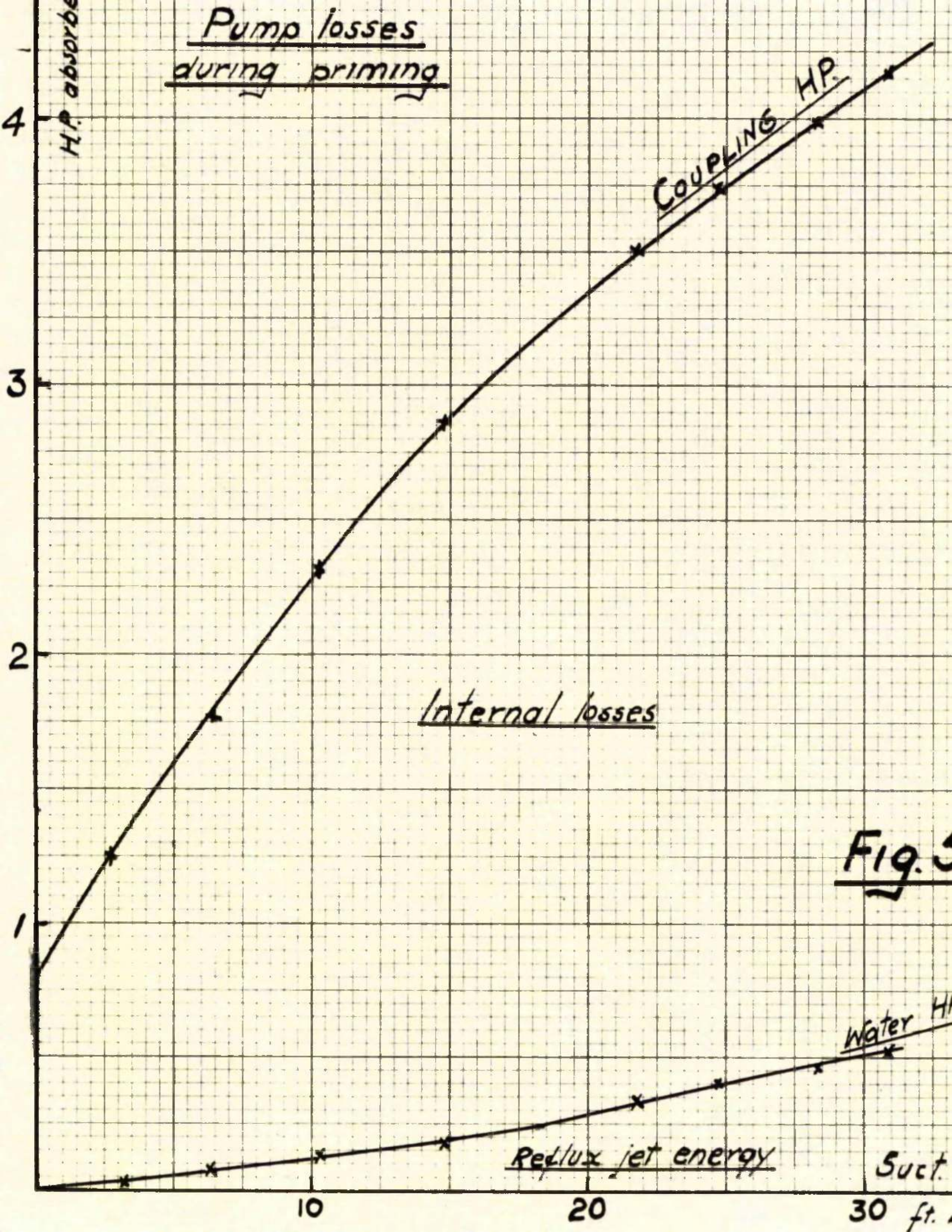


Apparent axial velocity distribution in the volute

Fig

● 31·8'
◇ 15·9'
□ 9·3'
△ 3·45'
○ 12"
× 2"

Suction lift ft. water



H.P. absorbe
Pump losses
during priming
COUPLING HP.
Internal losses
Water HP
Reflux jet energy
Suct.
4
3
2
1
10
20
30
ft.

Fig. 5

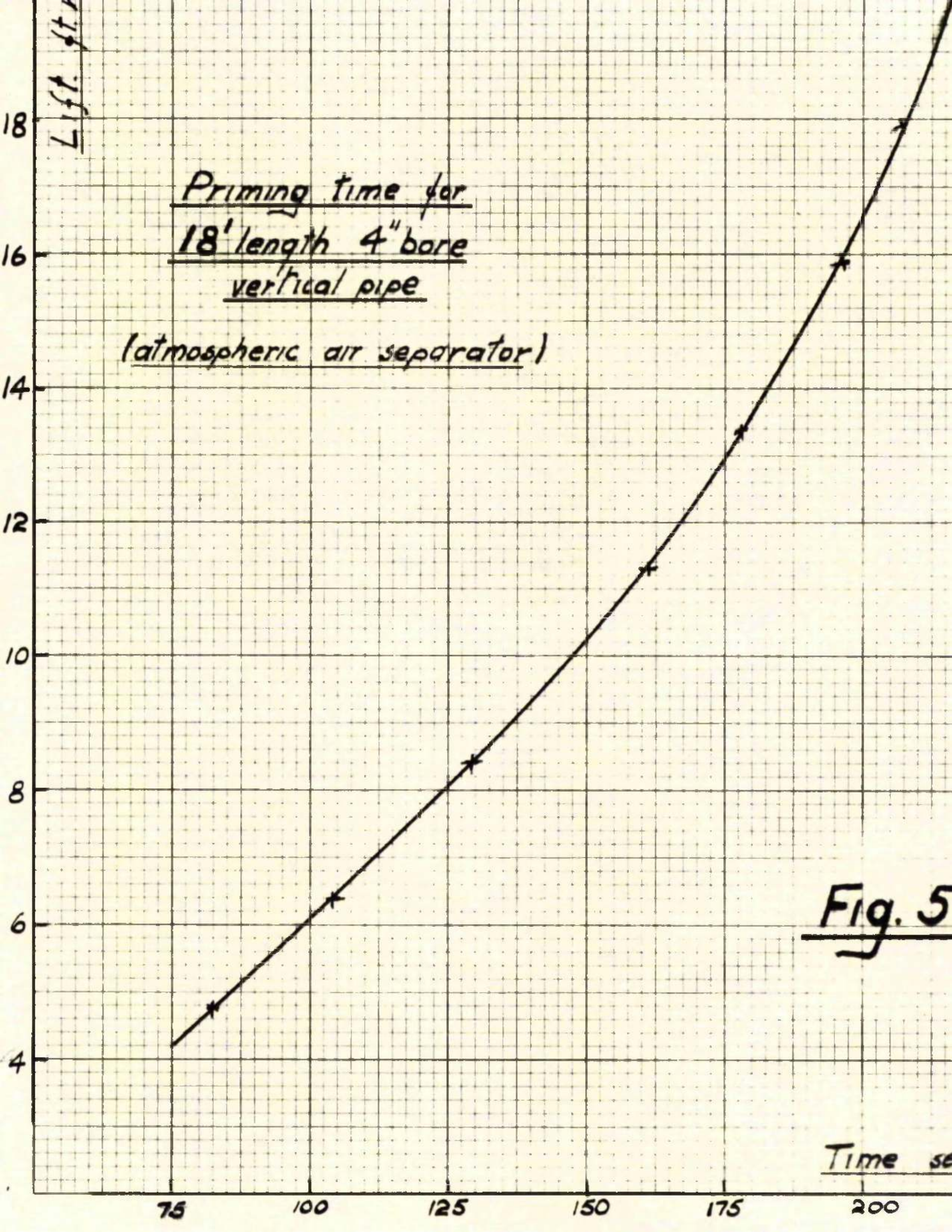

Lift ft.
Priming time for
18' length 4" bore
vertical pipe
(atmospheric air separator)
18
16
14
12
10
8
6
4
75
100
125
150
175
200
Time se

Fig. 5

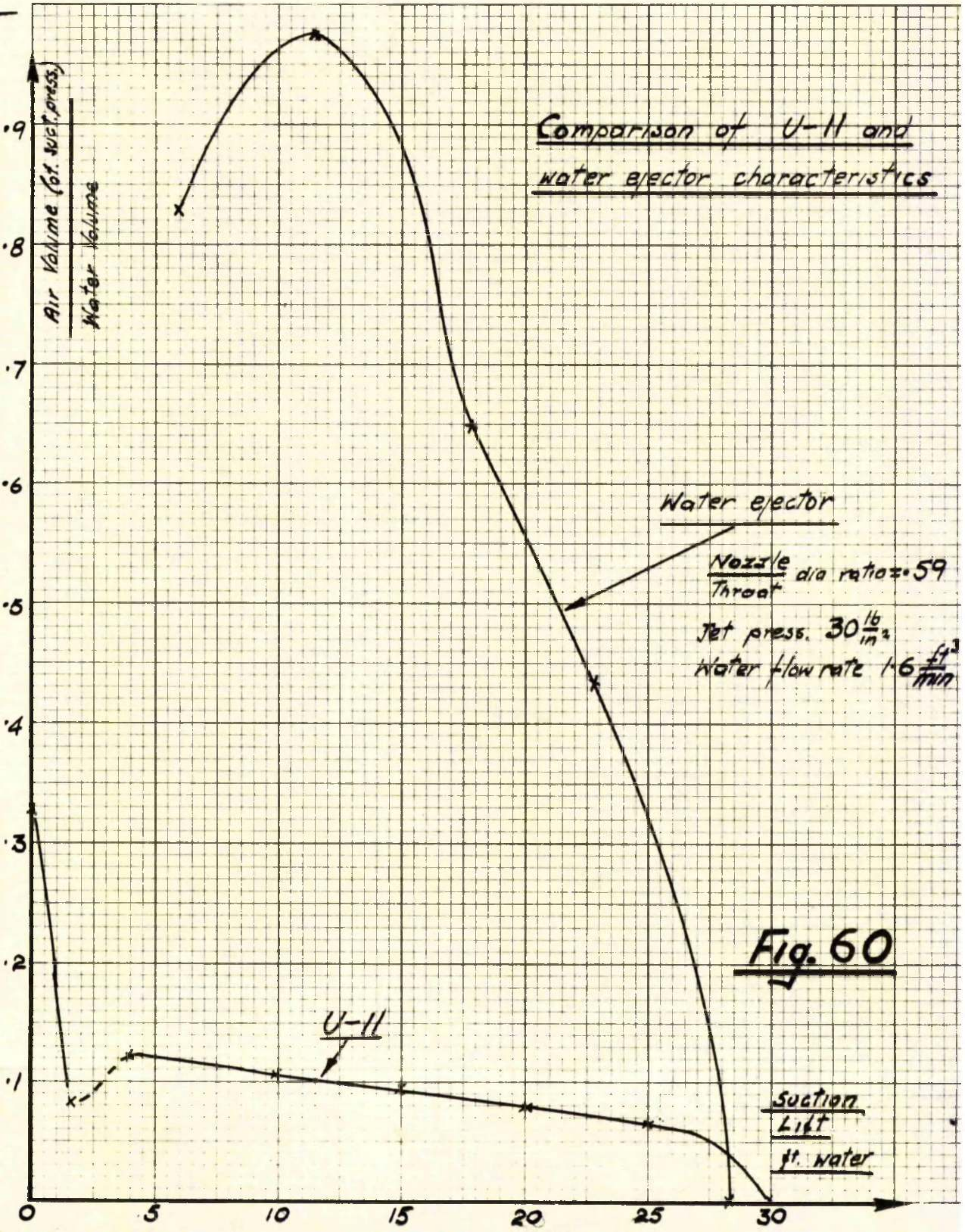
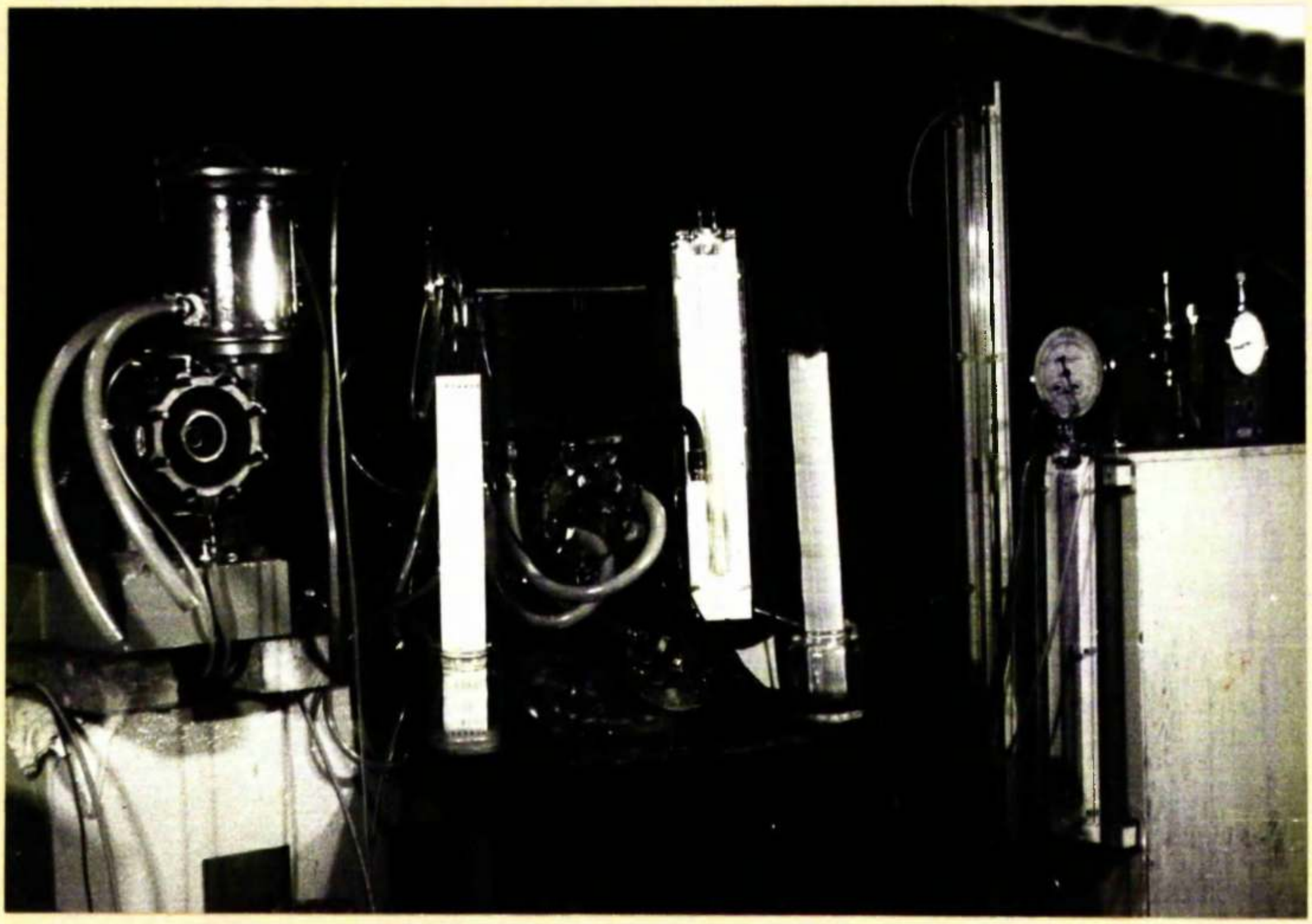

Comparison of U-11 and water ejector characteristics
Air Volume (at suct. press.) / Water Volume
Water ejector
Nozzle/Throat dia ratio = ·59
Jet press. 30 lb/in²
Water flow rate 1·6 ft³/min
U-11
Suction Lift ft. water
·1
·2
·3
·4
·5
·6
·7
·8
·9
0
5
10
15
20
25
30

Fig. 60



Assembly of equipment used in the investigation

Fig.

Original pump showing the 1" DIA. refluxing connection.