INVESTIGATIONS ON THE PERFORMANCES OF CENTRIFUGAL PUMPS WITH VARIOUS ALTERATIONS OF THE IMPELLER

Thesis by

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In Partial Fulfillment of the Requirements

For the Degree of Master of Science

CALIFORNIA INSTITUTE OF TECHNOLOGY

PASADENA, CALIFORNIA

1933

This Thesis consists of a research conducted on centrifugal pumps operating with the impellers altered in various ways such as to give various performance characteristics.

Performed under the direction of Dr. Robert T. Knapp.

Signed Keith Murdock

TABLE OF CONTENTS

PREFACE

It has been a source of enjoyment and satisfaction to make this research on centrifugal pumps inasmuch as the problem had previously presented itself to me while actively engaged in engineering work, but no solution had at that time been obtainable.

I wish to express my appreciation to Dr. Robert T. Knapp and George Wislicenus of the Mechanical Engineering Department for their kindly interest, valuable suggestions, and cooperation in making available pumps and other equipment sometimes to the inconvenience of Mr. Wislicenus.

Expression of appreciation is also due Mr. Frank E. Holmes of the Department for his valuable suggestions and aid in assembling and perfecting apparatus.

Keith Murdock

INTRODUCTION

Several years ago the writer conferred in settling the problem of changing the operation of a pump delivering 5500 gal. per minute at 168 feet head to that of delivering 3000 gal. per minute at 146 feet head. The driving motor was a 2200 volt A.C. synchronous motor.

The driving speed could not be altered, and the diameter of the impeller could be cut down only a certain amount before the delivery head would be below that required. At the lowest permissible head of operation the delivery was still excessive unless throttling was resorted to. The head delivery curve was so flat at that operating range, throttling of considerable extent would have to be resorted to in order to have sufficient delivery head leaway to meet the slightly fluctuating operating conditions.

The writer made the suggestion that the impeller be altered by blocking off part of the ports, having in mind that such an alteration would permit the pump to continue delivering at sufficient head to meet the requirements, but the capacity would be reduced to the desired amount without the necessity of throttling. It was hoped to gain a substantial savings in operating power by such a procedure.

Inquiry was made of the pump manufacturers to get all possible information concerning any such alterations as might be made to so change the operating characteristics of the pump. They had no information on that phase of the subject they said, and suggested a different impeller. Inasmuch as the change was deemed to be but temporary, it was thought inadvisable to purchase a different impeller, and further, inasmuch as the suggested alteration of the impeller was apparently in a strange field with no information available on the subject, and continuity of service of the pump was of paramount importance, it was thought inadvisable to alter the impeller other than turning down the diameter as far as the delivery head would permit, a good margin of safety being necessary on account of the almost horizontal characteristic of the head-delivery curve at that operating condition.

Due to the experience recounted the following research was conducted to obtain the information desired concerning the alteration of the impeller of a centrifugal pump, such that in the field the impeller may be readily and cheaply altered so the pump will deliver more economically, with the desired factor of safety, at operating conditions of lesser capacity than those for which the pump was originally designed.

OBJECT

The object of this research is to determine what the characteristics and efficiencies are of a centrifugal pump with its impeller altered in various ways compared to its characteristics and efficiencies when operated with no alterations, at various speeds. Thus a full family of values is obtained for various speeds and interpolation can readily be made for any speed within reasonable range, for any of the operating conditions investigated, both for unaltered and altered impellers.

The practical application of the research is to determine whether or not sealing off full ports, or parts of ports, of a centrifugal pump offers a field method of reducing the capacity of a pump without sacrificing delivery head, at a gain of efficiency over throttling, and to determine the gain or loss of efficiency over throttling; also to determine relative performances and control by speed variation as well as impeller alteration in order to determine what conditions give the highest efficiency at any one operating condition.

EQUIPMENT

Tests were made on two pumps. The equipment used for the first consisted of a Jennings Hytor vacuum pump for a heating system, size E-10, speed 1100 to 2200 R.P.M., manufactured by the Nash Engineering Company. The vacuum end of the pump was dismantled. The pump was direct connected to a General Electric variable speed shunt motor of 5 horse power capacity and speed 1100 to 2200 R.P.M. operating on 230 volts D.C. One interpole of the motor was missing such that a maximum speed of 1580 R.P.M. was attainable only by inserting extra resistance in the field circuit. Excessive armature currents made it inadvisable to operate at higher speeds.

An adjustable rheostat of one ampere capacity and 700 ohms resistance was placed in series with the field in order that exact speed control could be obtained between the various steps of the standard speed control resistance of the motor.

The motor was mounted on ball bearing supports as a dynamometer with a torque arm of 21 inches length, resting on a small pair of platform scales. The scales were made by the Howe Scale Company of Rutland, Vermont, and were graduated to divisions of one one-hundredth of a pound.

The speed was determined by an electric tachometer, model F, manufactured by the Electric Tachometer Company, Tetco type magneto No. 60,071, and Weston Electrical Instrument Company milivoltmeter model 45, no. 19,815.

Rates of flow were determined by means of calibrated orifices and a mercury manometer, or a water manometer, according to the head. All readings above 37 inches of water were read on the mercury manometer. Manometer was of U tube type.

Static pressure was determined by means of a mercury U tube manometer 87 inches high.

A recirculating tank $20\frac{1}{2}$ inches in diameter by 34 inches in height was used. The discharge and suction of the pump were both connected to the tank, the discharge dropping in at the open top.

A throttle valve on the discharge side of the pump was the means of controlling the head of discharge, and amount discharged. A sketch of the setup, Fig. 1, accompanies the curves at the back part of this report.

Tests on this setup of apparatus are distinguished from the setup whose description is to follow by the facts that discharge is given in gallons per minute instead of cubic feet per second, and revolutions per minute are either 750, 1000, 1200, 1400, or 1580, instead of 800 or 940.

On account of the fact that the pump just described was of real low specific speed and capacity, having a delivery of 110 gallons per minute, at a head of 56.7 feet when operating at 1400 R.P.M., which gives a specific speed of 710, it was thought advisable to test a pump of higher specific speed that would be more in line with modern average conditions of pump use. Consequently a Kimball Krogh pump of 986 gallons per minute delivering at 53.5 feet head at 940 R.P.M., which gives a specific speed of 1500, was also tested. The equipment of the second test consisted of the following:

No. 6, type DS, Kimball Krogh centrifugal pump direct connected through a torsion dynamometer to a Fairbanks Morse direct current motor of 20 horse power capacity at 115 volts and 148 amperes, type TR, serial no. 12596E, and 900 R.P.M. speed. A special small series rheostat of .43 ohms resistance and 7 amperes carrying capacity was constructed out of Chromel resistance wire in order to get exact speed adjustment between the steps of the standard speed control rheostat. It was connected in series with the field.

The torsion dynamometer for measuring the pump brake horse power consisted of a strip of steel 18 inches long by $1\frac{3}{4}$ inches wide by 3/16 inches thick, of heat treated chrome vandium structure, held solidly keyed and locked to the pump shaft on one end and the motor shaft on the other end. The pump end of the torsion member carried a 20 inch diameter disc with a graduated scale in degrees of angle, mounted on it. The motor end of the shaft carried a pointer by means of a rigid hollow tube mounted concentric with the line of center of the shafts and torsion member but supported from the motor end only. The pointer was located within one-eighth inch of the disc. A breaker and cam were located at the motor end of the shaft, together with a 6-8 volt storage battery, an induction coil and condenser taken from a Nash automobile, and another condenser of one microfarad capacity connected across the breaker points, and a neon tube closing the high tension circuit of the induction coil. The neon tube was located inside of a large box which covered the rotating disc and pointer. The combination acted as a stroboscope lighted dynamometer.

The electric tachometer was the same one already described.

The rate of discharge was determined by means of an 8 inch Venturi meter tube No. 1627 which has a 4 inch throat and square root scale mounted on a mercury U tube such that the reading is direct in cubic feet per second. As a consequence, all tests on this pump are recorded and plotted with discharge in terms of second feet. This detail will serve to differentiate the tests between the two pumps.

Static pressure was measured by means of a mercury column graduated in terms of feet of water. Suction pressure likewise was measured by means of a mercury column graduated in terms of feet of water.

A throttle valve on the discharge pipe served as a means of control for regulating the test. The suction was taken from the sump and the discharge returned to the sump.

Figure 17 shows pictures of this pump setup.

An account of the procedure of test with the small "Nash" pump is as follows:

PROCEDURE

The first step of preparation consisted of dissembling the pump, removing the vacuum runner, making a spacer to replace it and assembling the pump again as a simple centrifugal pump. The unused openings on the air end then had to be blocked off.

The dynamometer arm on the motor then had to be assembled, measured, and the motor in general put in running shape.

The piping for suction and delivery, and a recirculating tank were then assembled. The first run showed that much air was going into the suction due to the excessive turbulence caused by the high velocity discharge stream. This difficulty was overcome by putting a board partition down the center of the tank from one inch below the top of the water to the very bottom. The discharge went into one side and the suction came out of the other side. A small mesh wire screening was also placed over the lower part of the suction side and a series of baffle plates added as well. This served very well to provide water at the suction of the pump, both free from air and excessive turbulence.

Two orifices were then made, one for low and reasonably high discharges and one for extra high discharges. The outside diameters were 2.06 inches and the inside diameters were respectively .937 inches and 1.344 inches. The down stream sides were beveled off. A holder was made out of a half coupling, to hold the orifice plate tightly to the end of the discharge pipe. The dynamic pressure opening was tapped at 4-5/8 inches above the end of the discharge pipe.

Two manometers were then made of glass tubing mounted on a board . They were filled with mercury, and wooden scales were attached for reading in terms of inches. The dynamic manometer was 70 inches effective height. The static manometer was 87 inches effective height.

The static manometer was attached to the discharge side of the pump below the throttle pipe such that its elevation was one inch below the water level in the suction tank.

The dynamic manometer was attached to the tapped location 4-5/8 inches above the orifice plate. On low discharges a 37 inch water manometer, or stand pipe of glass, was used instead of the mercury manometer.

All manometer readings were made directly in inches. The mercury readings were then corrected for water head on the pump side, and elevation difference, and finally recorded as feet of water for static pressure and inches of water for dynamic pressure.

A section of the discharge piping, and the attached manometer, were then set up discharging to a tank on weight scales and the small orifice was calibrated over a wide range. (See the attached data sheet for figures). The calibrated orifice showed a discharge coefficient of 3.12 (times the square root of the head in inches of water for total discharge) as compared to 3.14 as calculated from the formula and coefficients determined by Spitzglass of the Republic Flow Meter Company. The check was so close that the large orifice was not calibrated but the coefficient was calculated directly.

The piping and manometer were installed again on the test pump and a series of runs was made at speeds of 750,1000, 1200, 1400, (1550 one only) and 1580 R.P.M. At each speed the pump was operated from no delivery shut off to maximum discharge, over a series of intermediate values of discharge such that a curve of discharge against head could be plotted. At full impeller a maximum speed of 1550 R.P.M. was the limit due to the static manometer. This run was corrected to 1580 R.P.M. inasmuch as the following maximum speed runs were all made at 1580 R.P.M.

At the end of the first series of tests the pump was dissembled and two ports of the impeller were sealed off by means of soldering a thin sheet of shim brass over the port opening at the periphery of the impeller. Opposite ports were covered in order not to destroy the balance. See Fig. 2, alteration "A" type. The surface was carefully smoothed after soldering, and the pump assembled and the same series of tests repeated as before. This test was called the 2/3 impeller test type "A" alteration. On the whole, this test was not so satisfactory as the others because of a vibration which at times appeared in the pump. There also seemed to be a slight drag which at times caused fluctuation. When the pump was dissembled to prepare the ports for the next test, a small stone was found lodged in one of the open ports near the driving side. It undoubtedly affected the test some, but not enough to warrant a reassembly and retest. It will be noticed that the horse power curves at low delivery on this test are excessive and not uniform with the rest of the curve at higher discharge.

The third test was made with four of the six ports blocked off and was called the 1/3 impeller test type "A" alteration. Symmetry was observed in preparing ports and the same series of tests duplicated. Readings consisted of R.P.M. taken with an electric tachometer and adjusted to the exact speed of the test by means of the secondary field rheostat. This was done after the approximate delivery had been set by the throttle valve. The weight of the dynamometer arm was then read, together with the static and dynamic pressures. The included data sheets show the system of testing.

The fourth test was made with three of the six ports, 120 degrees apart, sealed off and three left open. This was known as the 1/2 impeller test type "A" alteration. All of the tests made with sealed off ports tended to show a slight vibration at higher speeds but not enough for any concern.

It might be well to note here that two tests were run with a full open impeller. The first was made before securing the secondary field rheostat. Consequently the speed varied considerably between shut off and full discharge opening. Also much turbulence and air bubble trouble was experienced in the tank, because all of the refinements already described were not then installed. Rather than correct the rather faulty data thus obtained, another set of tests was made, and the first test merely used as an experience from which was drawn the information that resulted in changes and refinement of apparatus sufficiently that all subsequent tests were satisfactory.

Upon the completion of the test on the impeller, altered according to type "A", just described, a second series of tests was made with the impeller altered in a manner called type "B". This consisted of closing off a portion of each port on the follower side of each vane. See illustration on Fig. 2. The first test was made with all of the port openings 2/3 blanked off with shim brass soldered over the opening, and carefully trimmed and smoothed. It is called the "1/3 impeller test type "B" ".

The pump was again taken apart and part of each blanking strip shaved off until only half of each port remained covered. The pump was assembled and the test run known as the "1/2 impeller test type "B" ".

Again the pump was taken apart and more of the blanking strips shaved off until but 1/3 of each port opening remained covered. The pump was again assembled and the test run known as the "2/3 impeller test type "B" ".

All of the type "B" tests operated smoothly. The power used and water delivered were both greatly reduced as will be discussed later.

A third series of tests was next run, known as type "C" tests. They consisted of blanking off the impeller port openings on the other side from type "B", that is on the driving side of the vane. See Fig. 2 for the illustration.

In general, the same procedure of altering and testing was followed as in type "B" except that the pump operated quite differently. Much more power and much stronger delivery of water were noted. The pump vibrated much more than at any former test, in fact so much vibration was obtained at a speed of 1400 R.P.M. that higher speeds were inadvisable with some of the impeller alterations.

A one-half blanking was tried first. The more severe operating conditions caused a portion of four of the blankings to be torn away such that in the end the pump was really operating at 59% port opening type "C" rather than 50%. Details are noted on the data sheet. The delivery characteristics were quite different from what was expected, the pump actually maintaining much higher head and delivery than had been expected. In order to get the desired low delivery three quarters of the port openings were next blanked off rather than 2/3 as in previous tests. Fairly smooth operation was secured at this test.

From the 1/4 impeller test type "C" the blanking strips were shaved of such that 2/3 port opening was obtained and the final test of type "C" series was run.

A fourth series of tests was next made, known as type "D". The impeller was altered by completely blocking off three of the ports in symmetrical sequence at the inside or suction entrance. It was very difficult to solder, or otherwise fasten the blanking strips in place. A special soldering copper had to be devised for the purpose. During the test one of the blanking strips came loose such that it became entirely ineffective and the major part of the test was run with but two ports blocked. This test only was made with type "D" alterations, inasmuch as it practically coincided in results with type "A" alterations.

It was at this stage of the research thought advisable to investigate a larger more standard size pump, to see how results from it would check results obtained from the "Nash"pump. The "Nash" pump was very well made and efficient for its size and purpose. The impeller was approximately $10\frac{1}{4}$ inches in diameter and had six port openings. At the periphery the openings each measured 1/8 inches wide by 3-1/8 inches long. The impeller was of special built up construction.

The Kimball Krogh pump next tested had an impeller of 14 inches effective diameter and port openings $1\frac{1}{2}$ inches in width. Narrow vanes 1/8 inches thick were the only separation between ports. The impeller was symmetrically divided into six port channels.

The pump was new and had not been previously operated. The procedure of test with this pump is as follows:

PROCEDURE WITH KIMBALL KROGH PUMP

Manometers for measuring static and suction head had to be constructed and connected. Good instruments for permanent future use were made and installed. The photograph, Fig. 17, shows the two manometers. Large diameter containers were installed on the heavy pressure side of the manometer and 9/32 inch bore pyrex glass tubing was used for the visible leg of the manometer. Special scales were then constructed and graduated such that the total differential, corrected for the water leg superimposed on the mercury on the pressure side, was read directly on the manometer scale in terms of feet of water. The piping connections to the pump were made permanent, with 1/8 inch iron pipe and the manometer permanently located. The zero position correcting for the constant water head due to the slight difference in elevation between the working zero of the manometer and the horizontal center line of the pump, was determined and the scale zero set at that point, after which the scale was nailed into that position permanently. As a consequence both pressure and suction head are read directly in terms of feet of water and no correction is necessary.

The next task consisted in getting the torsion dynamometer into working shape. The calculations for its size, strength, deflection, etc., had already been made by its designer, Dr. Knapp. Mr. F. E. Holmes had already made the torsion member and designed and made some very effective and accurate connections for fastening the torsion member to the shafts of the motor and pump in a rigid concentric manner that would permit of no loosening or shifting of position after it was set up. Direct angular deflection by torsion stress was the only relative motion permitted the two shafts by the torsion member.

The torsion member with its connections, attached to stub shafts, was taken to the materials testing laboratory and mounted in the torsion testing machine. There it was calibrated through an angle of twist of $30\frac{1}{2}$ degrees. Fig. 26 gives the calibration curve.

Upon mounting the torsion assembly on the motor and pump shafts again, extreme care was taken to mount it in exactly the same manner and dimensions as it was mounted in the test machine. This was accomplished by having first marked all parts before dismounting from the pump in the first place before testing.

While the torsion member was dismounted and free from all external strain, two long straight edges were clamped, one at each end, but on the same side, and at right angles to the long axis of the torsion member. The straight edges thus defined a plane which was really plane, rather than warped in any manner. Upon mounting the member in place between the pump and motor again, the straight edges could again be clamped into the same position and the machines set to make them define a plane again. This would put the torsion member under zero stress and the zero position of scale and pointer be set.

The problem of devising a scale and indicator for the rotating torsion member was next attacked. The stroboscope had already been assembled such that it worked, although in a very erratic and fluctuating manner. The amplitude of vibration of the stroboscope beam was about three degrees average value. Erratic shifts of larger degree were at times noticed. A good base work had already been accomplished, including the assembly of battery, wiring, condensers, induction coils, breaker, neon light, and cover box. The problem of refinement remained to be worked out.

In brief the setup is as follows: Current from a storage battery flows through an induction coil and a breaker mechanism and back again to the battery. A condenser is connected in parallel across the breaker points, in order to damp out the arc at break. The breaker points, are held open all but about 30 degrees of each revolution. A notch in the holding collar on the motor end of the torsion member serves as the make and break cam of the breaker mechanism. A 60 degree angle in the notch makes a sharp break. This causes a surge through the induction coil, the high potential circuit of which is closed through the neon tube located in the cover box over the torsion member scale and indicator. At the break the neon tube lights instantly and then extinguishes. The lightflash is supposed to occur in the same position of each revolution. Then due to retinal fatigue and persistance of vision, the eye looking at the rotating member in the light of the flashing neon tube, sees the rotating member as though it were standing still. Any relative displacement of the disc carrying a graduated scale in degrees of arc, with the indicator carried on the other end of the torsion member, is readily discerned. The reading thus obtained is the angle of twist due to the power transmitted by the torsion member from the motor to the pump. From the calibration, the inch pounds torque for each degree twist is known. The horse power is readily calculated from the torque and the R.P.M. value. The formula is given on the data sheet of the calibration curve.

The only difficulty experienced was that the break did not always occur in the same place. Thus a rapidly vibrating appearance was given to the scale and indicator. This was largely due to the fact that the arc of break was not fully extinguished at the time of break. This persistance of arc was not uniform thus the effective break, which did not occur until the arc was fully extinguished, was not uniform either. A larger capacity condenser and smaller ratio induction coil helped some but not sufficiently to give a clear cut appearance sufficiently definite to be read accurately.

In order to solve the problem a scale was devised that could be accurately read even though vibration were quite intense.

The photograph of the disc and indicator in Fig. 17 shows the scale. Each division is a section of arc 1/4 inch in length and displaced circumferentially from each other by 1/5 of a degree of arc. Each fifth line was marked distinctly wider and in red. These were the full degree divisions. The fifth degree, or center, marker had a small pilot marker at the same radius placed about one inch away from it. This served to identify each fifth degree. Divisions of arc in sections of ten degrees were made from circumference towards the center, then the pattern was repeated. The tens, fives, and units were thus easily distinguished and the units divided into fifths of one degree. Thus by interpolation, readings could be made to the nearest tenth of a degree.

The indicator was made with the lead side radial and the width exactly 1/4 inch wide, or the same width as the divisions of the scale.

It was so set that at zero it exactly masked the zero division. Likewise at any other division it would exactly mask the division to be read. A small part of the preceding and following division could be discerned, thus the missing division could be identified and the reading recorded. In cases where the vibration was so strong the indicator could not be seen, the graduations, being sections of arc, would repeat on themselves and appear as longer lines. The line masked by the indicator, which was light in color, would be missing and adjacent lines, being partly masked, would be faint. Thus the location of the indicator could be determined by the faint section in the scale and the missing line, or center point of the faint section, was the reading to be recorded.

Later it was discovered that by reducing the impressed voltage across the breaker points to 2 volts, a steady break could be obtained. The masked line and indicator position could thus be readily determined. Readings could readily be made to one-fifth of a degree of arc and estimated to the tenth of a degree of arc.

The next problem was to secure constant speed at the points of test. Two rheostats were in parallel on the control of the field circuit. The speed variation was too small, so the rheostats were reconnected in series, thus giving a range from 800 R.P.M. to 1000 R.P.M. A test was made to determine the amount of resistance between the steps of the main control rheostat. A secondary rheostat of Chromel wire was then made of a capacity equal to the largest step. The secondary rheostat was adjustable by a slide contactor. Thus exact speed control could be obtained by means of the adjustments of the primary and secondary field rheostats.

With the apparatus all set up and perfected tests were run on the pump with full impeller. The torsion dynamometer scale and indicator appeared to be vibrating strongly due to the stroboscope. Readings were made as above indicated. A test was made at 940 R.P.M. speed and also at 800 R.P.M. The upper speed of 940 was chosen in order not to exceed 30 degrees distortion of the dynamometer, at full loading.

The strohoscope was further perfected and the tests repeated on the pump with the impeller unaltered. A very satisfactory check of values was obtained. See data and Fig. 18 for details of these duplicate tests.

The cover was then removed from the pump and the impeller altered to 1/3 capacity according to type "A" by blocking off four of the six ports at the periphery. Calculations were first made of the maximum possible stress that would come on the blocking strips. It figured 110 lbs. per square inch for centrifugal force and a possible 700 lbs. per square inch for a sudden possible surge pressure of full head. The total figured 810 lbs. per square inch. The blanking sections were made of 1/16 inch sheet steel and held in place by two 12/24 steel machine screws on each end, screwed into holes drilled and tapped into the ends of the impeller vanes. The center was held with three such screws, or seven screws in all for the strip blanking two adjacent ports.

Tests were run at 800 and 940 R.P.M. with the impeller thus altered to 1/3 capacity type "A" alteration. Because the curve at 940 looked irregular the 940 R.P.M. test was duplicated. Both values are given.

The cover was again removed and the strips cut off such that two ports only, on opposite sides, remained blanked. The pump was re-assembled and tests run at 800 and 940 R.P.M. speeds. This was known as the 2/3 capacity test type "A" alteration.

The next alteration consisted of blocking three ports and testing again as before. This was known as the 1/2 capacity test type "A" alteration.

The pump was again dissembled and the blanking strips all removed. The next alteration consisted of placing two bands of brass stripping 3/8 inches wide around the periphery, one at each side. This closed off 50% of the impeller opening. The ends of the stripping were fastened by interlocking, soldering, and finally placing a screw through the interlocked ends and into a hole tapped into an impeller vane. The strip was fastened at each vane with a rivet. This alteration was known as type "E" 1/2 capacity.

The pump was again assembled and tested. A capacity and head greater than the unaltered impeller was obtained at the 940 R.P.M. speed.

On account of the failure of the banding of 1/2 the impeller opening to reduce the pump capacity the next alteration consisted of blanking off 5/6 of the impeller opening by means of two strips each 5/8 inches wide banded around the outside portions of the periphery according to type "E".

The pump was again assembled and tested at 800 and 940 R.P.M. speeds. The first test at 800 R.P.M. was made with the suction valve but partly open. The test was repeated with the suction valve all the way open. Both values are plotted. They coincide very satisfactorily.

This test completed the series as projected for this research.

A discussion of the curves will next be given followed by the data and curves themselves.

DISCUSSION

Type A Curves - Nash Pump

From the curves it is evident that the characteristics and efficiencies are of much the same order for different speeds when the impeller is the same. The more the ports are blocked off the steeper becomes the curve of discharge against head, and the lower becomes the overall efficiency of the pump. This latter is reasonable since leakage will remain constant as will also friction to a large extent, but discharge is reduced. With effective work reduced and lost work remaining essentially constant, effiency must drop off. The larger the pump the less noticeable this tendency would be.

Fig. 3 shows the series of head-discharge curves for all speeds and all conditions of impeller altered as indicated. Fig. 4 shows the series of brake horse power for all discharge values and head values of Fig. 3. The brake horse power of any point on Fig. 3 can be secured by locating the corresponding curve on Fig. 4 with reference to speed, impeller, and discharge.

Fig. 5 takes one particular series at the constant speed of 1400 R.P.M. and shows the brake horse power and overall pump efficiency for each impeller condition. This is illustrative of change of pump characteristics with change of impeller.

Fig. 6 shows a single series of impeller changes operating at a constant speed of 1580 R.P.M. and delivering from a theoretical pipe line at an elevation difference of fifty feet and a long uniform pipe that had a pipe friction of 19.6 feet at a full discharge of 140 G.P.M. The friction was assumed to vary as the square of the velocity of flow. The head-discharge curves and the brake horse power-discharge curves are copies of the corresponding curves on Figs. 3 and 4. The efficiencies are the total overall efficiencies of pump and pipe line and are secured by dividing the water horse power at delivery by the corresponding brake horse power read from the curves. This curve is of interest in showing what could be done in the field at trifling expense in altering an existing pump runner to deliver at lower capacity, if conditions were such that it was inadvisable, or impossible to change either the diameter of the runner or the speed of the driving motor.

Fig. 7 shows a copy of the same theoretical pipe line, and part copies of all of the curves on Fig. 3 above the theoretical discharge head curve. Efficiencies were figured the same as before only values of brake horse power were taken directly from Fig. 4.

These curves are of interest in showing what could be done if the peripherial speed of the runner as well as port openings were subject to change. It is readily seen that the highest efficiencies are obtained for any single discharge by altering speed only. The difficulty of this method is that as the speed is lowered the curve becomes progressively more flat. The practical consequence of this is that anything which would raise the head of delivery for any cause would also seriously cut off the delivery. The altered impellers provide steeper characteristics at reduced efficiencies over speed reduction only. They would provide ability to overcome added resistance with but little drop in delivery. For any single condition the most economical runner and speed can be picked from these curves, or interpolations between them.

Type B Curves

Figs. 8, 9 and 10 show the same kind of curves for type "B" alteration as Figs. 3, 4 and 5 do for type "A". It will be noted that the head-delivery-curves take a steeper characteristic for the same proportion of impeller blanking. Values taken from a set of curves at constant speed, and delivery head from Fig. 3 plotted as ordinates against proportion of impeller being used, as abscissa, fall on a straight line which would seem to intersect the horizontal axis before the origin was reached. Similar values plotted in similar manner for Fig. 8 would fall on a curved line, curving upward, which also projected, would seem to intersect the horizontal axis before the origin was reached.

For a set of curves at constant speed, Fig. 3, if the value of discharge be taken at the point of maximum efficiency for each curve and plotted as ordinates against the same abscissa as before, the points will again fall on an approximately straight line which, towards the vertical axis, would have to take a downward curve in order not to intersect the vertical axis above the origin. Similar values plotted for Fig. 8, in similar manner, take the form of a curve, less steep than before but curving upwards. However, towards the origin it also would have to curve downward in order not to intersect the vertical axis above the origin. A set of curves so plotted from values taken from any set of constant speed curves on Fig. 8 could be used to determine beforehand the proportion of each impeller port to be blocked in order to secure a pre-determined delivery at a certain head or a pre-determined delivery at the point of maximum efficiency, in which case the head will be somewhat different.

In general, it may be stated that type "B" blocking is less efficient than type "A" blocking. A comparison between values of curves Fig. 5 and Fig. 10 indicate this to be the case. However, type "B" blocking is entirely flexible over the entire range of port opening, whereas type "A" blocking is limited in flexibility by the number of ports in the impeller.

Fig. 11 shows the possibilities of the two types of blocking. It can be seen that at certain capacities type "B" blocking might be used to advantage, in case flexibility of effective impeller speed were not permissible.

Figs. 12, 13 and 14 show a set of curves for type "C" blocking corresponding to Figs. 3, 4 and 5. It is evident that this type of alteration causes the water to leave the impeller, with a larger radial component of velocity than normal, just as type "B" blocking causes the water to leave with a smaller radial component than normal. Type "C" curves consequently have very much flattened characteristics. Strong cross currents apparently are set up in the volute of the pump. Certainly vibration is increased and horse power is increased even at lowered delivery, above that of full normal operation, as shown by Fig. 13. This blocking is apparently of no practical importance unless it were desired to increase the capacity and delivery of a pump regardless of the lowered efficiency and added power used. The characteristics shown by these three figures are very interesting. It will be noted that the lowest set of curves is for 1/4 impeller capacity rather than 1/3 capacity as in the two previous types discussed. This was done in order to more clearly show the directional trend of this type of blocking.

Fig. 15 shows the results of type "D" alteration. When the impeller was examined at the finish of this test, one of the blanking strips was loose except for one end. It was totally ineffective as far as its original purpose was concerned. Apparently from the 1000 R.P.M. curve, the break occurred towards the end of that run.

Fig. 16 shows a collection of curves each with 2/3 of the impeller open. Each type of blocking so far discussed is shown. It is evident that but little difference exists between blocking total ports from the inside or the outside as type "D" and type "A". The difference shown is within the range of experimental error. However, such difference as does appear is in favor of blocking according to type "D". The difficulty of installing this type of alteration is a disadvantage in its use. That is why but one series of tests was made, inasmuch as no particularly different characteristic from type "A" was shown by the test made.

DISCUSSION OF THE KIMBALL KROGH PUMP TESTS

Fig. 17 shows photographs taken of the pump, the torsion dynamometer, and the manometers. The high efficiency short radius elbow at the suction of the pump is of interest. It is made efficient by means of a series of deflecting vanes installed along the plane of the diagonal junction of the horizontal with the vertical piping.

The lower scale shown on the dynamometer disc was used for research in developing the scale that was finally used. It is of no further use.

Fig. 18 shows the head delivery curve, the brake horse power curve and the efficiency of the pump operating at normal or unaltered impeller conditions. Two separate and complete tests were made at 800 R.P.M. speed and also at 940 R.P.M. speed. The excellent coinciding of results shows the experimental setup to be reliable. An efficiency in excess of 77% was obtained. The manufacturer's guarantee was 78%. At higher speeds this value would likely be obtained, inasmuch as the tests show a gain of over 1% between 800 and 940 R.P.M.

The characteristic curves show the pump to be of good design. Flattened discharge delivery curves, and high efficiency over a wide range indicate considerable flexibility of operation at little sacrifice of efficiency. An efficiency of 77% with this size of pump is very good, although not at all uncommon. Some designs will actually give better efficiencies but they are seldom met with in practice at this time.

Fig. 19 shows the characteristic curves of the pump when the impeller is type "A" altered to 2/3 capacity. The curves show that the pump still retains much of its desirable characteristics although the capacity is reduced and it is made of lower specific speed. The total efficiency is lowered from 77 to 71%.

Fig. 20 shows the same type alteration reducing the pump to 1/2 capacity. The head-delivery curves are much steeper. The efficiency curves are still relatively broad and flat on the top, although the peak is reduced to 65%.

Fig. 21 shows the characteristic curves at 1/3 impeller capacity having four ports blocked off according to "A" type alteration. A rather peculiar shape was shown in the efficiency curve for 940 R.P.M. speed so the test was duplicated to check the values. Both tests check within close limits as indicated by the experimental points on the curves. The head-delivery curve is very steep and the efficiency curve becomes sharper at the top. The maximum efficiency drops to 59%.

The same general reasons for reduction of efficiency with progressive alteration apply for this pump as stated in the discussion of the Nash pump test.

Fig. 22 is a composite of curves for the Kimball Krogh pump similar to Fig. 7 for the Nash pump. They show the progressive gain in overall efficiency of the combination of pump and pipe line when friction is a fair proportion of total head. A condition comparable to this is often found in industrial plants. A lower static head was chosen because of the lower delivery head of the larger pump. The pipe line friction was retained the same value as before.

It is readily seen that for a condition such as the one chosen, a very substantial power savings can be made for lowered delivery conditions by means of altering the impeller. As before the most efficient change is to lower the speed either by a lowered R.P.M. rate or by cutting the impeller to a smaller diameter and retaining the same R.P.M. speed. The limiting condition as before is the increasing flatness of the head-discharge curve. A point is generally reached where the leaway between operating head and shut off head is too small for safe operation. Or for construction reasons the R.P.M. rate cannot be changed, neither can the impeller be turned smaller for one reason or another. Blanking of impeller ports gives a good solution to the problem, by means of procuring a steeper characteristic that makes for sufficient safety leaway in operation where continuous flow must be maintained although actual quantities may vary somewhat.

It is notable that the percentage drop in efficiency resulting from altering the impeller in type "A" fashion is decidedly less for the larger than for the smaller pump. Average figures for the two highest speeds give the efficiency of alteration as follows, dividing the maximum efficiency of the altered impeller by the maximum efficiency of the unaltered impeller:

	2/3 impeller	$\frac{1}{2}$ impeller	1/3 impeller
Nash Pump	83.7%	74.0%	57.8%
Kimball Krogh Pump	91.8%	83.9%	74.2%

This indicates that the larger and more efficient the pump the higher will be the overall efficiency under altered conditions. It is also evident that a comparatively efficient pump can still be had even under altered impeller conditions. However, it is to be stressed that such alteration should be resorted to only as an emergency measure, or as a temporary measure when it is inadvisable to secure a different impeller, or pump in its entirety, to meet the changed conditions.

Fig. 23 shows the characteristic curves of the pump with its impeller altered by means of two bands around its periphery such that 1/2 of the discharge area is blanked off. The alteration is known as type "E".

The overall efficiency is lowered by this change and the head-discharge curve is given a rising and flattened characteristic. The capacity of the pump is increased rather than diminished. This will be discussed further with regards to Fig. 25.

The discontinuity in the 800 R.P.M. test is an actual discontinuity. When it appeared in the data of the first test it was thought that one of the bands must have broken in some manner so the test was stopped and the pump cover removed and the impeller examined. The banding was intact. The pump was reassembled again and the test started from the beginning. At the same position when the readings had been partly taken, just as in the first case, the head of discharge suddenly increased although the speed remained constant. The test was continued and the data plotted as obtained. See the data sheet of this test for further particulars. Conditions of unstable flow within the runner and volute must be the reason for the break.

Fig. 24 shows the results of a radical change made on the impeller through banding it in type "E" manner until but 1/6 of the impeller port openings remained open. Inasmuch as closing off 1/2 the opening area as shown in Fig. 23, failed to reduce the delivery the change was this time made sufficiently extreme that some reduction in delivery must result. Strongly rising, then more sharply falling characteristics in the head discharge curve is the result. Also a decided increase in brake horse power resulted. Two tests were run at 800 R.P.M. as stated in the discussion. The results indicate that the pump will perform equally well on a 13 foot suction head as on an 8.4 foot suction head.

Fig. 25 is a collection of all the curves of type "E" altered tests plotted with the curves of the standard test made with unaltered impeller. The characteristics of type "E" alterations are very similar to those of type "C" alterations. Apparently the small opening caused by banding causes the water to pass through in like manner to passing a thin plate orifice. The increased velocity causes the water to depart from the impeller with a larger radial component of velocity. This must be so since due to the constriction added relative pressure must be built up within the impeller itself with respect to the volute. Since hydrostatic pressure always manifests itself at right angles to the containing vessel, all of the added hydrostatic pressure must manifest itself in giving a direct radial component to the water being discharged from the impeller. There is no deflecting vane section to modify this action as in type "B" alteration. The added radial component thus imparted to the water by this type alteration explains the rising characteristic, and increased head and discharge obtained. It also explains the marked increase of power used, and the lowered efficiency. These two factors are associated with radial discharge from an impeller, due to the inability of a volute case to efficiently transform radial velocity into effective pressure head.

Practically, it would seem that there is no use for this type of alteration unless as in type "C", added head or capacity or both are desired regardless of power and efficiency.

Fig. 26 shows the calibration curve of the torsion member of the torsion dynamometer. It was tested both by adding load and also by taking load off. Very good conformity with Hook's Law is manifested.

ACCURACY AND ERRORS

The tests on the "Nash" pump were subject to error from the following sources: a slight amount of air at high discharge values remained entrapped in the water as it was recirculated through the small supply tank. This tended to cause a somewhat unstable condition, resulting in fluctuating mercury columns in the manometers, making them somewhat difficult to read at time. The water also tended to warm up some, thus changing its specific gravity. Readings as high as 90° F. and as low as 70° F were observed. No correction was made for this difference. The dynamometer was a little sluggish varying over a range of 2 to 3% in its readings. Mean values were taken. The orifices could be relied upon to read accurately within 2%. Particular care was taken to take all readings in the same manner so as to eliminate as far as possible, error in the relative tests. It is believed that the tests were run within an accuracy of 2% error although individual readings may vary beyond that limit.

The Kimball Krogh pump tests were subject to error from slightly different water temperatures on different days, inability to read the torsion dynamometer closer than 4% at lowest readings and 0.6% at maximum readings. A large suction lift tended to make the manometers fluctuate rather widely. Mean position had to be read. A check with a revolution counter on the electric tachometer showed it to be reading 1.0% high. Although this would introduce an absolute error it would not introduce a relative error, between comparative tests. It is believed that the accuracy of the tests in general is within 2% error, although here again individual values may vary more than this. In general a Venturi flow meter can be relied upon at normal rates of flow for which it was designed, to give results accurate within 1%.

CONCLUSION

The conclusions to be drawn from these various tests are:

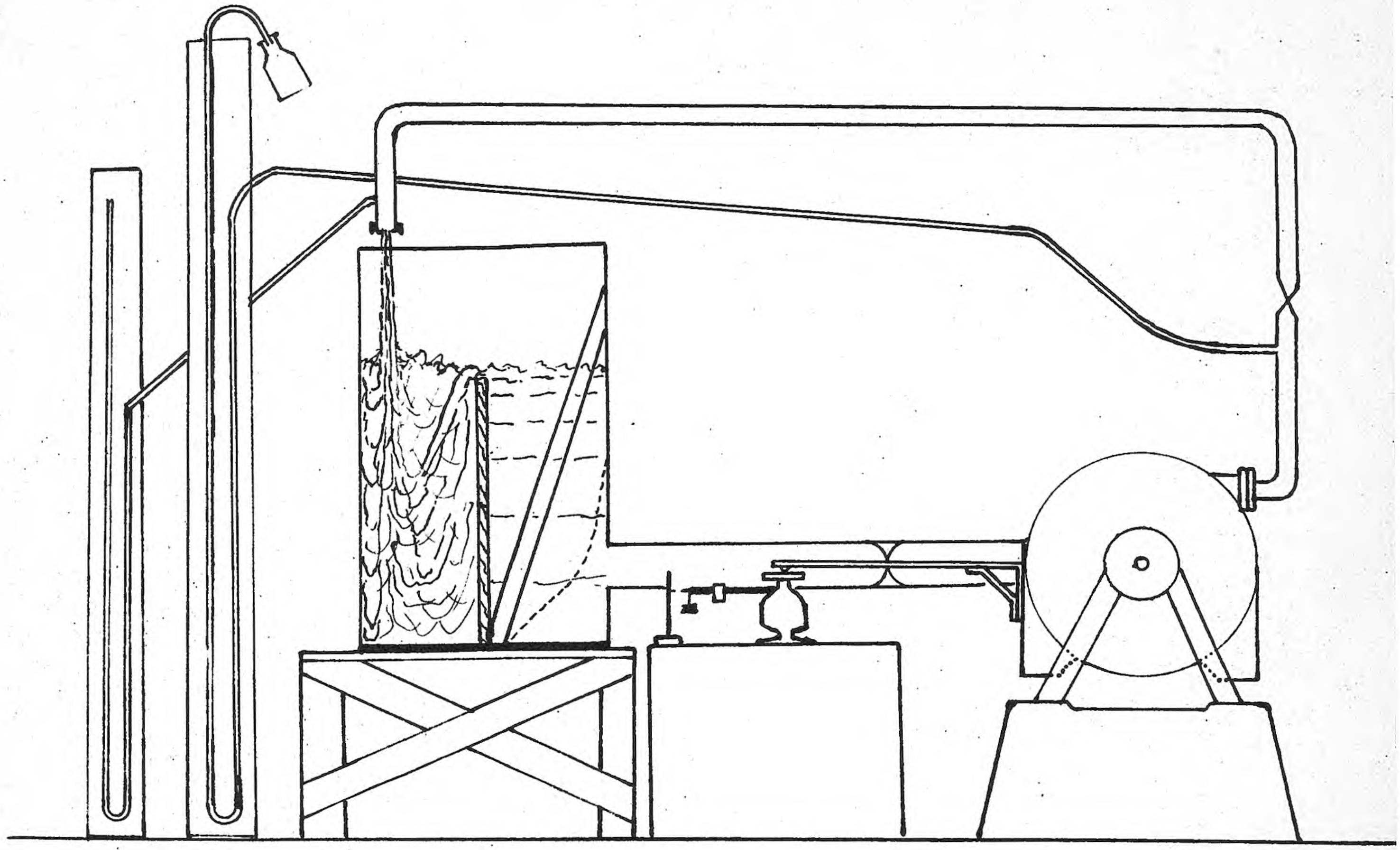
1. Alteration of the impeller of a centrifugal pump by means of closing off impeller ports by either type "A" or "D" or "B" methods, provides a means of decreasing the power input of the pump when it is operating at a lower capacity than that for which it was designed. Type "B" is the less efficient but more flexible method of alteration.

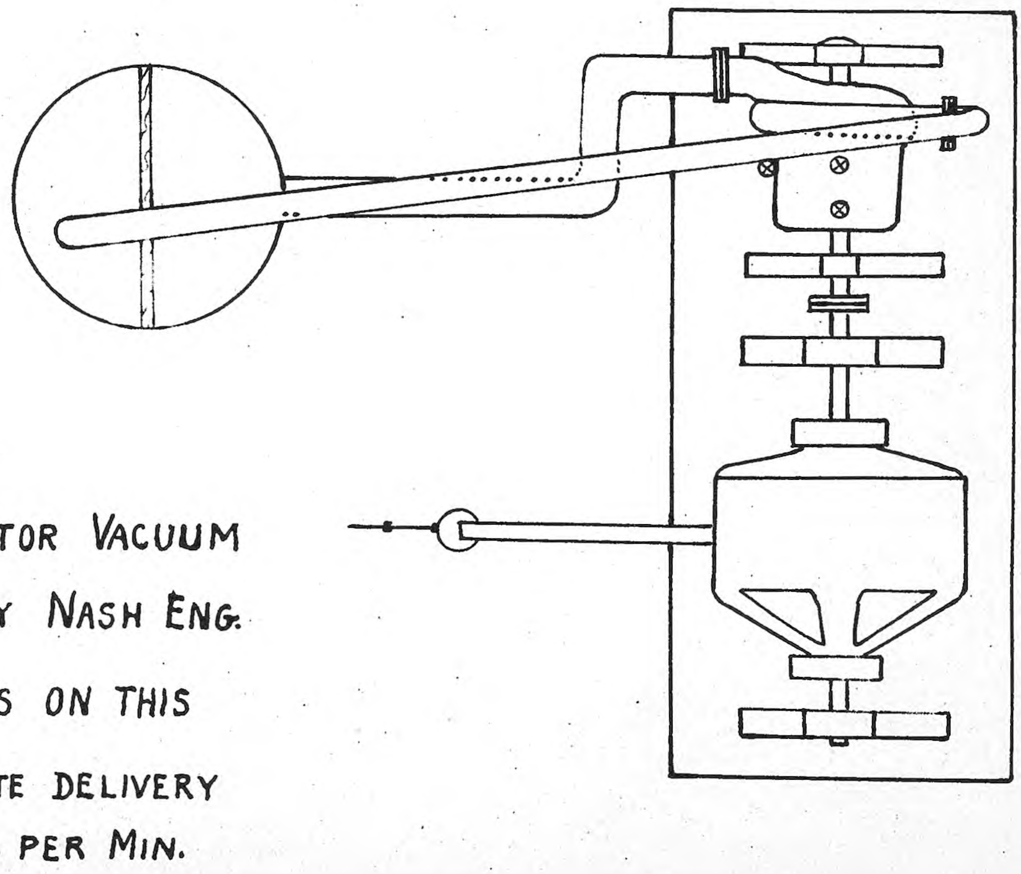
2. Any of the above named alterations are practicable inasmuch as a pump so altered will nevertheless perform essentially as smoothly and as free from vibration as it did before alteration, provided due care be taken not to unbalance the impeller during the course of alteration.

3. Either type "C" or "E" alterations can be used to secure an increased delivery of small amount but at the cost of decreased efficiency and increased power.

4. The main study of this research is revealed by the various curves and interpolations and developments suggested by them. The reader is respectfully referred to them and the data sheets for detailed, as well as general, information. They have been made as far as possible self explanatory.

FIG. 1





JENNINGS HYTOR VACUUM PUMP MFG. BY NASH ENG. CO. ALL TESTS ON THIS PUMP DESIGNATE DELIVERY IN GALS. PER MIN.

FIG. 2

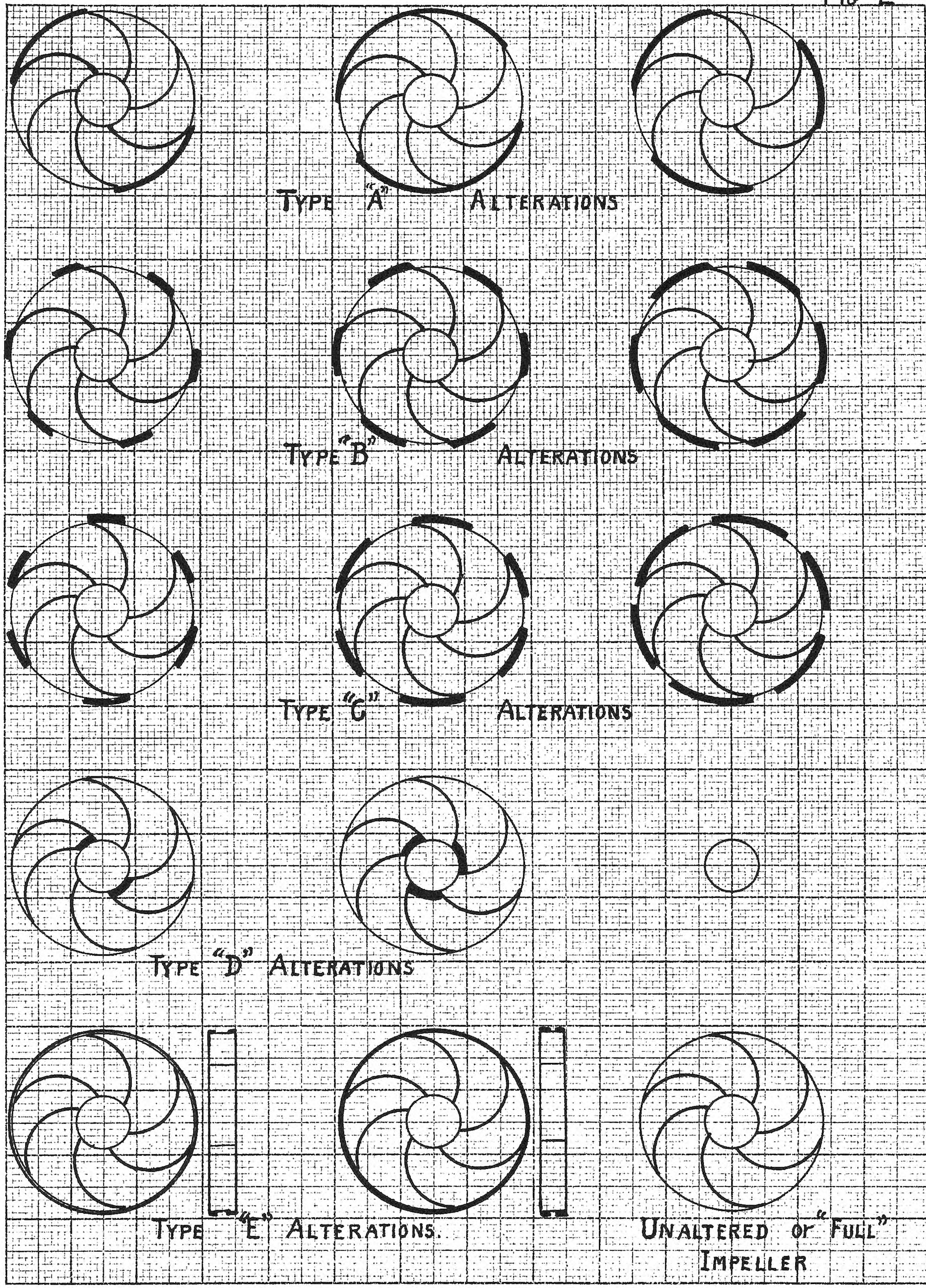

TYPE "A" ALTERATIONS
TYPE "B" ALTERATIONS
TYPE "C" ALTERATIONS
TYPE "D" ALTERATIONS
TYPE "E" ALTERATIONS.
UNALTERED or "FULL" IMPELLER

Fig. 3

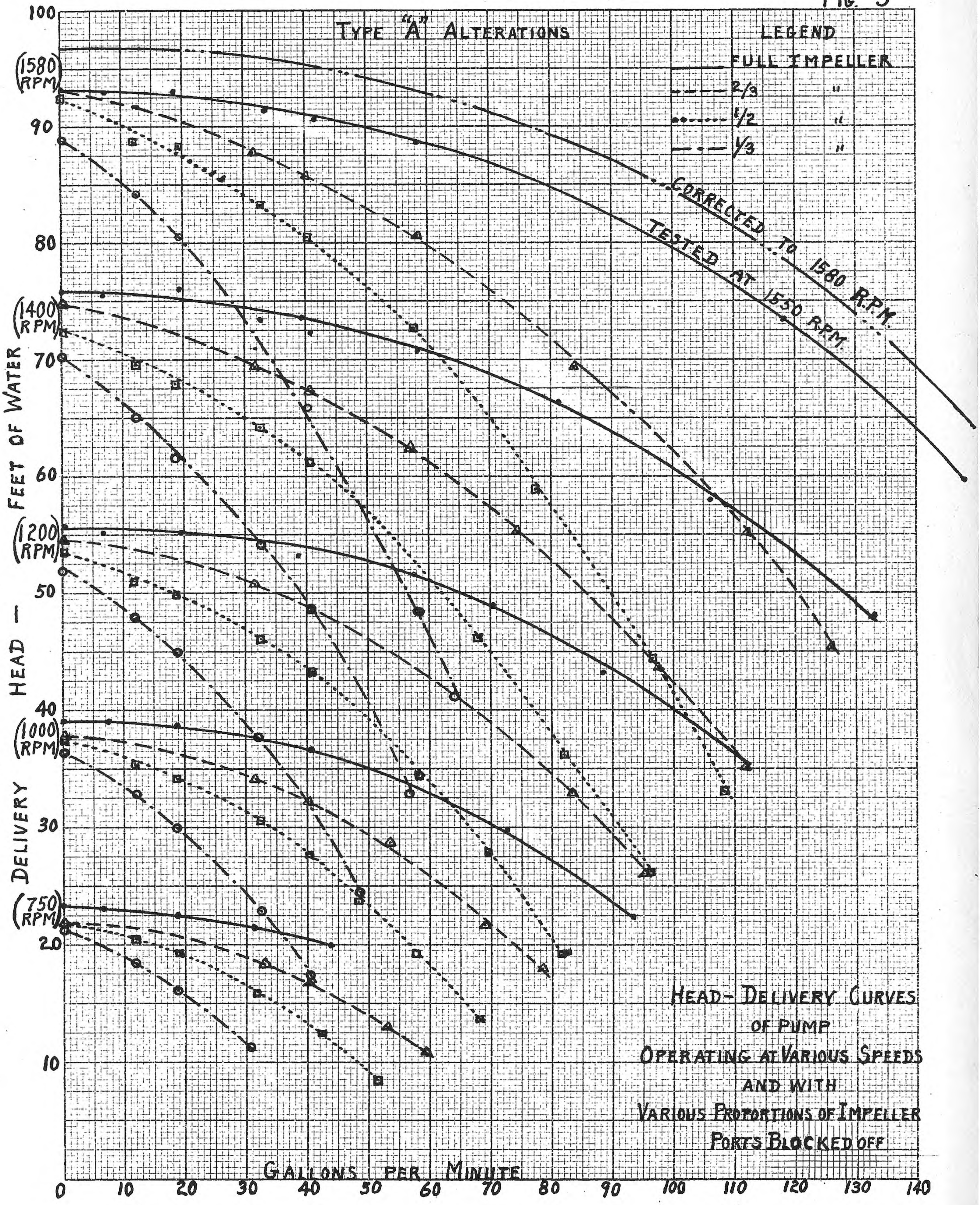
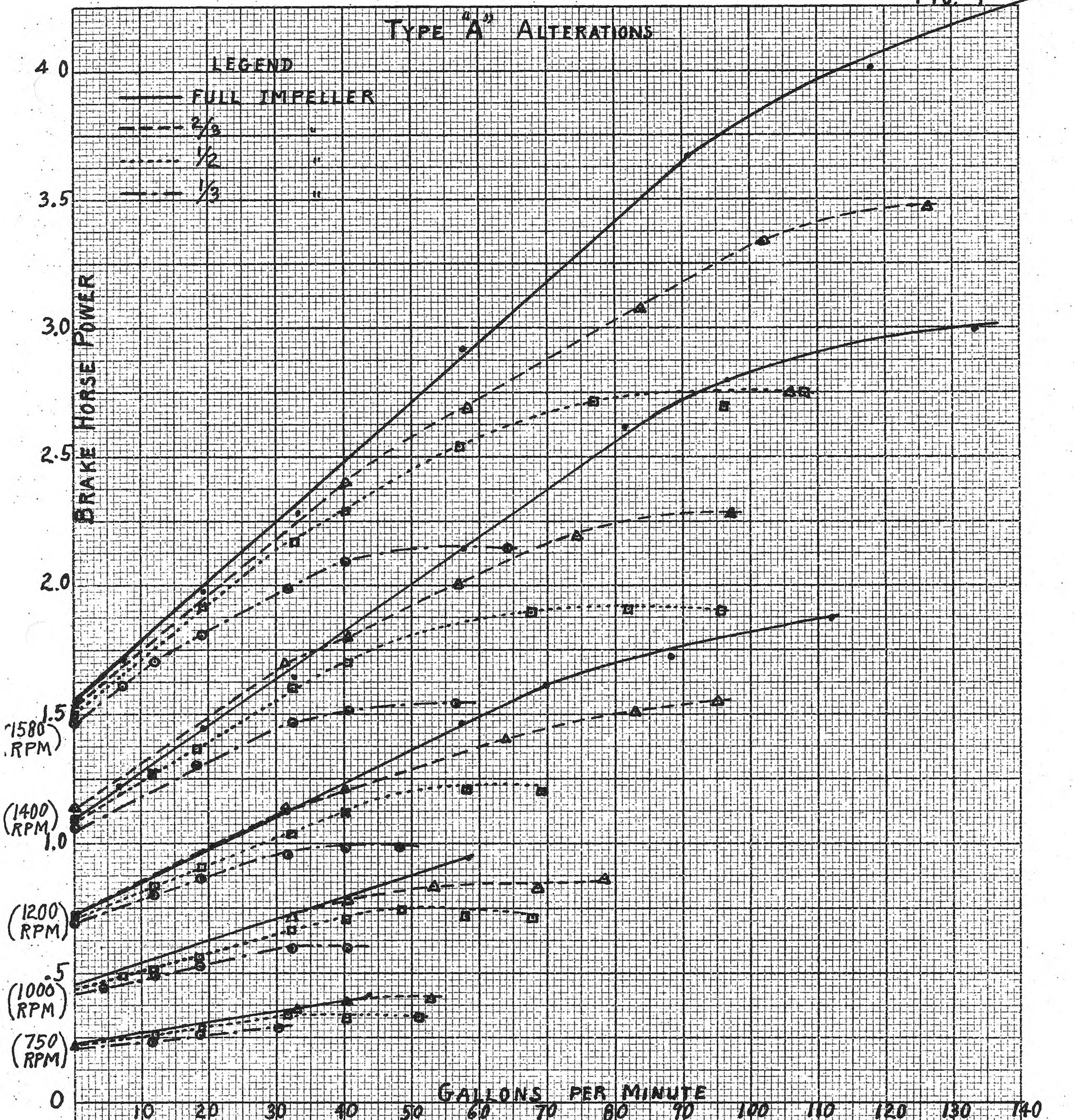

Type "A" Alterations
Legend
Full Impeller
2/3 "
1/2 "
1/3 "
Corrected to 1580 R.P.M.
Tested at 1550 RPM.
(1580 RPM)
(1400 RPM)
(1200 RPM)
(1000 RPM)
(750 RPM)
Delivery — Head — Feet of Water
100
90
80
70
60
50
40
30
20
10
Gallons per Minute
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
Head-Delivery Curves
of Pump
Operating at Various Speeds
and with
Various Proportions of Impeller
Ports Blocked off

Fig. 4



Brake Horse Power of Pump Operating at Various Speeds and with Various Proportions of the Impeller Ports Blocked off

Fig. 5

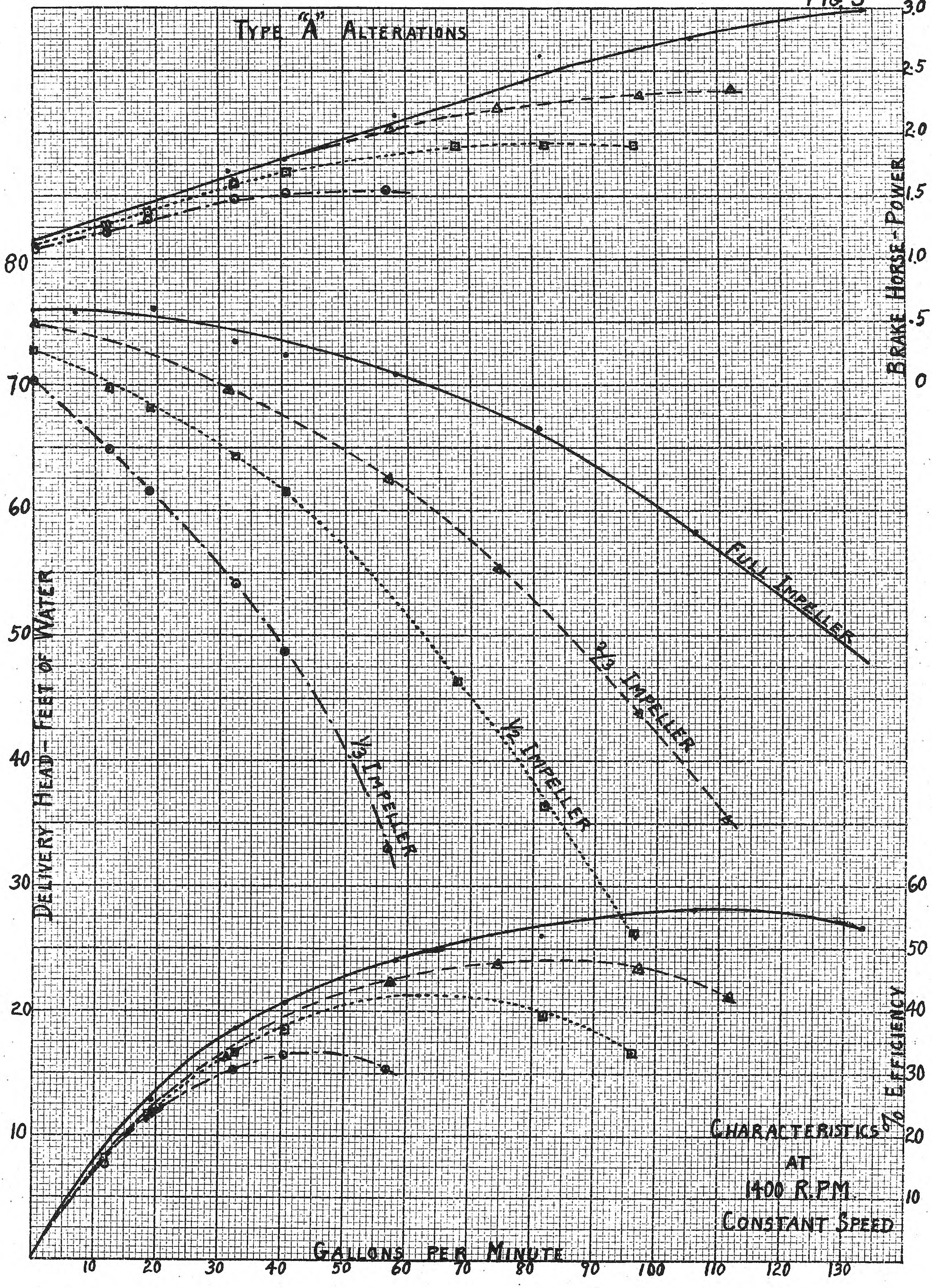

Type "A" Alterations
Full Impeller
2/3 Impeller
1/2 Impeller
1/3 Impeller
Delivery Head - Feet of Water
Brake Horse-Power
% Efficiency
Gallons per Minute
Characteristics at 1400 R.P.M. Constant Speed
10 20 30 40 50 60 70 80 90 100 110 120 130
10 20 30 40 50 60 70 80
0 .5 1.0 1.5 2.0 2.5 3.0
10 20 30 40 50 60

FIG. 6

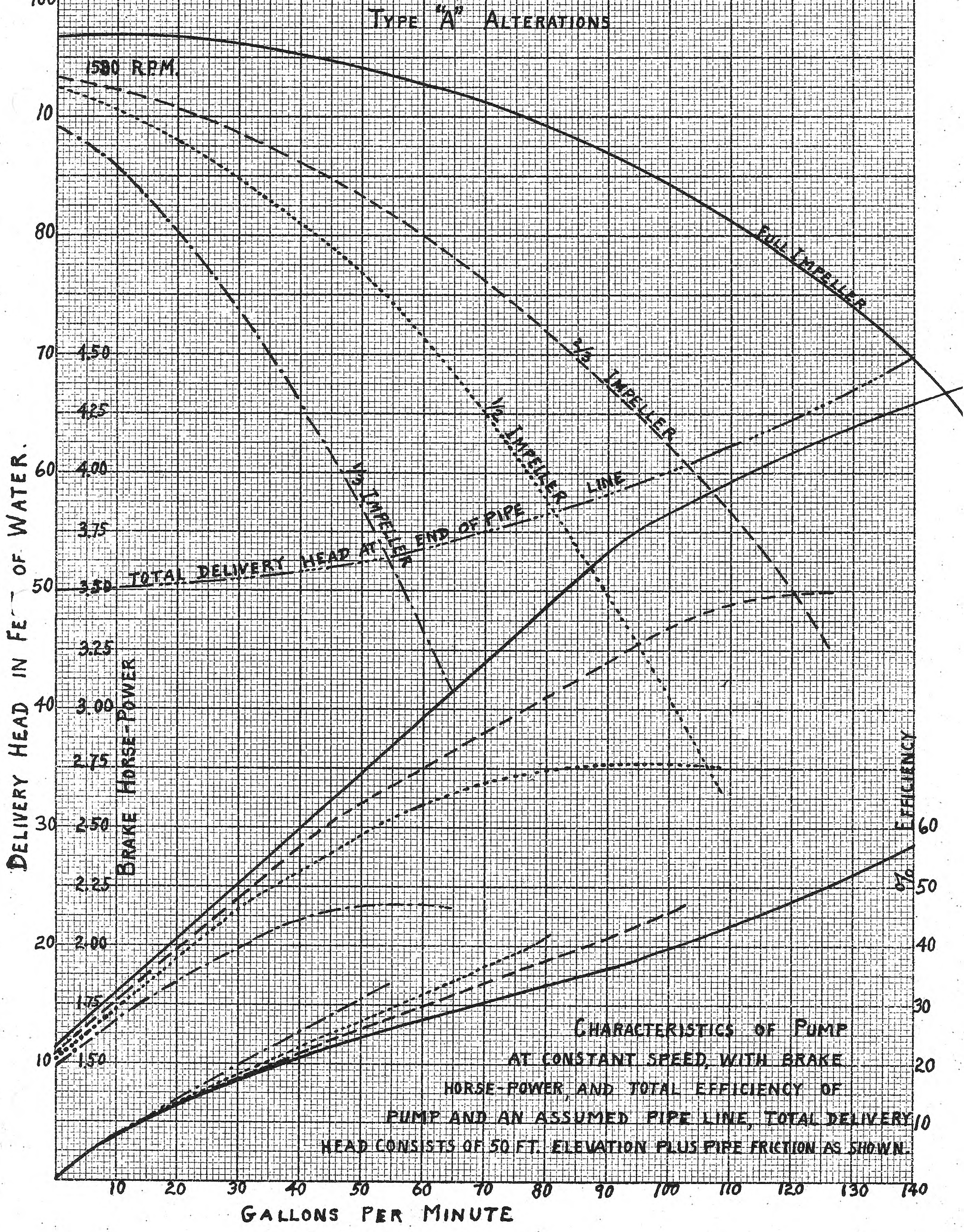

TYPE "A" ALTERATIONS
1500 R.P.M.
FULL IMPELLER
2/3 IMPELLER
1/2 IMPELLER
1/3 IMPELLER
TOTAL DELIVERY HEAD AT END OF PIPE LINE
DELIVERY HEAD IN FEET OF WATER.
BRAKE HORSE-POWER
% EFFICIENCY
GALLONS PER MINUTE
CHARACTERISTICS OF PUMP
AT CONSTANT SPEED, WITH BRAKE
HORSE-POWER, AND TOTAL EFFICIENCY OF
PUMP AND AN ASSUMED PIPE LINE, TOTAL DELIVERY
HEAD CONSISTS OF 50 FT. ELEVATION PLUS PIPE FRICTION AS SHOWN.

FIG. 7

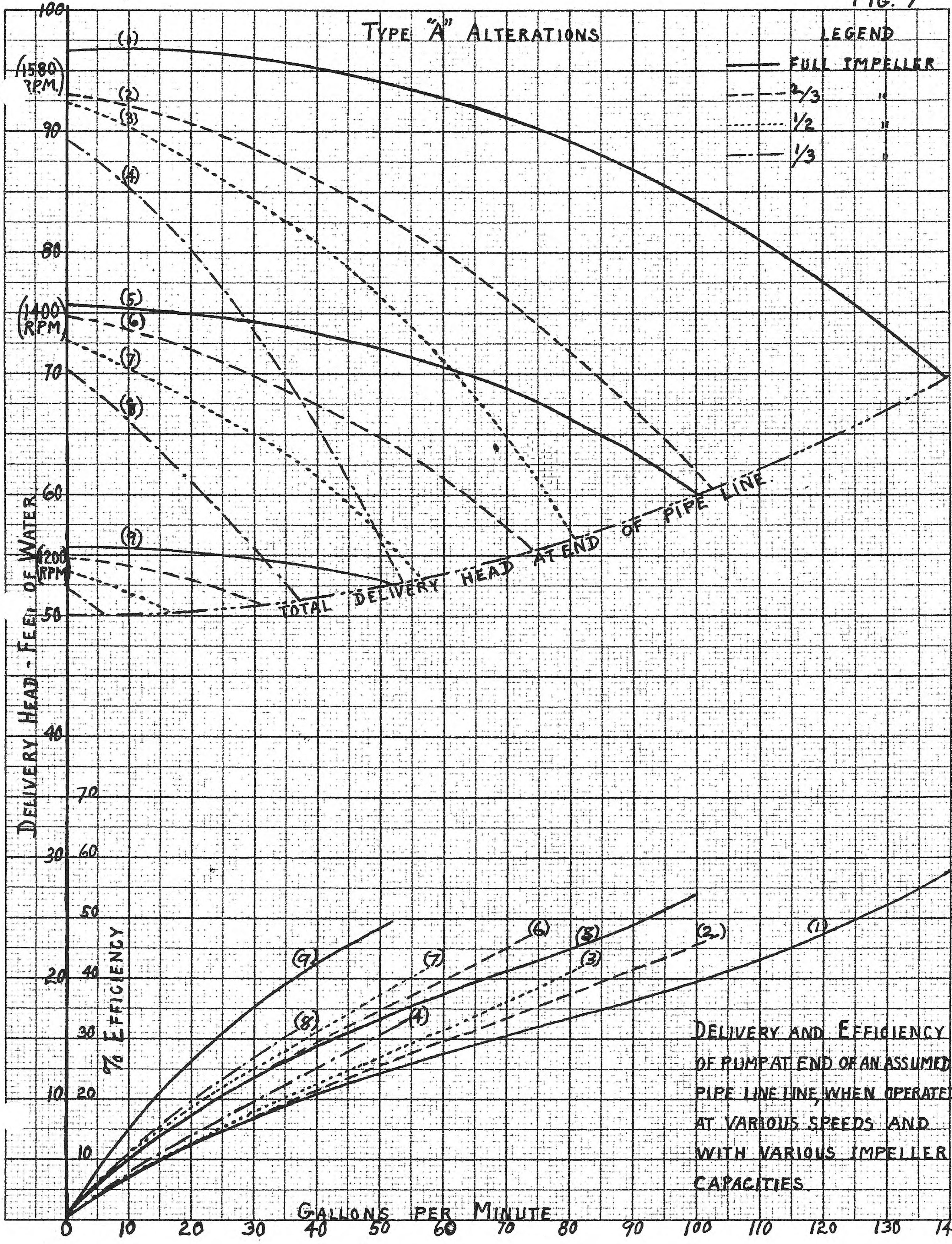

TYPE "A" ALTERATIONS
LEGEND
FULL IMPELLER
2/3 "
1/2 "
1/3 "
(1580 RPM)
(1400 RPM)
(1200 RPM)
(1)
(2)
(3)
(4)
(5)
(6)
(7)
(8)
(9)
TOTAL DELIVERY HEAD AT END OF PIPE LINE
DELIVERY HEAD - FEET OF WATER
% EFFICIENCY
GALLONS PER MINUTE
100
90
80
70
60
50
40
30
20
10
70
60
50
40
30
20
10
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
DELIVERY AND EFFICIENCY OF PUMP AT END OF AN ASSUMED PIPE LINE LINE, WHEN OPERATED AT VARIOUS SPEEDS AND WITH VARIOUS IMPELLER CAPACITIES

FIG. 8

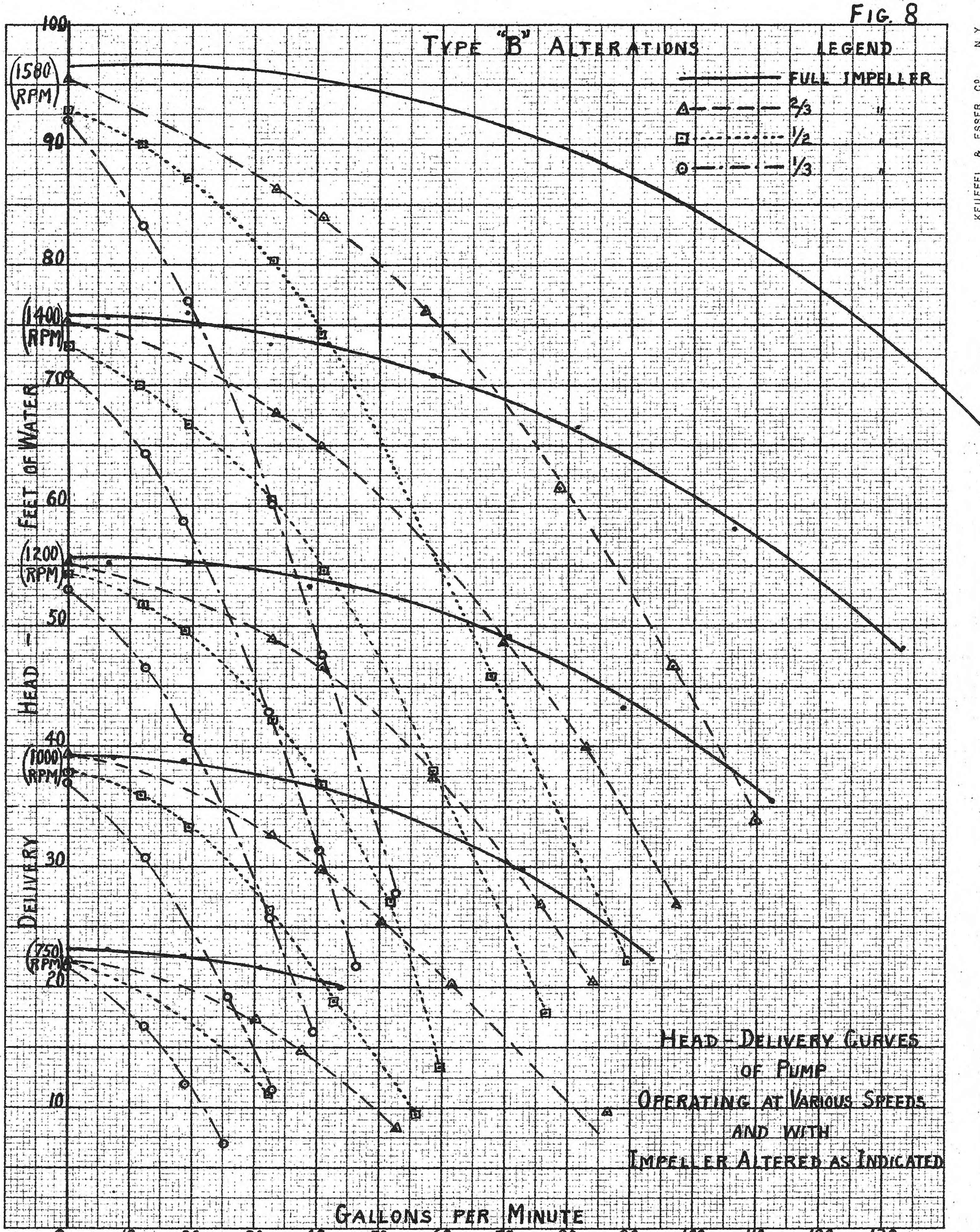
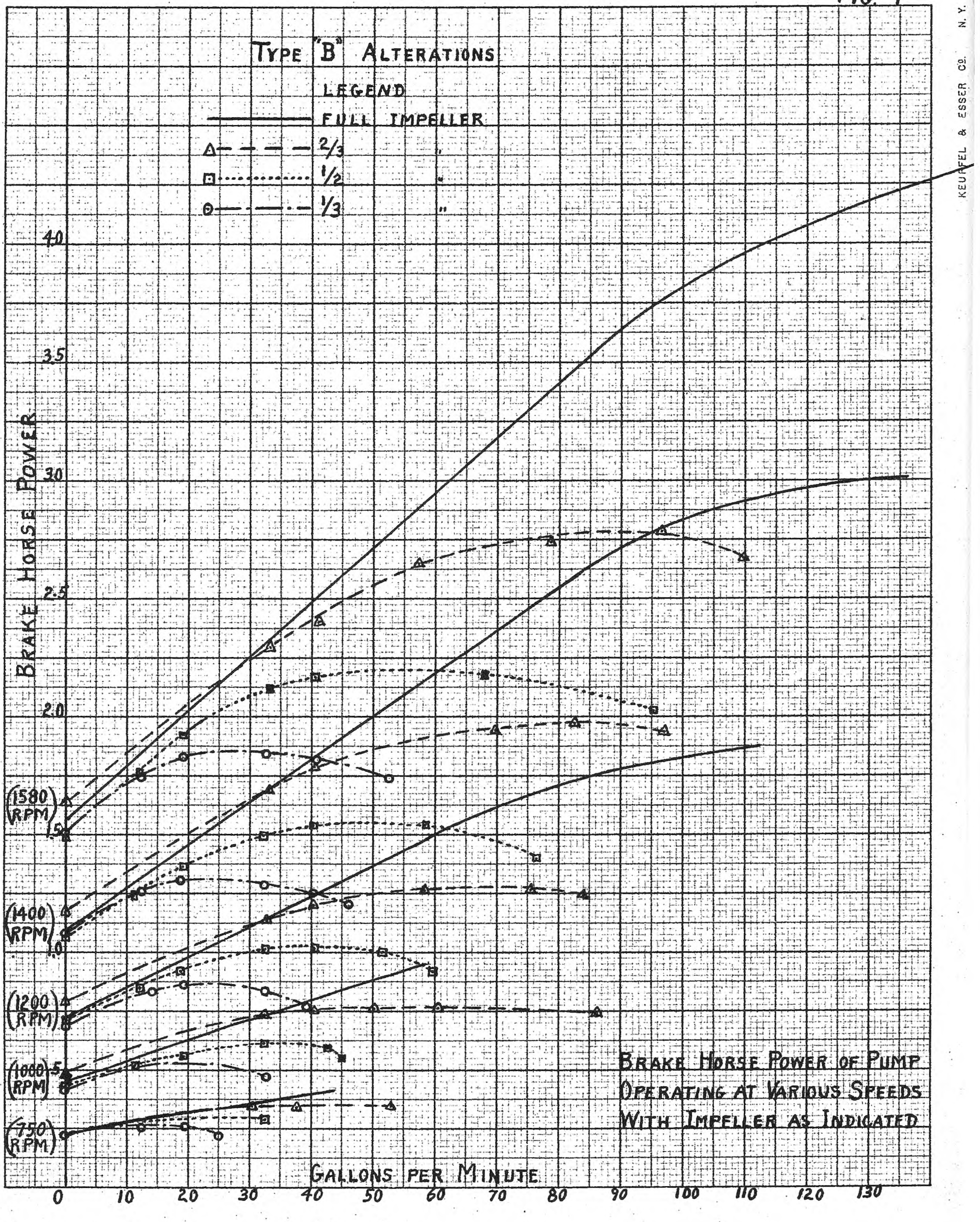

TYPE "B" ALTERATIONS
LEGEND
FULL IMPELLER
2/3 "
1/2 "
1/3 "
(1580 RPM)
(1400 RPM)
(1200 RPM)
(1000 RPM)
(750 RPM)
100
90
80
70
60
50
40
30
20
10
HEAD - FEET OF WATER
DELIVERY
0
10
20
30
40
50
60
70
80
90
100
110
120
130
GALLONS PER MINUTE
HEAD-DELIVERY CURVES
OF PUMP
OPERATING AT VARIOUS SPEEDS
AND WITH
IMPELLER ALTERED AS INDICATED

FIG. 9


TYPE "B" ALTERATIONS
LEGEND
FULL IMPELLER
2/3 "
1/2 "
1/3 "
BRAKE HORSE POWER
4.0
3.5
3.0
2.5
2.0
1.5
1.0
.5
(1580 RPM)
(1400 RPM)
(1200 RPM)
(1000 RPM)
(750 RPM)
BRAKE HORSE POWER OF PUMP
OPERATING AT VARIOUS SPEEDS
WITH IMPELLER AS INDICATED
GALLONS PER MINUTE
0
10
20
30
40
50
60
70
80
90
100
110
120
130

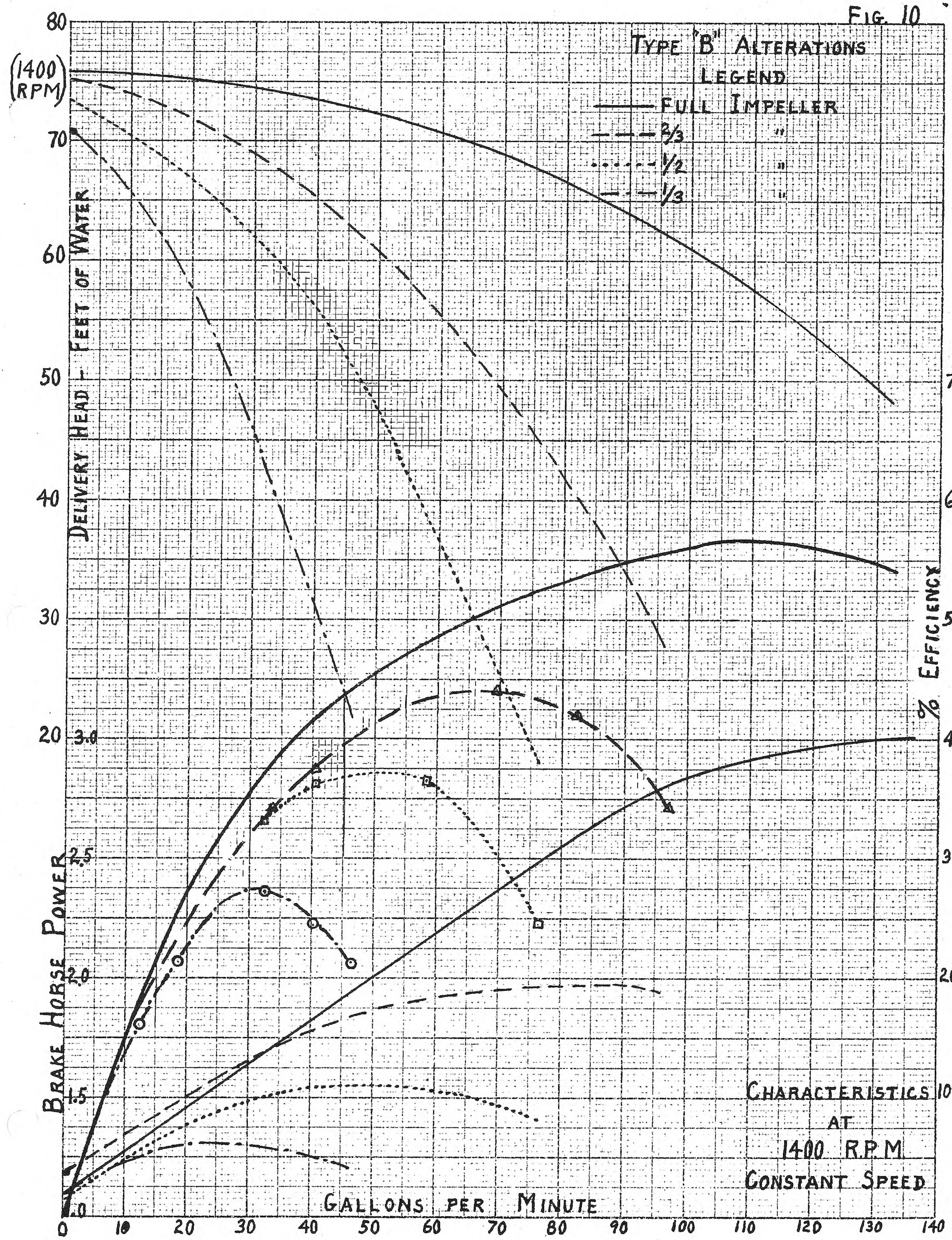


Fig. 10
Type "B" Alterations
Legend
Full Impeller
2/3 "
1/2 "
1/3 "
(1400 RPM)
Delivery Head - Feet of Water
Brake Horse Power
% Efficiency
Gallons per Minute
Characteristics
at
1400 R.P.M.
Constant Speed
80
70
60
50
40
30
20
3.0
2.5
2.0
1.5
1.0
70
60
50
40
30
20
10
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140

FIG. 11

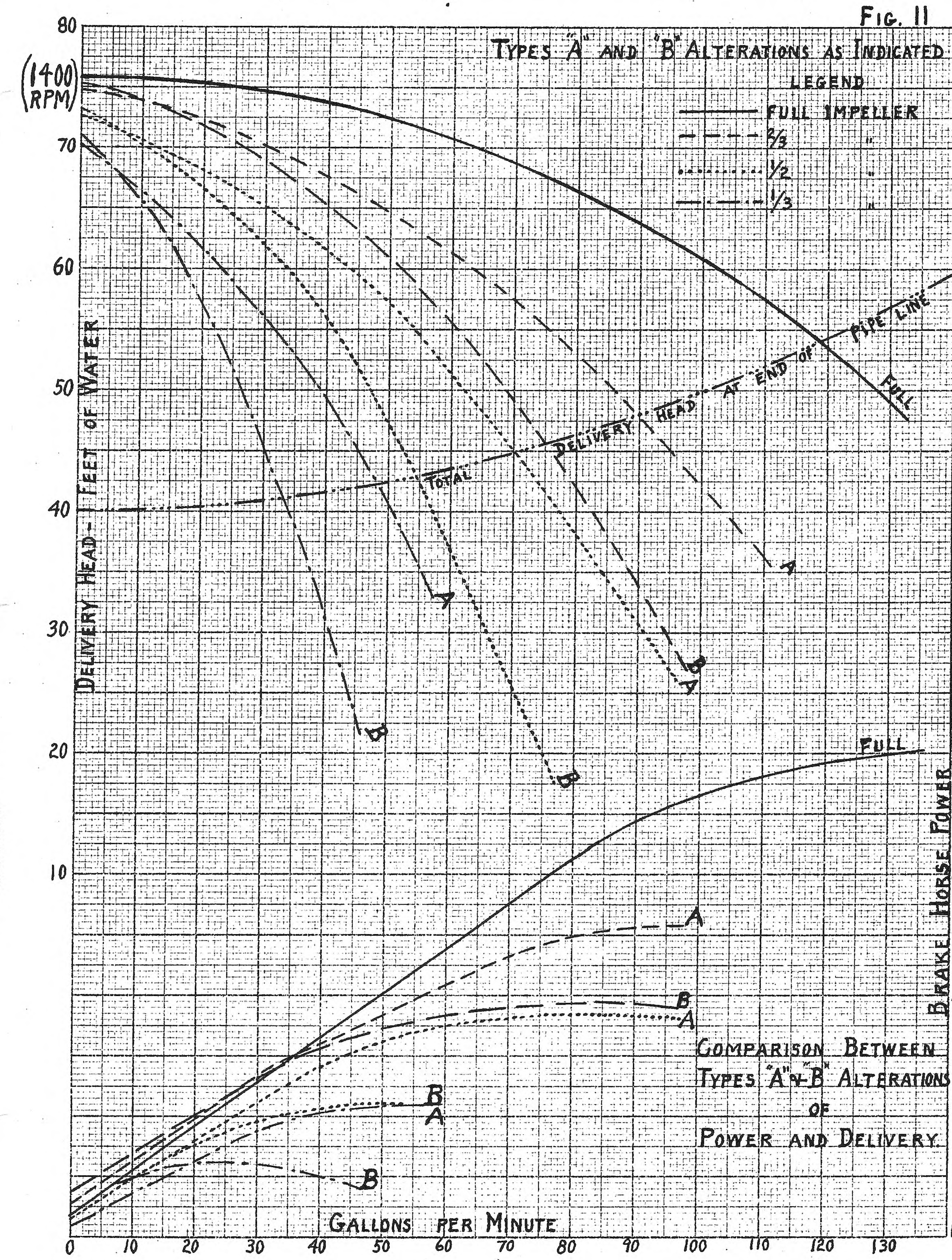

TYPES "A" AND "B" ALTERATIONS AS INDICATED
LEGEND
FULL IMPELLER
2/3 "
1/2 "
1/3 "
(1400 RPM)
DELIVERY HEAD - FEET OF WATER
TOTAL DELIVERY HEAD AT END OF PIPE LINE
FULL
A
B
BRAKE HORSE POWER
COMPARISON BETWEEN TYPES "A" & "B" ALTERATIONS OF POWER AND DELIVERY
GALLONS PER MINUTE
80
70
60
50
40
30
20
10
3.0
2.5
2.0
1.5
1.0
0
10
20
30
40
50
60
70
80
90
100
110
120
130
KEUFFEL & ESSER CO., N. Y. NO. 359-11
20 × 20 to the inch.

FIG. 12

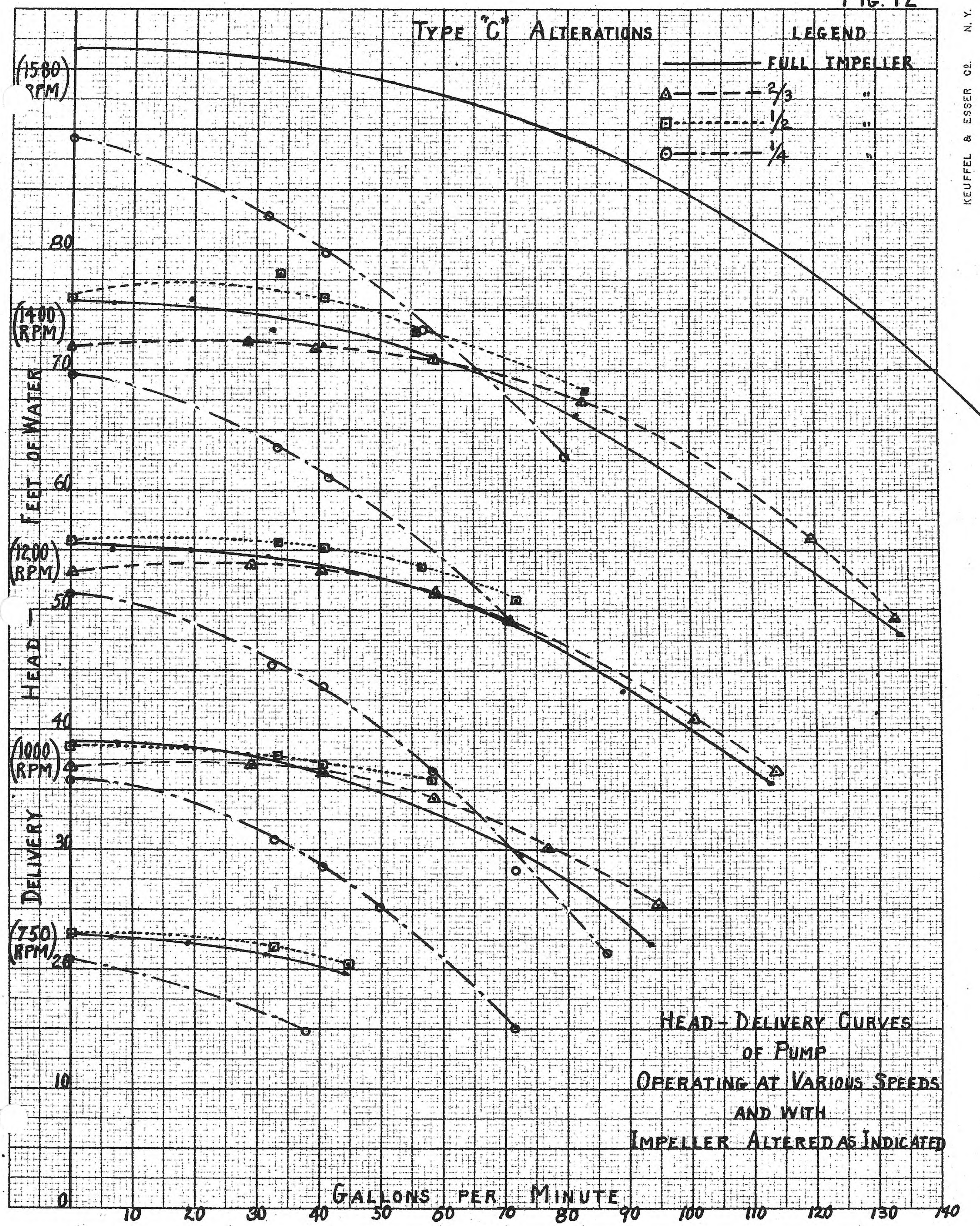

TYPE "C" ALTERATIONS
LEGEND
FULL IMPELLER
2/3 "
1/2 "
1/4 "
(1580 RPM)
(1400 RPM)
(1200 RPM)
(1000 RPM)
(750 RPM)
DELIVERY HEAD FEET OF WATER
0
10
20
30
40
50
60
70
80
GALLONS PER MINUTE
10
20
30
40
50
60
70
80
90
100
110
120
130
140
HEAD-DELIVERY CURVES
OF PUMP
OPERATING AT VARIOUS SPEEDS
AND WITH
IMPELLER ALTERED AS INDICATED

FIG 13

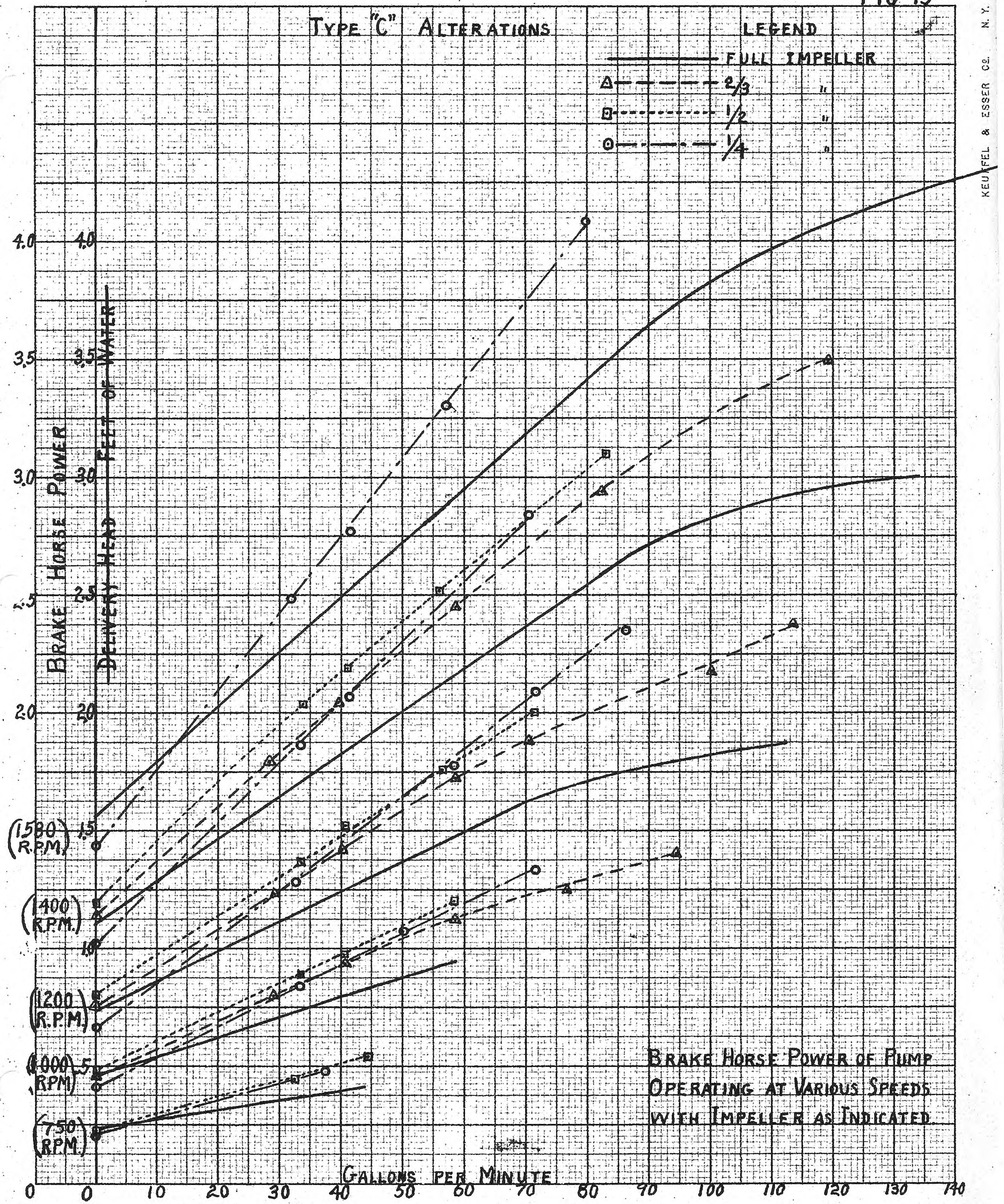

TYPE "C" ALTERATIONS
LEGEND
FULL IMPELLER
2/3 "
1/2 "
1/4 "
BRAKE HORSE POWER
DELIVERY HEAD — FEET OF WATER
(1580 R.P.M.)
(1400 R.P.M.)
(1200 R.P.M.)
(1000 RPM)
(750 RPM)
GALLONS PER MINUTE
0 0 10 20 30 40 50 60 70 80 90 100 110 120 130 140
.5 1.0 1.5 2.0 2.5 3.0 3.5 4.0
BRAKE HORSE POWER OF PUMP OPERATING AT VARIOUS SPEEDS WITH IMPELLER AS INDICATED.
KEUFFEL & ESSER CO. N.Y.

FIG. 14

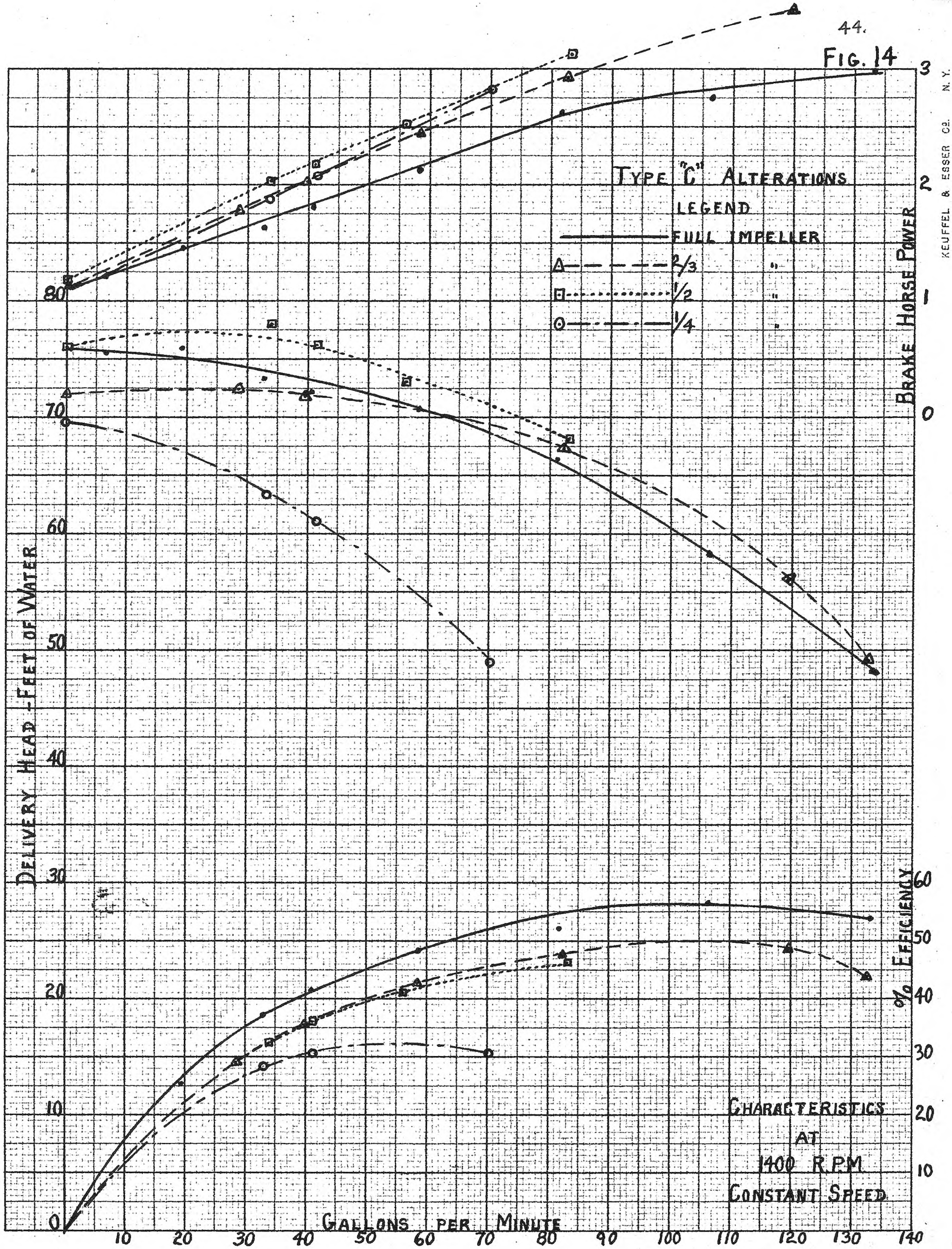

TYPE "C" ALTERATIONS
LEGEND
FULL IMPELLER
2/3 "
1/2 "
1/4 "
BRAKE HORSE POWER
DELIVERY HEAD - FEET OF WATER
% EFFICIENCY
GALLONS PER MINUTE
CHARACTERISTICS
AT
1400 R.P.M.
CONSTANT SPEED
KEUFFEL & ESSER CO. N.Y.

FIG 15

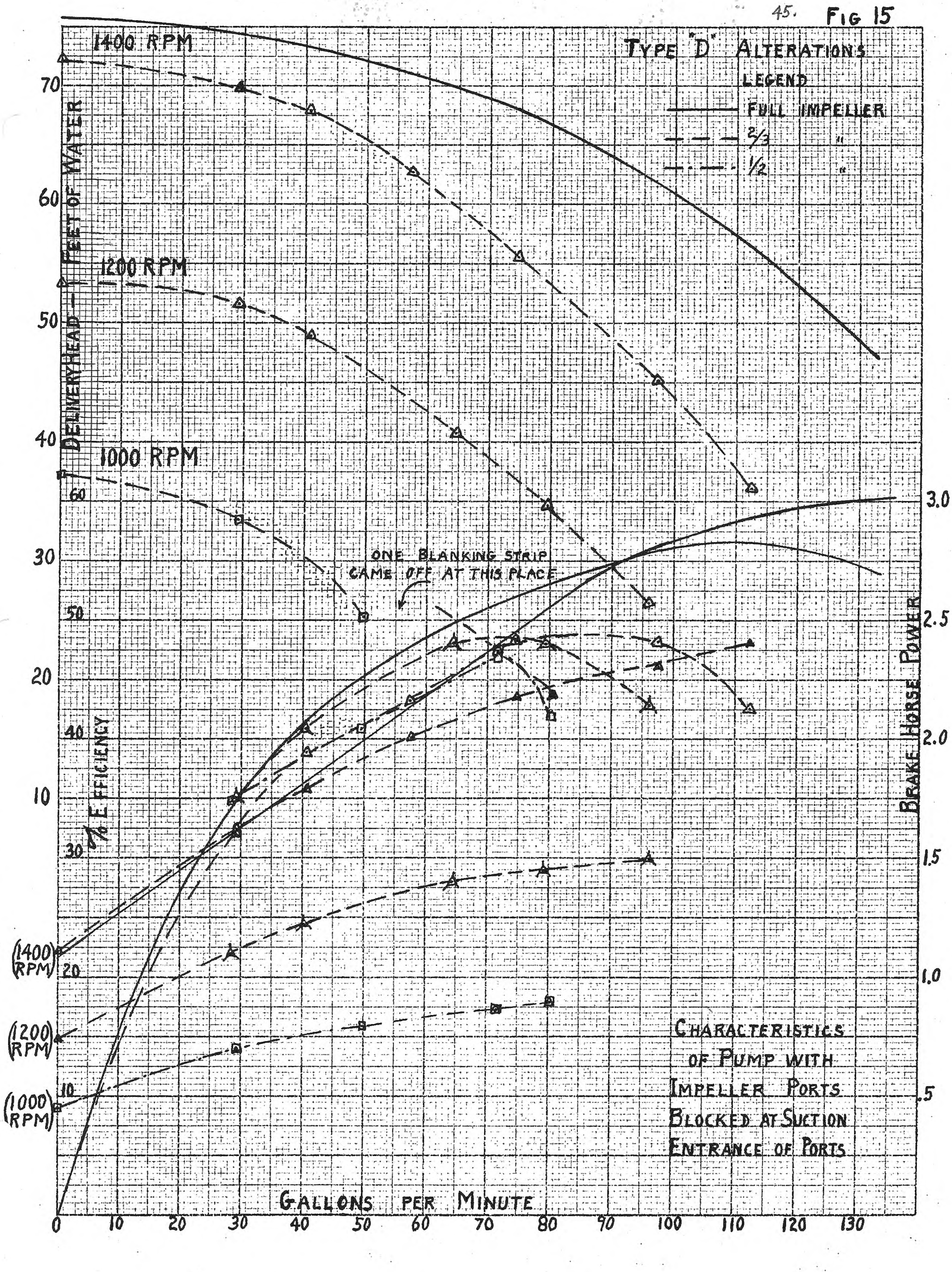

TYPE "D" ALTERATIONS
LEGEND
FULL IMPELLER
2/3 "
1/2 "
1400 RPM
1200 RPM
1000 RPM
DELIVERY HEAD — FEET OF WATER
% EFFICIENCY
BRAKE HORSE POWER
ONE BLANKING STRIP CAME OFF AT THIS PLACE
(1400 RPM)
(1200 RPM)
(1000 RPM)
GALLONS PER MINUTE
0 10 20 30 40 50 60 70 80 90 100 110 120 130
10 20 30 40 50 60 70
10 20 30 40 50 60
.5 1.0 1.5 2.0 2.5 3.0
CHARACTERISTICS OF PUMP WITH IMPELLER PORTS BLOCKED AT SUCTION ENTRANCE OF PORTS

FIG. 16

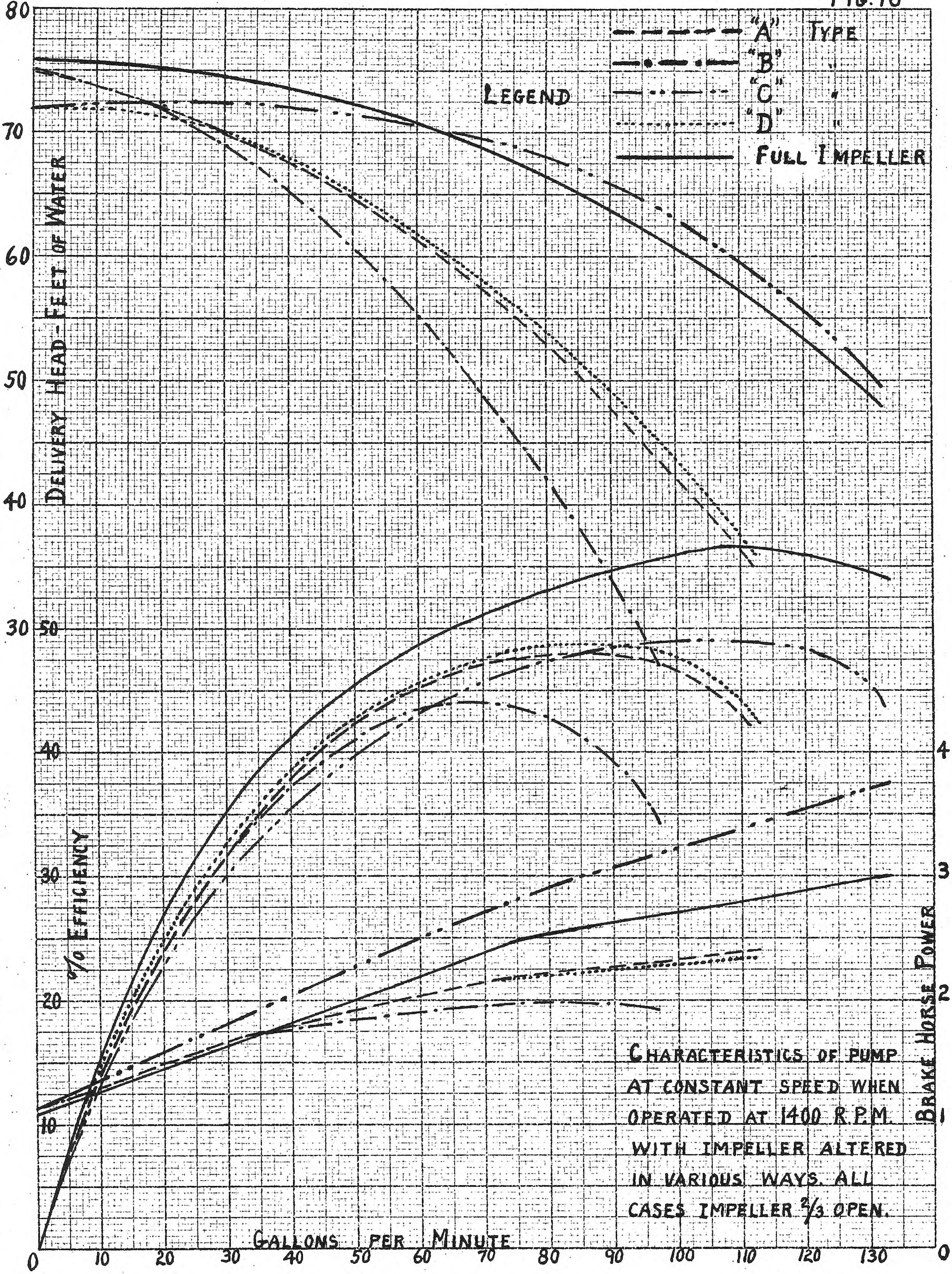
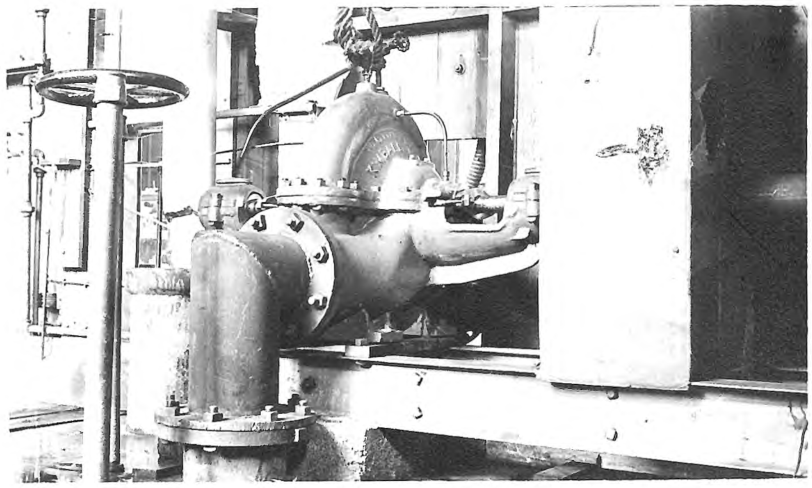
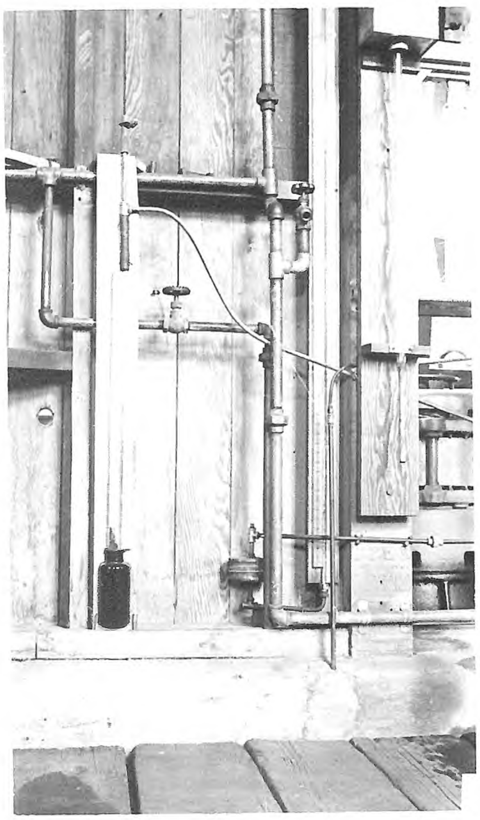
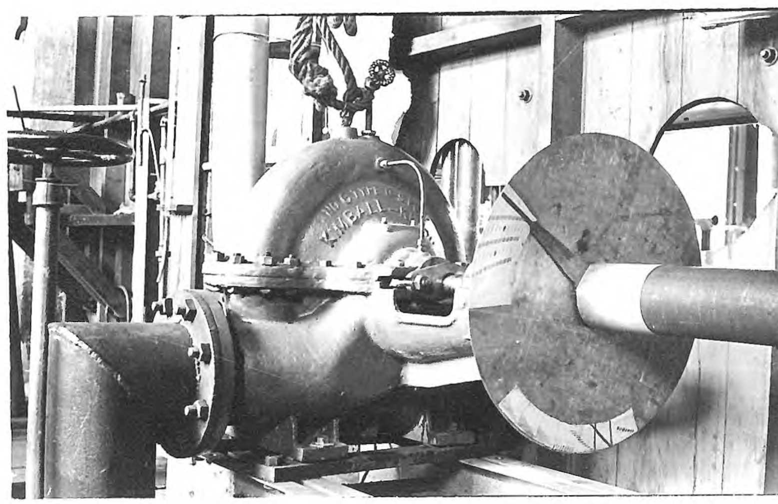

LEGEND
"A" TYPE
"B" "
"C" "
"D" "
FULL IMPELLER
DELIVERY HEAD - FEET OF WATER
% EFFICIENCY
BRAKE HORSE POWER
GALLONS PER MINUTE
CHARACTERISTICS OF PUMP
AT CONSTANT SPEED WHEN
OPERATED AT 1400 R.P.M.
WITH IMPELLER ALTERED
IN VARIOUS WAYS. ALL
CASES IMPELLER 2/3 OPEN.
80
70
60
50
40
30
20
10
0
10
20
30
40
50
60
70
80
90
100
110
120
130
1
2
3
4

FIG. 17







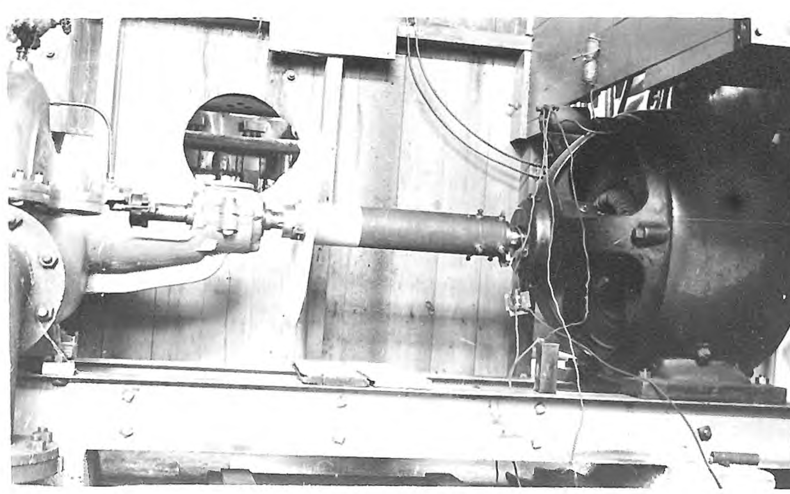


FIG. 18

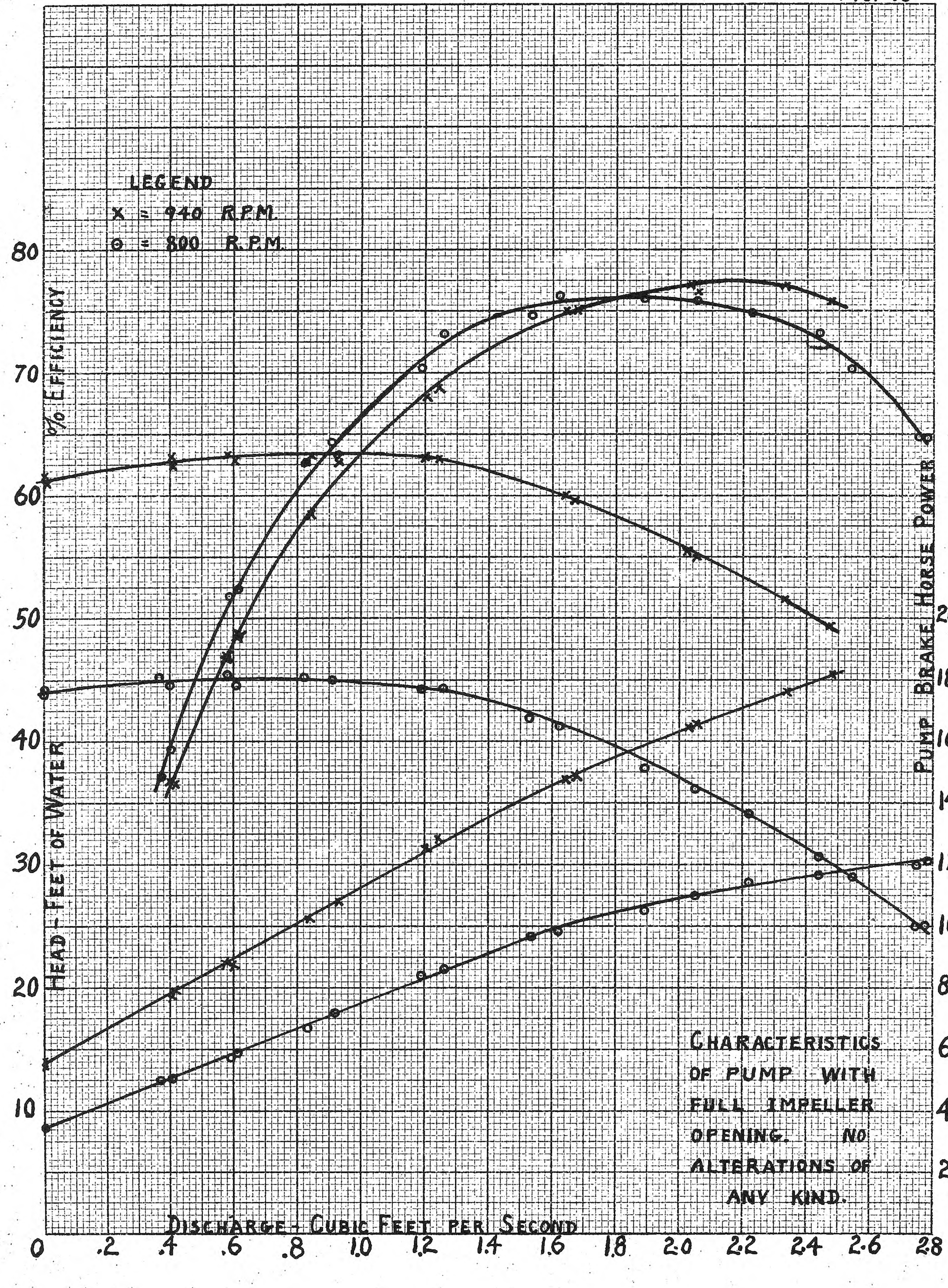

LEGEND
X = 940 R.P.M.
⊙ = 800 R.P.M.
% EFFICIENCY
HEAD - FEET OF WATER
PUMP BRAKE HORSE POWER
DISCHARGE - CUBIC FEET PER SECOND
CHARACTERISTICS OF PUMP WITH FULL IMPELLER OPENING. NO ALTERATIONS OF ANY KIND.
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
2.8
10
20
30
40
50
60
70
80
2
4
6
8
10
12
14
16
18
20

FIG. 19

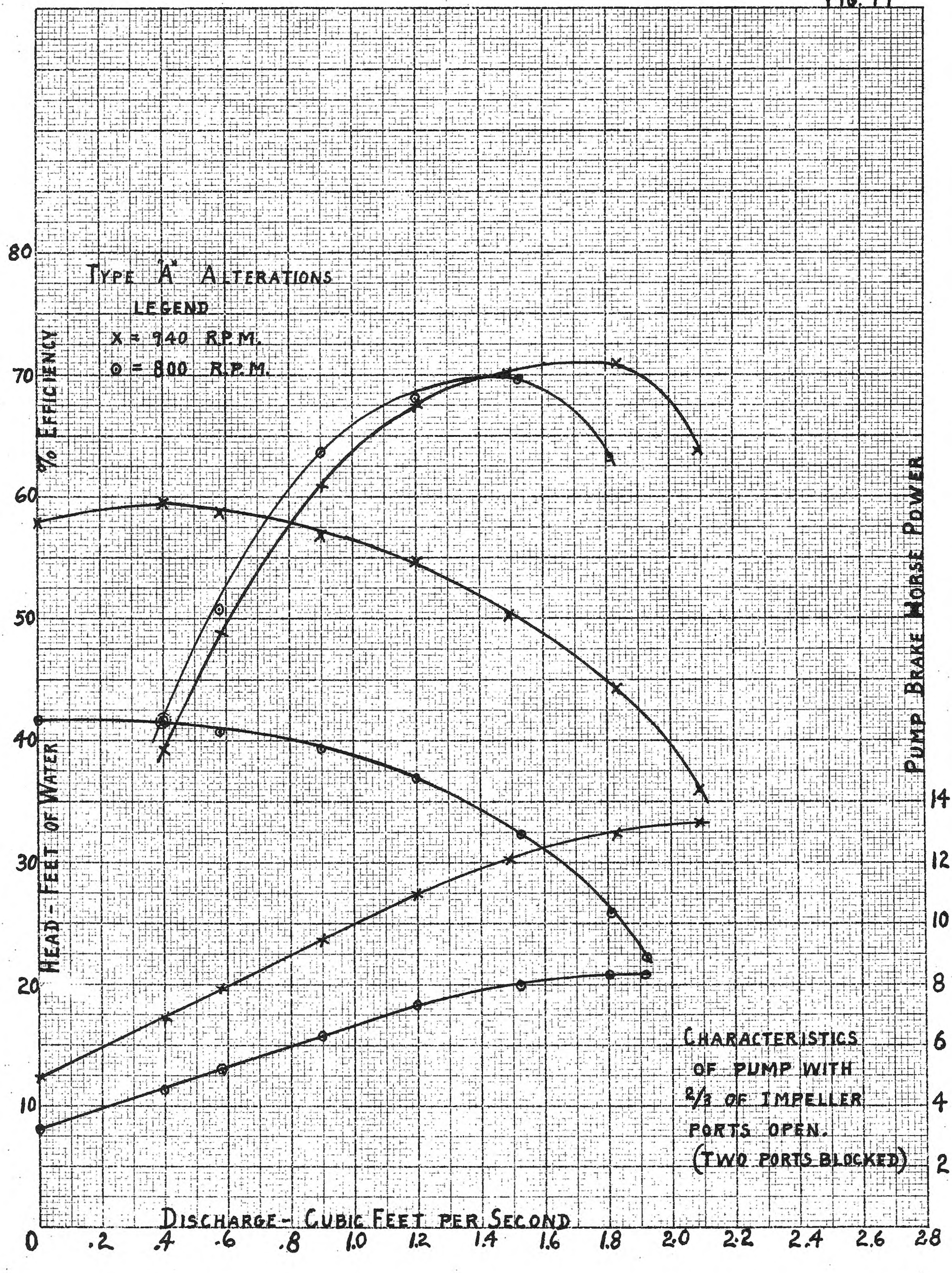
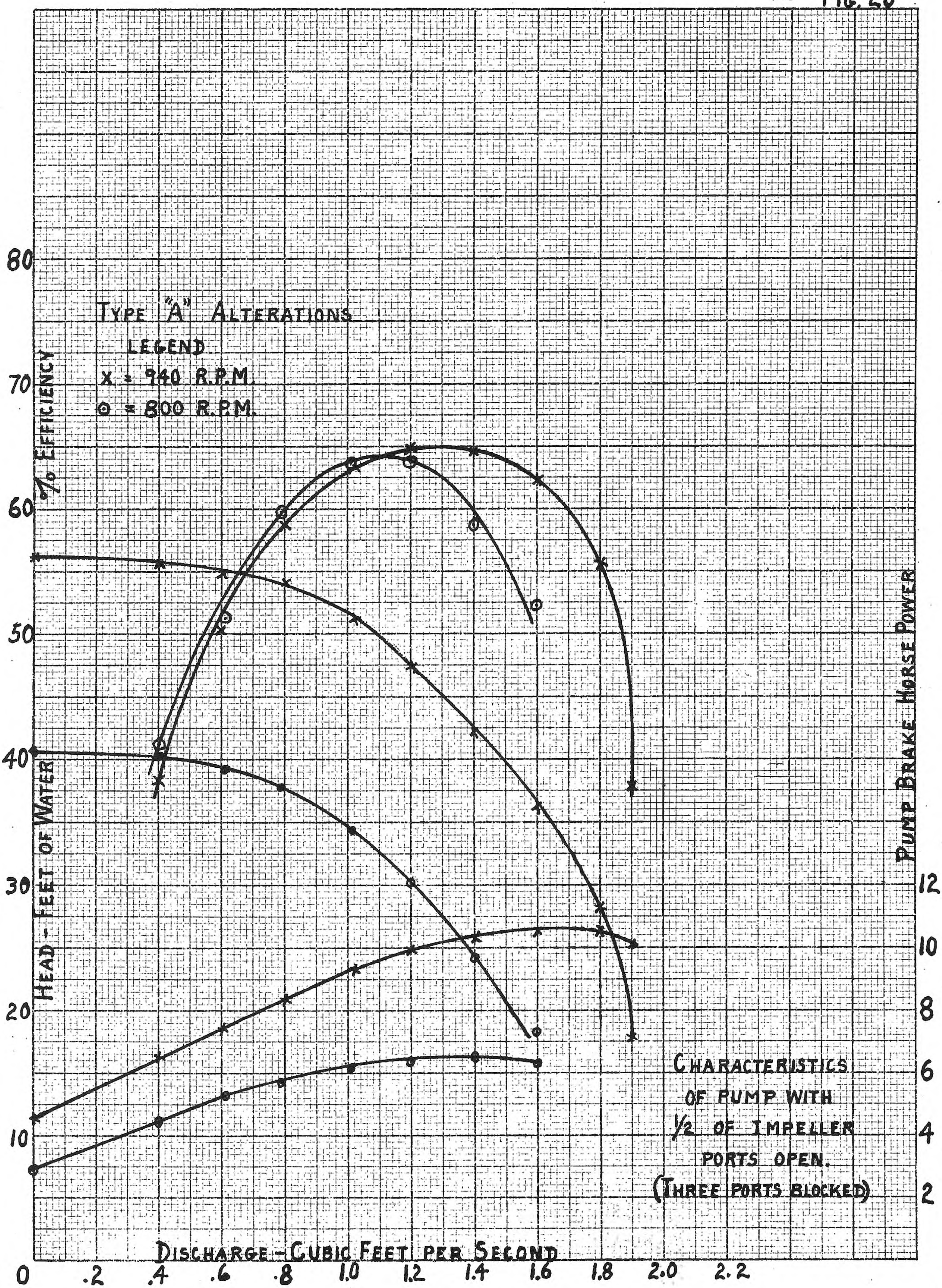

TYPE "A" ALTERATIONS
LEGEND
X = 940 R.P.M.
⊙ = 800 R.P.M.
% EFFICIENCY
HEAD - FEET OF WATER
PUMP BRAKE HORSE POWER
DISCHARGE - CUBIC FEET PER SECOND
CHARACTERISTICS OF PUMP WITH 2/3 OF IMPELLER PORTS OPEN. (TWO PORTS BLOCKED)
80
70
60
50
40
30
20
10
14
12
10
8
6
4
2
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
2.8

FIG. 20



20 × 20 to the inch.

FIG. 21

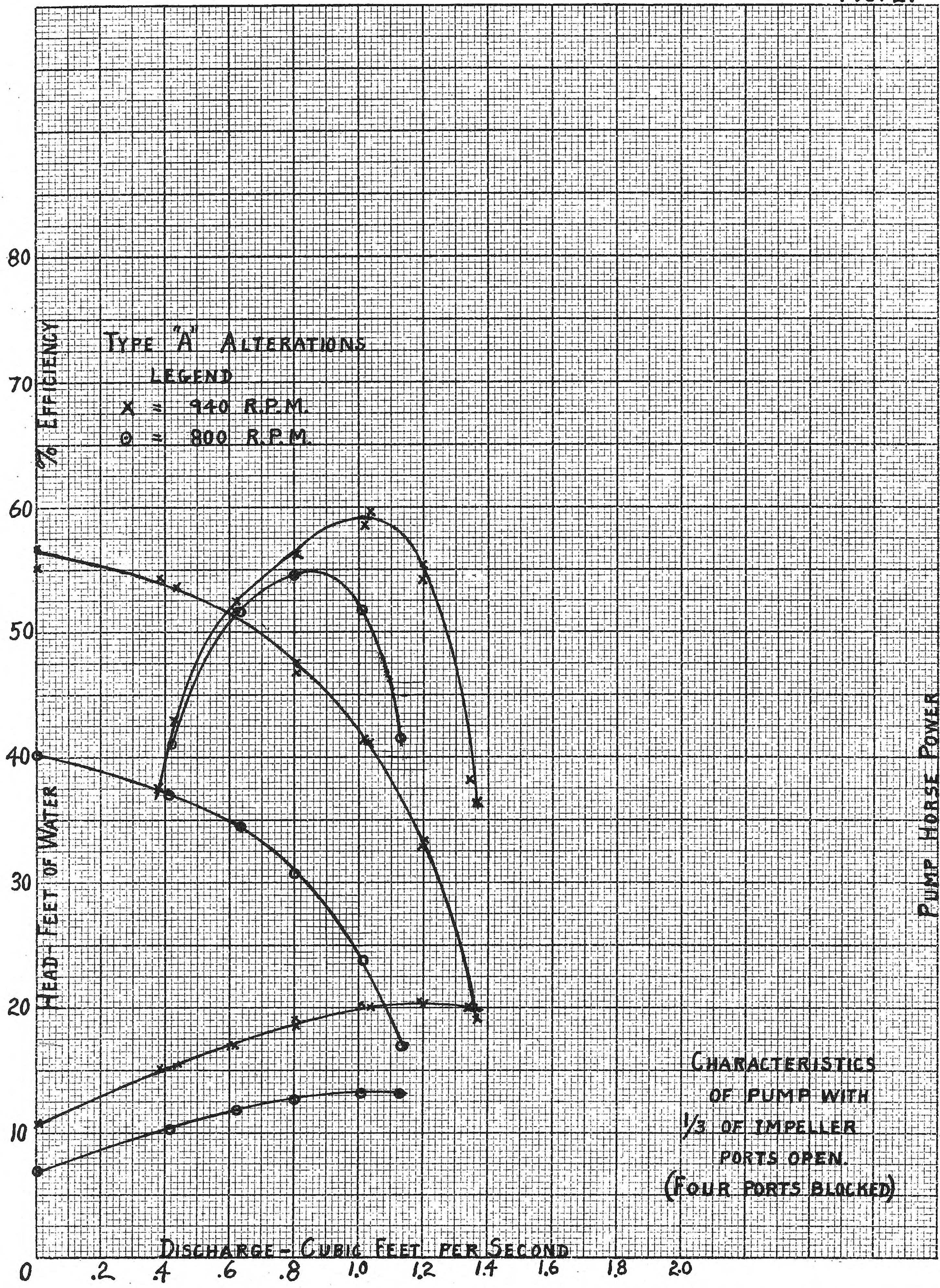

TYPE "A" ALTERATIONS
LEGEND
X = 940 R.P.M.
O = 800 R.P.M.
% EFFICIENCY
HEAD-FEET OF WATER
PUMP HORSE POWER
CHARACTERISTICS OF PUMP WITH 1/3 OF IMPELLER PORTS OPEN. (FOUR PORTS BLOCKED)
DISCHARGE - CUBIC FEET PER SECOND
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
1.8
2.0
10
20
30
40
50
60
70
80
2
4
6
8
10

FIG. 22

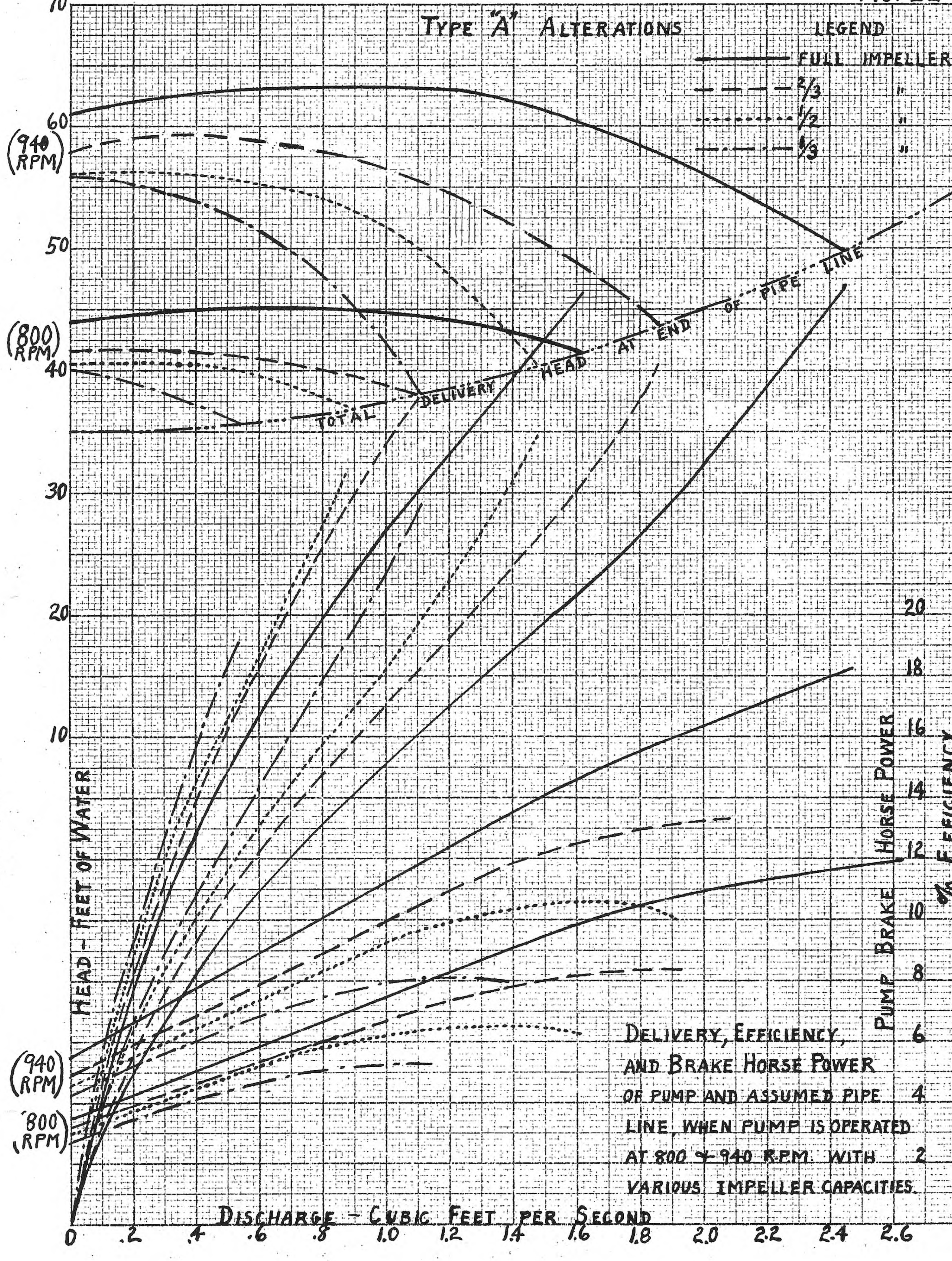

TYPE "A" ALTERATIONS
LEGEND
FULL IMPELLER
2/3 "
1/2 "
1/3 "
TOTAL DELIVERY HEAD AT END OF PIPE LINE
(940 RPM)
(800 RPM)
HEAD - FEET OF WATER
PUMP BRAKE HORSE POWER
% EFFICIENCY
DISCHARGE - CUBIC FEET PER SECOND
0 .2 .4 .6 .8 1.0 1.2 1.4 1.6 1.8 2.0 2.2 2.4 2.6
10 20 30 40 50 60 70
2 4 6 8 10 12 14 16 18 20
10 20 30 40 50 60 70 80
DELIVERY, EFFICIENCY, AND BRAKE HORSE POWER OF PUMP AND ASSUMED PIPE LINE, WHEN PUMP IS OPERATED AT 800 + 940 R.P.M. WITH VARIOUS IMPELLER CAPACITIES.

FIG. 23

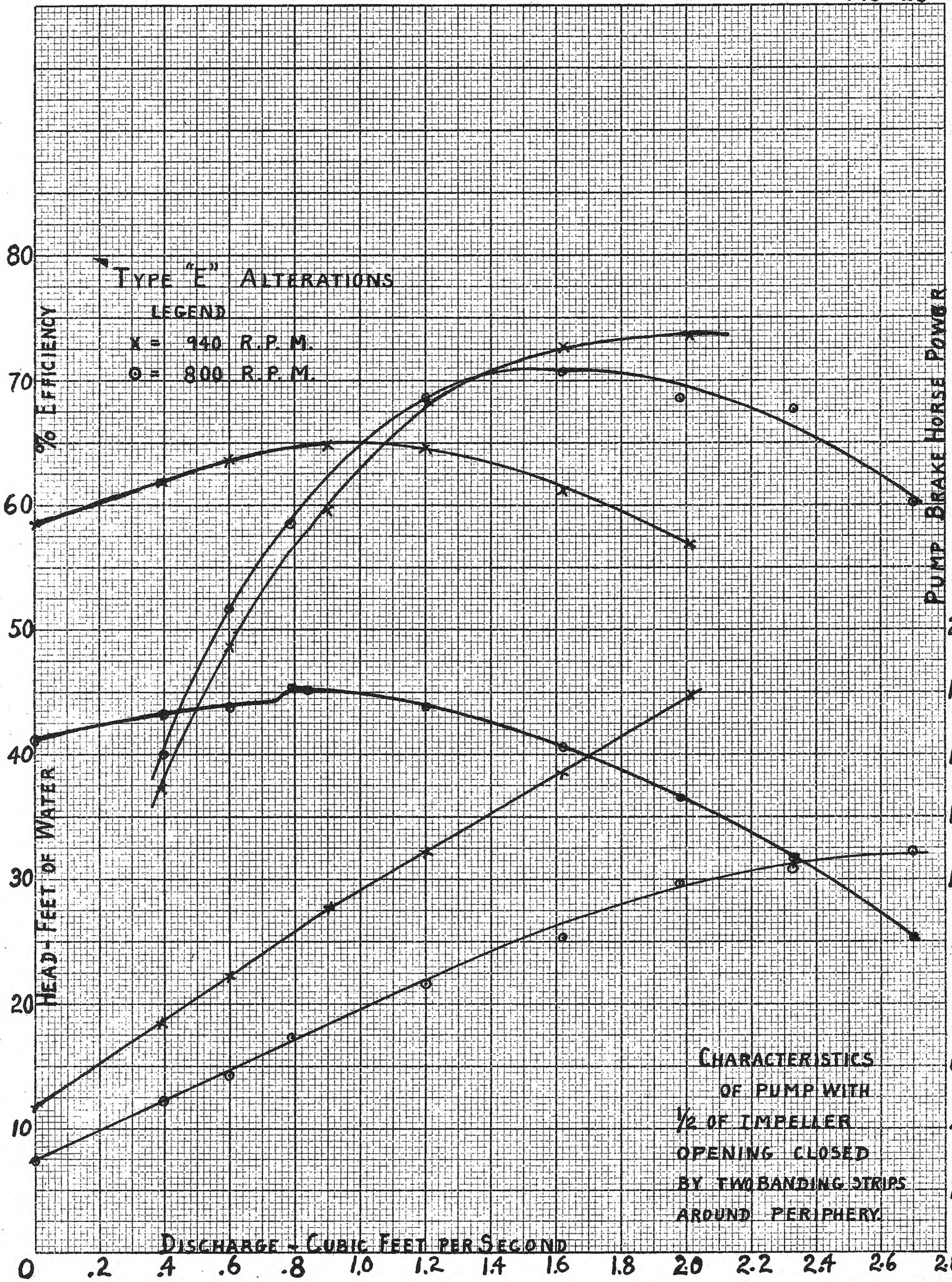

TYPE "E" ALTERATIONS
LEGEND
X = 940 R.P.M.
⊙ = 800 R.P.M.
% EFFICIENCY
HEAD - FEET OF WATER
PUMP BRAKE HORSE POWER
DISCHARGE - CUBIC FEET PER SECOND
CHARACTERISTICS OF PUMP WITH ½ OF IMPELLER OPENING CLOSED BY TWO BANDING STRIPS AROUND PERIPHERY.
80
70
60
50
40
30
20
10
20
18
16
14
12
10
8
6
4
2
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
2.8

FIG. 24

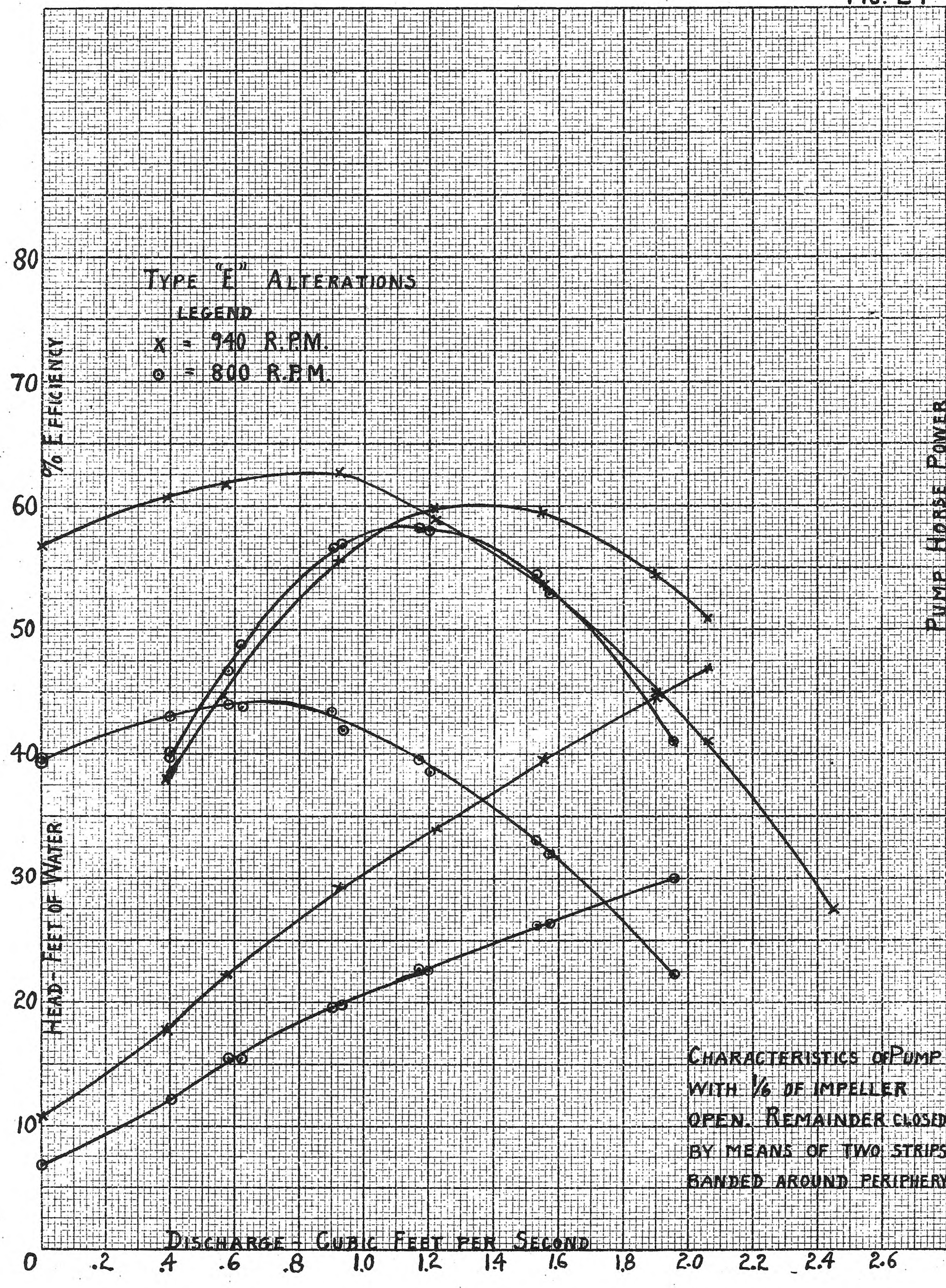
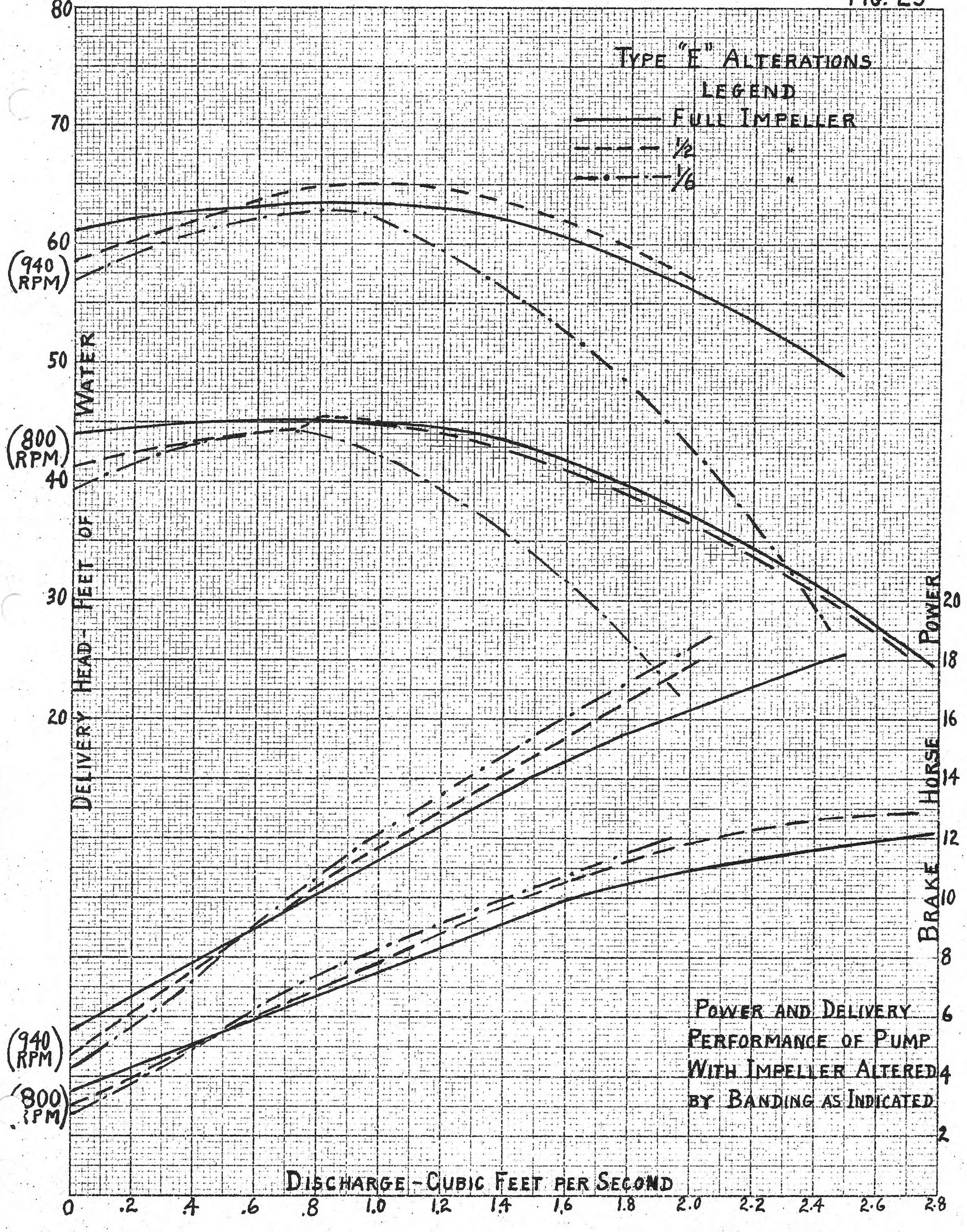
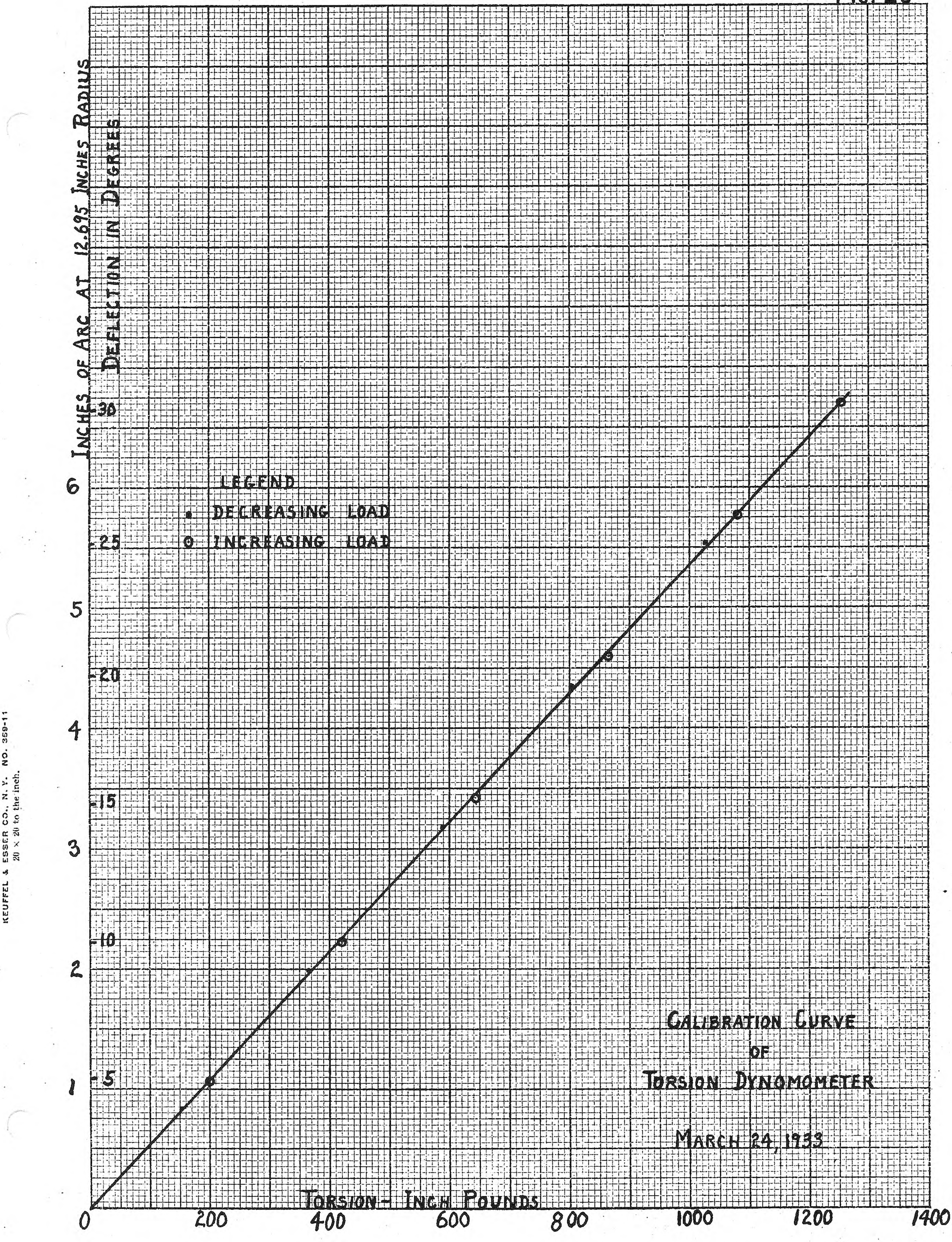

TYPE "E" ALTERATIONS
LEGEND
X = 940 R.P.M.
⊙ = 800 R.P.M.
% EFFICIENCY
HEAD - FEET OF WATER
PUMP HORSE POWER
DISCHARGE - CUBIC FEET PER SECOND
CHARACTERISTICS OF PUMP WITH 1/6 OF IMPELLER OPEN. REMAINDER CLOSED BY MEANS OF TWO STRIPS BANDED AROUND PERIPHERY
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
10
20
30
40
50
60
70
80
2
4
6
8
10
12
14
16
18

FIG. 25



20 × 20 to the inch.

FIG. 26


INCHES OF ARC AT 12.695 INCHES RADIUS
DEFLECTION IN DEGREES
LEGEND
• DECREASING LOAD
O INCREASING LOAD
CALIBRATION CURVE
OF
TORSION DYNOMOMETER
MARCH 24, 1933
TORSION - INCH POUNDS
0
200
400
600
800
1000
1200
1400
1
2
3
4
5
6
5
10
15
20
25
30
KEUFFEL & ESSER CO., N. Y. NO. 359-11
20 × 20 to the inch.

Impeller	GPM (1580 R.P.M.)	BHP	WHP	TOTAL EFF. P&P.L.	GPM (1400 R.P.M.)	BHP	WHP	TOTAL EFF. P&P.L.	GPM (1200 R.P.M.)	BHP	WHP	TOTAL EFF. P&P.L.
FULL IMPELLER	20	2.05	.254	.124	20	1.45	.254	.175	10	.85	.126	.148
	40	2.48	.522	.210	40	1.82	.522	.287	20	.97	.254	.262
	60	2.95	.813	.276	60	2.17	.813	.375	30	1.10	.386	.351
	80	3.42	1.139	.333	80	2.55	1.139	.446	40	1.23	.522	.424
	100	3.82	1.516	.397	100	2.82	1.516	.538	50	1.36	.663	.488
	120	4.12	1.950	.473								
	140	4.28	2.460	.575								
2/3 IMPELLER	20	1.97	.254	.129	20	1.48	.254	.172	10	.85	.126	.148
	40	2.40	.522	.218	40	1.78	.522	.293	20	.98	.254	.259
	60	2.73	.813	.298	50	1.92	.663	.346	30	1.10	.386	.351
	80	3.03	1.139	.376	60	2.04	.813	.399				
	100	3.33	1.516	.455	70	2.15	.970	.451				
1/2 IMPELLER	20	1.93	.254	.131	10	1.23	.126	.102				
	40	2.30	.522	.227	20	1.40	.254	.182	10	.82	.126	.154
	50	2.46	.663	.270	30	1.55	.386	.249				
	60	2.58	.813	.315	40	1.70	.522	.307				
	70	2.68	.970	.362	50	1.79	.663	.371				
	80	2.73	1.139	.417								
1/3 IMPELLER	10	1.67	.126	.075	10	1.17	.126	.108				
	20	1.83	.254	.139	20	1.32	.254	.193				
	30	1.97	.386	.196	30	1.44	.386	.268				
	40	2.10	.522	.249								
	50	2.15	.663	.308								

TYPE "A" ALTERATION

Data used for curves on Fig. 6 and Fig. 7

Brake horse power and water horse power are calculated from values read from curves. The brake horse power is taken from the curves on Fig. 4. The water horse power is based on the delivery head of the pipe line curve shown on Fig. 6 and Fig. 7. It was assumed the elevation difference that the water was delivered at was 50 feet, and that the friction head at the flow of 140 G.P.M. was 19.6 feet of water, and that the friction varied directly as the square of the water flow. This would be a true assumption for a long pipe line with uniform cross section and smoothness.

Net Wt. of Water (lbs.)	Time in Sec.	Flow lbs. sec.	Flow gal. min.	Dynamic Head In. of water (H)	$\sqrt{H}$	Coeff. of Dischg.	Remarks for Orifice Calibration
305	32.6	9.36	67.5	469	21.66	3.12	Experimental value
292	31.1	9.39	67.5	469	21.66	3.12	used in tests with
287	30.8	9.32	67.1	466	21.59	3.11	small orifice.
$288\frac{1}{2}$	31.0	9.32	67.1	467	21.61	3.10	Calculated value
300	36.1	8.31	59.9	378	19.44	3.08	used for large
$290\frac{1}{2}$	38.4	7.57	54.5	304	17.45	3.12	orifice.
298	39.4	7.57	54.5	308	17.55	3.11	
323	48.6	6.64	47.8	239.5	15.48	3.09	
$333\frac{1}{2}$	57.6	5.78	41.7	182.5	13.51	3.08	
330	71.0	4.65	33.5	117.3	10.83	3.09	
349	93.1	3.75	27.0	75.3	8.68	3.11	
$361\frac{1}{2}$	129.9	2.782	20.0	41.3	6.43	3.12	↑
379	194.8	1.946	14.0	19.82	4.45	3.14	Mercury manometer
356	171.2	2.080	15.0	22.94	4.79	3.13	Water manometer ↓
370	224.2	1.650	11.9	14.32	3.79	3.14	
396	331.9	1.194	8.6	7.38	2.72	3.17	
389	394.2	.986	7.1	5.06	2.25	3.16	G.P.M. = $3.12\sqrt{H}$
						3.12	AVERAGE

REPUBLIC FLOW METER DATA - VALUES FROM SPITZGLASS

$$\text{G.P.M.} = 100 \times C_v \times D^2 \times \frac{K_2}{60} \times \sqrt{H}$$

C_v = Experimental Coeff.

D^2 = Pipe Inside Diam. Squared

K_2 = Density factor

$\sqrt{H}$ = As above

Small Orifice I.D. = 15/16 inches - $100 \times .13 \times 4.272 \times \frac{3.405}{60} \times \sqrt{H} = 3.14\sqrt{H}$

Large Orifice I.D. = 1.344 inches - $100 \times .295 \times 4.272 \times \frac{3.405}{60} \times \sqrt{H} = 7.13\sqrt{H}$

RPM	Wt. on Dyn'r. Arm-lb	Dynam. Press. In. of Water (H)	Stat. Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Effic. %	$\sqrt{H}$	Remarks
									Full Impeller
750	.90	.00	23.3	0.00	.225	.000	00.0	.00	
"	1.00	4.80	22.9	6.84	.250	.039	15.8	2.19	
"	1.25	37.22	22.4	19.04	.312	.108	34.5	6.10	
"	1.41	102.7	21.5	31.6	.352	.171	48.5	10.13	
"	1.66	199.0	19.9	44.0	.415	.221	53.2	14.11	
1000	1.35	.00	39.0	0.00	.450	.000	00.0	.00	
"	1.49	5.24	38.9	7.14	.497	.702	14.2	2.29	
"	1.82	36.17	38.6	18.75	.607	.182	30.1	6.01	
"	2.15	108.0	37.6	32.4	.717	.307	42.8	10.39	
"	2.36	166.0	36.5	40.2	.787	.371	47.1	12.88	
"	2.80	350.6	34.3	58.4	.934	.505	54.1	18.72	
"	3.00	104.0	29.6	72.7	1.000	.542	54.3	10.20	Large Orifice
"	3.30	172.0	24.8	93.5	1.100	.584	53.2	13.11	" "
1200	1.72	.00	55.6	0.00	.688	.000	00.0	.00	
"	2.00	4.97	55.1	6.95	.800	.097	12.1	2.23	
"	2.42	38.0	55.1	19.22	.968	.268	27.7	6.16	
"	2.80	102.4	54.7	31.6	1.120	.436	39.0	10.12	
"	3.02	152.0	53.1	38.5	1.208	.517	42.7	12.33	
"	3.66	343.0	51.5	57.8	1.465	.756	51.2	18.52	
"	4.03	507.0	49.0	70.3	1.613	.869	53.9	22.52	
"	4.30	154.0	43.2	88.5	1.720	.963	56.0	12.41	Large Orifice
"	4.68	249.0	35.4	112.5	1.873	1.004	53.6	15.78	" "
1400	2.32	.00	75.8	0.00	1.082	.000	00.0	.00	
"	2.62	4.87	75.6	6.89	1.222	.132	10.8	2.21	
"	3.10	38.12	76.1	19.25	1.447	.370	25.6	6.17	
"	3.51	111.4	73.3	32.9	1.638	.609	37.2	10.56	
"	3.86	171.0	72.3	40.8	1.801	.746	41.4	13.08	
"	4.58	347.0	70.6	58.1	2.140	1.035	48.4	18.63	
"	5.40	682.0	66.2	81.5	2.620	1.365	52.1	26.12	
"	5.89	222.0	57.9	106.2	2.748	1.555	56.5	14.90	Large Orifice
"	6.42	349.0	47.9	133.1	2.996	1.612	53.9	18.68	" "
1550	2.83	.00	93.1	0.00	1.462	.000	00.0	.00	
"	3.12	5.43	93.1	7.27	1.612	.178	11.1	2.33	
"	3.61	34.24	93.1	18.28	1.866	.448	24.0	5.85	
"	4.19	113.2	91.4	33.2	2.165	.766	35.3	10.64	
"	4.56	173.2	90.6	41.1	2.360	.937	39.7	13.16	
"	5.35	350.2	88.7	58.4	2.765	1.307	47.2	18.71	
"	6.70	853.0	82.0	91.1	3.460	1.888	54.5	29.21	
"	7.36	275.0	73.3	118.2	3.805	2.190	57.5	16.58	Large Orifice
"	8.00	431.0	59.7	148.0	4.130	2.234	54.0	20.76	" "

RPM	Wt.on Dyn'r Arm-lb	Dynam. Press. In. of Water (H)	$\sqrt{H}$	Stat. Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Eff. %	Remarks
									2/3 Impeller
750	.90	0.0	0.00	21.9	0.0	.225	.000	00.0	
"	1.47	112.0	10.58	18.2	33.0	.367	.152	41.3	Type "A"
"	1.54	166.0	12.88	16.6	40.2	.385	.168	43.8	Alteration
"	1.57	55.1	7.42	12.8	53.0	.392	.171	43.7	Large Orifice
"	1.51	69.9	8.36	10.6	59.6	.377	.160	42.3	" "
1000	1.31	0.0	0.00	37.8	0.0	.437	.000	00.0	
"	2.13	100.6	10.03	34.1	31.3	.710	.270	38.0	
"	2.31	166.0	12.88	32.2	40.2	.770	.327	42.5	
"	2.50	293.0	17.12	28.6	53.5	.834	.386	46.3	
"	2.47	94.0	9.69	21.5	69.0	.824	.374	45.5	Large Orifice
"	2.60	122.0	11.05	17.8	78.7	.867	.354	40.8	" "
1200	1.84	0.0	0.00	54.6	0.0	.735	.000	00.0	
"	2.86	103.0	10.15	50.7	31.6	1.144	.405	35.4	
"	3.00	167.0	12.92	48.6	40.3	1.200	.495	41.2	
"	3.50	422.0	20.54	41.2	64.1	1.400	.667	47.7	
"	3.77	138.0	11.75	32.8	83.8	1.509	.694	46.0	Large Orifice
"	3.85	179.0	13.38	26.0	95.3	1.540	.626	40.7	" "
1400	2.44	0.0	0.00	74.8	0.0	1.138	.000	00.0	
"	3.65	102.0	10.10	69.4	31.5	1.703	.552	32.4	
"	3.85	167.0	12.92	67.2	40.3	1.796	.684	38.1	
"	4.30	335.0	18.30	62.4	57.1	2.005	.900	44.8	
"	4.69	571.0	23.90	55.1	74.6	2.188	1.038	47.5	
"	4.88	186.0	13.64	43.6	97.3	2.278	1.070	47.0	Large Orifice
"	5.01	246.0	15.68	35.0	111.8	2.340	.988	42.2	" "
1580	2.93	0.0	0.00	93.2	0.0	1.543	.000	00.0	
"	4.30	104.0	10.20	87.8	31.8	2.265	.705	31.1	
"	4.57	165.0	12.85	85.7	40.1	2.405	.868	36.1	
"	5.10	349.0	18.68	80.5	58.3	2.685	1.185	44.1	
"	5.84	725.0	26.93	69.4	84.0	3.075	1.470	47.9	
"	6.35	248.0	15.75	55.0	112.3	3.345	1.560	46.6	Large Orifice
"	6.60	316.0	17.78	45.2	126.8	3.475	1.447	41.7	" "

RPM	Wt. on Dyn'r Arm-lbs.	Dynam. Press. In. of Water (H)	$\sqrt{H}$	Stat. Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Eff. %	Remarks
									$\frac{1}{2}$ Impeller
750	.91	.00	.00	21.8	0.0	.228	.0000	00.0	
"	1.02	14.52	3.81	20.4	11.9	.255	.0613	24.0	Type "A"
"	1.11	37.12	6.09	19.2	19.0	.277	.0921	33.2	Alteration
"	1.37	104.0	10.20	15.8	31.8	.342	.1269	37.1	
"	1.31	35.0	5.92	12.4	42.2	.327	.1320	40.3	Large Orifice
"	1.30	51.9	7.20	8.3	51.4	.325	.1083	33.3	" "
1000	1.38	.00	.00	37.5	0.0	.460	.0000	00.0	
"	1.55	14.67	3.83	35.4	11.9	.517	.1069	20.7	
"	1.71	36.12	6.01	34.1	18.7	.570	.1615	28.3	
"	2.02	107.4	10.36	30.6	32.3	.673	.2496	37.1	
"	2.14	165.0	12.85	27.8	40.1	.713	.2815	39.4	
"	2.22	242.0	15.56	23.7	48.5	.740	.2900	39.2	
"	2.17	66.2	8.14	19.3	58.0	.724	.2826	39.0	Large Orifice
"	2.15	91.0	9.54	13.7	68.0	.717	.2354	32.8	" "
1200	1.77	.00	.00	53.5	0.0	.708	.000	00.0	
"	2.09	14.52	3.81	51.0	11.9	.836	.153	18.3	
"	2.28	36.77	6.06	49.9	18.9	.912	.238	26.1	
"	2.62	107.4	10.36	46.1	32.3	1.048	.376	35.9	
"	2.82	170.0	13.04	43.3	40.7	1.128	.445	39.4	
"	3.04	354.0	18.81	34.6	58.7	1.216	.513	42.2	
"	3.02	95.0	9.75	27.9	69.5	1.208	.490	40.5	Large Orifice
"	3.01	131.0	11.45	19.3	81.6	1.205	.398	33.0	" "
1400	2.35	.00	.00	72.5	0.0	1.097	.000	00.0	
"	2.71	14.62	3.82	69.6	11.9	1.264	.210	16.6	
"	2.92	35.62	5.97	68.0	18.6	1.363	.319	23.4	
"	3.42	109.4	10.46	64.3	32.6	1.596	.529	33.2	
"	3.62	168.0	12.96	61.4	40.4	1.690	.626	37.0	
"	4.05	475.0	21.79	46.3	68.0	1.890	.795	42.1	
"	4.07	132.2	11.50	36.2	82.0	1.900	.750	39.5	Large Orifice
"	4.06	182.0	13.49	26.1	96.1	1.895	.634	33.4	" "
1580	2.85	.00	.00	92.7	0.0	1.500	.000	00.0	
"	3.32	14.37	3.79	88.7	11.8	1.749	.265	15.1	
"	3.66	37.62	6.13	87.4	19.1	1.928	.422	21.9	
"	4.12	110.0	10.49	83.3	32.7	2.170	.688	31.7	
"	4.34	166.0	12.88	80.5	40.2	2.284	.817	35.8	
"	4.82	336.0	18.33	72.7	57.2	2.540	1.050	41.3	
"	5.16	614.0	24.78	58.9	77.3	2.720	1.150	42.3	
"	5.11	184.0	13.56	44.2	96.6	2.690	1.078	40.1	Large Orifice
"	5.22	231.5	15.22	32.8	108.5	2.750	.899	32.7	" "

RPM	Wt. on Dyn'r. Arm-Lbs	Dynam. Press. In. of Water (H)	$\sqrt{H}$	Stat. Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Eff. %	Remarks
									1/3 Impeller
750	.85	.00	.00	21.10	0.0	.212	.0000	00.0	Type "A"
"	.93	14.37	3.79	18.50	11.8	.232	.0552	23.8	Alteration
"	1.03	36.52	6.04	15.80	18.8	.258	.0752	29.2	
"	1.15	97.00	9.85	11.20	30.7	.288	.0868	30.2	
1000	1.29	.00	.00	36.4	0.0	.430	.0000	00.0	
"	1.48	14.92	3.86	32.7	12.0	.493	.0995	20.2	
"	1.58	35.87	5.99	29.9	18.7	.527	.1412	26.8	
"	1.78	107.0	10.34	22.9	32.3	.593	.1870	31.6	
"	1.79	169.0	13.00	17.0	40.6	.597	.1743	29.2	
1200	1.75	.00	.00	52.0	0.0	.700	.000	00.0	
"	2.02	14.62	3.82	48.0	11.9	.808	.145	17.9	
"	2.17	36.87	6.07	44.8	18.9	.868	.214	24.7	
"	2.43	105.0	10.25	37.5	32.0	.972	.303	31.2	
"	2.47	166.0	12.88	31.7	40.2	.988	.322	32.6	
"	2.48	246.0	15.68	24.2	48.9	.992	.299	30.2	
1400	2.26	.00	.00	70.2	0.0	1.055	.000	00.0	
"	2.57	14.77	3.84	64.8	12.0	1.200	.196	16.4	
"	2.76	35.12	5.93	61.5	18.5	1.289	.287	22.3	
"	3.11	109.3	10.45	54.0	32.6	1.451	.445	30.6	
"	3.22	166.0	12.88	48.5	40.2	1.502	.492	32.8	
"	3.29	333.0	18.25	32.7	57.0	1.535	.471	30.6	
1580	2.76	.00	.00	88.9	0.0	1.454	.000	00.0	
"	3.22	15.07	3.88	84.4	12.1	1.697	.258	15.2	
"	3.43	37.12	6.09	80.5	19.0	1.807	.386	21.4	
"	3.77	104.0	10.20	70.9	31.8	1.986	.569	28.6	
"	3.96	165.0	12.85	65.9	40.1	2.085	.667	32.0	
"	4.07	345.0	18.57	48.5	57.9	2.142	.709	33.1	
"	4.08	424.0	20.59	41.2	64.2	2.148	.668	31.1	

RPM	Wt. on Dyn'r. Arm-Lbs.	Dynamic Press. In. of Water H	$\sqrt{H}$	Static Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Eff. %
750	.88	--	--	2.22	--	.220	--	--
"	1.38	94	9.68	17.1	30.2	.345	.131	37.8
"	1.38	146	12.08	14.6	37.7	.345	.139	40.3
"	1.39	* 55	7.42	8.1	52.9	.348	.108	31.0
1000	1.46	--	--	39.3	--	.487	--	--
"	2.17	110	10.47	32.5	32.7	.724	.268	37.1
"	2.25	169	13.00	29.7	40.6	.750	.304	40.6
"	2.30	258	16.06	25.4	50.1	.766	.321	42.0
"	2.28	* 74	8.57	20.1	61.1	.760	.310	40.8
"	2.22	*146	12.08	9.7	86.1	.740	.211	28.6
1200	1.95	--	--	55.1	--	.780	--	--
"	2.84	111	10.51	48.7	32.8	1.136	.404	35.5
"	3.00	170	13.02	46.5	40.6	1.200	.477	39.7
"	3.15	347	18.63	37.3	58.1	1.260	.548	43.5
"	3.14	*112	10.58	26.8	75.4	1.257	.510	40.6
"	3.10	*139	11.79	20.4	84.0	1.240	.433	35.0
1400	2.49	--	--	75.2	--	1.162	--	--
"	3.60	114	10.68	67.6	33.3	1.680	.568	33.8
"	3.80	168	12.96	64.9	40.4	1.773	.662	37.4
"	4.17	501	22.38	48.5	69.8	1.943	.855	44.0
"	4.22	*134	11.59	39.8	82.6	1.970	.830	42.1
"	4.13	*185	13.60	26.9	97.0	1.930	.659	34.2
1580	3.09	--	--	95.4	--	1.628	--	--
"	4.35	115	10.74	86.3	33.5	2.290	.730	31.9
"	4.57	173	13.14	84.0	41.0	2.408	.870	36.1
"	5.02	337	18.36	76.2	57.2	2.644	1.100	41.6
"	5.21	638	25.26	61.5	78.8	2.742	1.224	44.6
"	5.27	*184	13.56	46.6	96.6	2.780	1.137	40.9
"	5.06	*237	15.39	33.9	109.7	2.665	.938	35.2

Test Feb. 7, 1933 - Type "B" Alteration

*Denotes large size orifice.

2/3 of each impeller port open. 1/3 of each blanked off on follower side of port opening. Pump operated smoothly.

RPM	Wt. on Dyn'r. Arm-Lbs.	Dynamic Press. In. of Water H	$\sqrt{H}$	Static Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Eff. %
750	.87	--	--	22.0	--	.217	--	--
"	1.13	107	10.36	11.1	32.3	.282	.091	32.0
1000	1.28	--	--	37.9	--	.427	--	--
"	1.55	14.4	3.79	36.0	11.8	.517	.107	20.8
"	1.70	38.1	6.17	33.4	19.3	.567	.163	28.7
"	1.82	106.8	10.34	26.2	32.3	.607	.214	35.2
"	1.79	189.0	13.75	18.8	42.9	.597	.204	34.2
"	1.64	* 61.6	7.85	9.5	56.0	.547	.134	24.6
1200	1.76	--	--	54.3	--	.704	--	--
"	2.11	15.1	3.88	51.8	12.1	.845	.158	18.7
"	2.27	36.8	6.06	49.6	18.9	.908	.237	26.1
"	2.54	110.7	10.52	42.3	32.8	1.015	.350	34.6
"	2.53	165.0	12.85	36.8	40.1	1.012	.373	36.8
"	2.52	276.0	16.61	27.1	51.8	1.008	.354	35.2
"	2.29	* 89.8	9.47	13.4	59.8	.915	.202	22.1
1400	2.27	--	--	73.3	--	1.060	--	--
"	2.66	13.7	3.70	70.1	11.5	1.240	.204	16.4
"	2.92	38.1	6.17	66.8	19.3	1.362	.326-	23.9
"	3.20	107.4	10.36	60.5	32.3	1.494	.493	33.0
"	3.28	165.1	12.85	54.6	40.1	1.530	.553	36.1
"	3.29	350.0	18.71	37.9	58.4	1.535	.559	36.4
"	3.00	*120.0	10.95	17.7	76.3	1.400	.341	24.4
1580	2.84	--	--	92.9	--	1.498	--	--
"	3.34	14.9	3.86	90.1	12.0	1.760	.273	15.5
"	3.64	38.1	6.17	87.3	19.3	1.919	.425	22.2
"	4.00	112.0	10.58	80.4	33.0	2.108	.670	31.8
"	4.10	168.0	12.96	74.3	40.4	2.160	.758	35.1
"	4.13	475.0	21.79	45.9	67.9	2.176	.786	36.2
"	3.83	*154.4	12.43	22.4	89.6	2.020	.502	24.8

Test Feb. 1, 1933 - Type "B" Alteration

*Denotes large size orifice.

1/2 of each impeller port blanked off on follower side of port opening. Pump operated smoothly.

RPM	Wt. on Dyn'r. Arm-Lbs.	Dynamic Press. In. of Water H	$\sqrt{H}$	Stat. Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Eff. %
750	.89	--	--	21.7	--	.222	--	--
"	1.01	15.7	3.96	16.8	12.4	.252	.052	20.8
"	1.03	37.5	6.13	12.0	19.1	.258	.058	22.4
"	.86	65.0	8.06	7.0	25.2	.215	.045	20.8
1000	1.22	--	--	36.9	--	.407	--	--
"	1.56	16.2	4.03	30.7	12.6	.520	.098	18.8
"	1.58	37.9	6.15	25.9	19.2	.527	.126	23.8
"	1.41	111.2	10.55	11.5	32.9	.470	.096	20.3
1200	1.69	--	--	53.0	--	.676	--	--
"	2.02	15.4	3.92	46.5	12.3	.808	.144	17.8
"	2.15	38.2	6.18	40.6	19.3	.860	.198	23.0
"	2.05	109.4	10.46	25.8	32.6	.820	.212	25.8
"	1.93	160.4	12.66	16.2	39.4	.772	.161	20.9
1400	2.32	--	--	70.8	--	1.082	--	--
"	2.70	15.7	3.96	64.4	12.4	1.260	.201	16.0
"	2.78	35.9	5.99	58.8	18.7	1.298	.278	21.4
"	2.75	107.4	10.36	42.7	32.3	1.283	.348	27.1
"	2.70	166.2	12.90	31.3	40.2	1.260	.318	24.4
"	2.58	220.0	14.83	21.7	46.3	1.204	.254	21.1
1580	2.86	--	--	92.0	--	1.508	--	--
"	3.30	14.9	3.86	83.2	12.1	1.740	.254	14.6
"	3.46	37.4	6.11	77.0	19.1	1.822	.371	20.4
"	3.50	109.4	10.46	60.0	32.6	1.844	.494	26.8
"	3.45	172.0	13.11	47.5	40.9	1.819	.490	27.0
"	3.29	284.0	16.85	27.8	52.5	1.732	.369	21.3

Test Jan. 24, 1933 - Type "B" Alteration

1/3 of each impeller port open. 2/3 of each impeller port blanked off on follower side of port opening. Pump operated smoothly.

RPM	Wt. on Dyn'r. Arm-Lbs.	Dynam. Press. In. of Water H	$\sqrt{H}$	Stat. Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Eff. %
1000	1.39	--	--	36.9	--	.464	--	--
"	2.40	86.1	9.28	37.1	29.0	.800	.272	34.0
"	2.80	169.5	13.02	36.4	40.6	.934	.373	40.0
"	3.38	352	18.76	34.3	58.5	1.127	.506	44.9
"	3.72	116	10.77	30.1	76.8	1.240	.584	47.1
"	4.20	175	13.23	25.4	94.3	1.400	.605	43.2
1200	1.89	--	--	53.1	--	.756	--	--
"	3.08	87	9.33	53.6	29.1	1.231	.394	32.0
"	3.56	165	12.85	53.1	40.1	1.423	.538	37.8
"	4.31	351	18.74	51.4	58.5	1.725	.759	44.0
"	4.71	509	22.56	49.1	70.4	1.882	.873	46.3
"	5.44	198	14.07	40.8	100.2	2.175	1.033	47.5
"	5.93	252	15.87	36.5	113.1	2.370	1.044	44.0
1400	2.45	--	--	72.0	--	1.143	--	--
"	3.84	83.2	9.12	72.4	28.4	1.792	.519	29.0
"	4.37	163	12.77	71.8	39.8	2.040	.722	35.4
"	5.26	352	18.76	70.8	58.5	2.457	1.047	42.6
"	6.29	695	26.36	67.4	82.2	2.935	1.400	47.7
"	7.48	281	16.76	56.0	119.5	3.490	1.690	48.5
"	8.10	345	18.57	49.2	132.4	3.780	1.646	43.5

Test Feb. 16, 1933 - Type "C" Alteration

2/3 of each impeller port open. 1/3 of each blanked off on pressure side of port. On opening pump 1/2", or 1/2 of one blanking piece, was torn off. Remaining ports intact. At higher speeds and deliveries there was an objectionable vibration developed. Excessive loading of motor made a 1580 RPM run inadvisable.

RPM	Wt. on Dyn'r. Arm-Lbs.	Dynam. Press. In. Water H	$\sqrt{H}$	Stat. Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Eff. %
750	.90	000	--	22.9	--	.225	--	--
"	1.78	110	10.49	21.7	32.7	.445	.179	40.2
"	2.16	205	14.32	20.4	44.7	.540	.230	42.6
1000	1.41	--	--	38.7	--	.470	--	--
"	2.68	114	10.68	37.8	33.3	.894	.318	35.6
"	2.91	166	12.88	37.0	40.2	.970	.356	36.7
"	3.60	349	18.68	35.7	58.2	1.200	.525	43.7
1200	2.00	--	--	55.8	--	.800	--	--
"	3.42	114	10.68	55.6	33.3	1.368	.468	34.2
"	3.79	170	13.04	55.2	40.7	1.517	.567	37.4
"	4.41	328	18.11	53.5	56.5	1.762	.764	43.3
"	5.00	525	22.91	50.8	71.5	2.000	.917	45.9
1400	2.56	--	--	76.0	--	1.195	--	--
"	4.36	117	10.83	78.0	33.8	2.035	.666	32.7
"	4.68	173	13.15	76.1	41.0	2.185	.788	36.0
"	5.40	322	17.94	73.1	56.0	2.520	1.034	41.0
"	6.64	707	26.59	68.2	83.0	3.100	1.430	46.1

Test Feb. 10, 1933 - Type "C" Alteration

1/2 of each impeller port blanked off on pressure, or driving side, of the port. When the pump was disassembled two port blankings were found perfect, two had 8% of each torn off, one had 17% torn off, and one had 60% torn off. As a whole the impeller was 85% intact.

RPM	Wt. on Dyn'r. Arm-Lbs.	Dynam. Press. In. of Water H	$\sqrt{H}$	Stat. Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Eff. %
750	.82	--	--	20.8	--	.205	--	--
"	1.92	146	12.08	14.7	37.7	.480	.140	29.2
1000	1.23	--	--	35.7	--	.410	--	--
"	2.52	111	10.52	30.9	32.9	.840	.257	30.6
"	2.84	168	12.96	28.6	40.4	.946	.292	30.8
"	3.23	254	15.94	25.1	49.8	1.077	.316	29.4
"	3.99	*101	10.04	15.0	71.6	1.330	.272	20.4
1200	1.66	--	--	51.4	--	.664	--	--
"	3.21	109	10.45	45.5	32.6	1.283	.374	29.2
"	3.59	169	12.99	43.6	40.5	1.436	.446	31.1
"	4.44	347	18.63	36.5	58.2	1.775	.536	30.2
"	5.22	*116	10.76	28.2	71.8	2.088	.511	24.4
"	5.89	*146	12.08	21.3	86.1	2.355	.463	19.7
1400	2.18	--	--	69.6	--	1.018	--	--
"	4.00	114	10.64	63.4	33.2	1.868	.531	28.5
"	4.44	175	13.22	61.0	41.3	2.072	.636	30.7
"	6.08	505	22.47	49.0	70.2	2.840	.869	30.6
1580	2.73	--	--	89.4	--	1.439	--	--
"	4.72	106	10.26	82.8	32.0	2.483	.669	26.9
"	5.26	175	13.22	79.8	41.3	2.771	.832	30.0
"	6.27	333	18.23	73.3	57.0	3.302	1.055	31.9
"	7.75	652	25.54	62.6	79.7	4.080	1.261	30.9

Test Feb. 14, 1933 - Type "C" Alteration

* Denotes large orifice.

3/4 of each port opening was blanked off on the pressure, or driving side, of each impeller port. A heavy hum and vibration developed at higher water flow rates. For this reason the test was not extended into as high a flow region as in some of the other tests. All blanking was intact at end of test.

RPM	Wt. on Dyn'r. Arm-Lbs.	Dynam. Press. In. of Water H	$\sqrt{H}$	Stat. Press. Ft. of Water	GPM	BHP	WHP	Overall Pump Eff. %
1000	1.35	--	--	37.3	--	.450	--	--
"	2.10	88	9.37	33.3	29.2	.700	.246	35.1
"	2.33	256	16.00	25.2	49.9	.776	.318	40.9
"	2.58	*101	10.04	22.3	71.6	.860	.403	46.9
"	2.69	*126	11.23	18.5	80.1	.896	.374	41.8
1200	1.85	--	--	53.2	--	.740	--	--
"	2.72	86	9.26	51.6	28.9	1.088	.377	34.6
"	3.05	168	12.94	48.9	40.4	1.220	.499	40.9
"	3.48	426	20.64	41.2	64.5	1.392	.670	48.1
"	3.60	*124	11.12	34.5	79.4	1.440	.692	48.0
"	3.73	*182	13.49	26.3	96.1	1.491	.639	42.8
1400	2.38	--	--	72.1	--	1.110	--	--
"	3.43	88	9.37	69.9	29.2	1.600	.515	32.2
"	3.84	171	13.07	67.9	40.7	1.791	.697	38.9
"	4.31	342	18.49	62.7	57.6	2.010	.912	43.2
"	4.66	575	23.98	55.5	74.8	2.175	1.049	48.2
"	4.93	*187	13.67	45.0	97.4	2.300	1.108	48.1
"	5.15	*249	15.78	35.9	112.5	2.400	1.020	42.5

Test Feb. 21, 1933 - Type "D" Alteration

* Denotes large size orifice.

Three port openings of the impeller were closed off on the inside, or suction entrance, of the ports. However, on disassembling the pump, one blanking strip was loose on one end such that two ports, only, were effectively blanked off.

This test compares with the 2/3 impeller test with ports fully blanked off on outside port opening. The pump operated smoothly regardless of the unbalance due to the one loosened blanking strip.

RPM	Torq. Angle Deg.	Press. Ft. of Water	Suc. Ft.of Water	H Total Ft.of Water	Q Dischg Sec. Ft. of Water	BHP	WHP	Eff. %	Remarks
800	6.6	37.3	6.8	44.1	--	3.46	--	--	Normal Pump Test
"	9.6	37.9	7.2	45.1	.37	5.04	1.89	37.4	Apr. 12, 1933
"	11.0	38.0	7.3	45.3	.58	5.78	3.00	51.8	Full Capacity
"	12.8	37.0	8.1	45.1	.82	6.72	4.21	62.6	
"	16.0	34.9	9.1	44.0	1.19	8.40	5.92	70.4	At 800 RPM H.P =
"	18.4	31.0	10.6	41.6	1.53	9.66	7.20	74.5	.525 x Torque angle
"	20.0	27.4	10.1	37.5	1.89	10.50	8.00	76.0	At 940 RPM H.P. =
"	21.7	21.3	12.7	34.0	2.22	11.40	8.53	74.9	.617 x Torque angle
"	22.4	14.5	14.3	28.8	2.54	11.76	8.27	70.4	
"	23.2	10.2	14.9	25.1	2.78	12.19	7.88	64.7	WHP = .113 x Q.H.
940	9.2	54.6	6.9	61.5	--	5.67	--	--	
"	12.6	55.9	7.3	63.2	.40	7.78	2.86	36.7	
"	14.3	56.1	7.4	63.5	.57	8.82	4.12	46.8	
"	16.6	55.3	7.7	63.0	.84	10.24	5.98	58.4	
"	20.8	54.6	8.3	62.9	1.24	12.83	8.81	68.6	
"	24.2	50.0	9.4	59.4	1.67	14.93	11.20	75.0	
"	26.9	44.2	10.6	54.8	2.05	16.60	12.70	76.5	
"	28.5	39.8	11.7	51.5	2.33	17.60	13.56	77.0	

800	6.5	36.2	7.6	43.8	--	3.41	--	--	Normal Pump Test
"	9.8	36.6	8.1	44.7	.40	5.15	2.02	39.2	
"	11.2	36.3	8.3	44.6	.61	5.88	3.07	52.3	Apr. 16, 1933
"	13.7	36.3	8.6	44.9	.91	7.19	4.61	64.2	Full Capacity
"	16.4	35.1	9.1	44.2	1.26	8.61	6.30	73.1	
"	18.8	31.1	9.9	41.0	1.62	9.87	7.51	76.1	
"	20.8	24.5	11.3	35.8	2.05	10.92	8.30	75.9	
"	22.1	17.1	13.2	30.3	2.44	11.60	8.50	73.2	
"	22.8	10.2	14.8	25.0	2.75	11.98	7.77	64.8	
940	9.0	53.0	7.9	60.9	--	5.55	--	--	
"	12.8	54.0	8.4	62.4	.41	7.89	2.89	36.6	
"	14.2	54.4	8.4	62.8	.60	8.76	4.25	48.6	
"	17.5	54.3	8.8	63.1	.92	10.80	6.80	62.9	
"	20.4	53.8	9.2	63.0	1.20	12.59	8.55	68.0	
"	24.0	49.5	10.3	59.8	1.64	14.80	11.09	75.0	
"	26.7	43.9	11.4	55.3	2.03	16.48	12.70	77.1	
"	29.5	36.2	13.1	49.3	2.47	18.20	13.80	75.8	

RPM	Torq. Angle Deg.	Press. Ft. of Water	Suc. Ft. of Water	Total Ft. of Water	Dischg Sec. Ft. of Water	BHP	WHP	Eff. %	Remarks
800	6.2	30.0	11.6	41.6	--	3.25	--	--	Test Apr. 20, 1933
"	8.6	29.7	12.0	41.7	.40	4.51	1.88	41.7	Two ports closed.
"	10.0	28.5	12.1	40.6	.58	5.25	2.66	50.7	
"	12.0	27.0	12.4	39.4	.90	6.30	4.01	63.6	Pump altered to
"	14.0	24.1	12.8	36.9	1.20	7.35	5.00	68.0	2/3 capacity.
"	15.2	18.8	13.5	32.3	1.52	7.98	5.56	69.7	Type "A" Altera-
"	15.8	11.2	14.6	25.8	1.80	8.30	5.26	63.3	tion.
"	15.8	6.6	15.6	22.2	1.92	8.30	4.82	58.0	
940	8.0	46.3	11.6	57.9	--	4.93	--	--	
"	11.1	47.5	12.0	59.5	.40	6.85	2.69	39.2	
"	12.8	46.6	12.1	58.7	.58	7.90	3.85	48.7	
"	15.4	44.5	12.4	56.9	.90	9.50	5.78	60.9	
"	17.8	42.0	12.8	54.8	1.20	10.99	7.43	67.6	
"	19.6	36.9	13.4	50.3	1.49	12.09	8.47	70.1	
"	21.0	30.1	14.3	44.4	1.83	12.96	9.20	71.0	
"	21.6	20.7	15.3	36.0	2.09	13.32	8.50	63.8	

RPM	Torq. Angle Deg.	Press. Ft. of Water	Suc. Ft. of Water	Total Ft. of Water	Dischg Sec. Ft. of Water	BHP	WHP	Eff. %	Remarks
800	5.7	32.0	8.6	40.6	--	2.99	--	--	Test Apr. 21, 1933
"	8.3	31.2	9.0	40.2	.40	4.41	1.82	41.2	Three ports closed.
"	10.0	29.9	9.2	39.1	.61	5.25	2.69	51.3	Pump altered to
"	10.8	28.4	9.3	37.7	.79	5.67	3.36	59.7	1/2 capacity
"	11.7	24.7	9.6	34.3	1.01	6.15	3.92	63.6	
"	12.2	20.3	9.8	30.1	1.20	6.40	4.08	63.7	Type "A" Altera-
"	12.4	14.0	10.2	24.2	1.40	6.51	3.83	58.7	tion.
"	12.0	6.4	11.8	18.2	1.60	6.30	3.29	52.2	
940	7.4	47.5	8.7	56.2	--	4.56	--	--	
"	10.6	46.5	9.1	55.6	.40	6.54	2.51	38.4	
"	12.0	45.7	9.2	54.9	.60	7.40	3.72	50.3	
"	13.5	44.7	9.4	54.1	.80	8.33	4.90	58.8	
"	15.1	41.6	9.6	51.2	1.02	9.31	5.90	63.4	
"	16.1	37.7	9.8	47.5	1.20	9.94	6.44	64.8	
"	16.7	31.9	10.2	42.1	1.40	10.30	6.66	64.7	
"	17.0	25.5	10.7	36.2	1.60	10.50	6.55	62.3	
"	17.1	16.7	11.5	28.2	1.80	10.55	5.86	55.5	
"	16.4	5.3	12.5	17.8	1.90	10.11	3.82	37.8	

RPM	Torq. Angle Deg.	Press. Ft. of Water	Suc. Ft.of Water	Total Ft.of Water	Dischg Sec. Ft.of Water	BHP	WHP	Eff. %	Remarks
800	5.3	28.4	11.7	40.1	--	2.78	--	--	Test Apr. 19, 1933
"	7.8	25.3	12.1	37.1	.41	4.10	1.72	41.9	Four ports closed.
"	9.0	22.2	12.3	34.5	.62	4.72	2.44	51.6	Pump altered to
"	9.7	18.4	12.4	30.8	.80	5.09	2.78	54.7	1/3 capacity.
"	10.0	11.2	12.6	23.8	1.01	5.25	2.72	51.7	
"	10.0	3.8	13.2	17.0	1.13	5.25	2.17	41.3	Type "A" Alteration.
940	6.9	43.4	11.7	55.1	--	4.26	--	--	
"	10.0	41.5	12.1	53.6	.43	6.17	2.60	42.2	
"	11.0	39.5	12.2	51.7	.61	6.79	3.56	52.5	
"	12.0	34.5	12.3	46.8	.80	7.40	4.23	56.2	
"	13.0	28.5	12.6	41.1	1.03	8.02	4.79	59.6	
"	13.2	20.5	12.8	33.3	1.20	8.15	4.52	55.4	
"	12.8	6.5	13.4	19.9	1.34	7.90	3.01	38.1	
940	7.0	45.3	11.5	56.8	--	4.32	--	--	Test for Check
"	9.8	42.2	11.9	54.1	.38	6.04	2.26	37.4	Apr. 20, 1933
"	10.9	39.1	12.0	51.1	.60	6.72	3.46	51.6	
"	12.4	35.5	12.0	47.5	.80	7.65	4.30	56.1	
"	13.0	28.8	12.3	41.1	1.01	8.02	4.69	58.5	
"	13.2	20.3	12.5	32.8	1.19	8.15	4.41	54.1	
"	12.8	5.7	13.0	18.7	1.36	7.90	2.87	36.3	

	940 REVS. PER MIN.				800 REVS. PER MIN.			
	Dischg. Cu.Ft. per Sec.	BHP	WHP	Total Pump & P. Line Eff. %	Dischg. Cu.Ft. per Sec.	BHP	WHP	Total Pump & P. Line Eff. %
Full Impeller	.4	7.82	1.60	20.4	.4	5.04	1.60	31.8
	.8	10.10	3.30	32.7	.8	6.65	3.30	49.7
	1.2	12.40	5.24	42.3	1.2	8.32	5.24	63.0
	1.6	14.56	7.50	51.5	1.4	9.12	6.31	69.2
	2.0	16.30	10.17	62.4	1.6	9.90	7.50	75.7
	2.4	17.85	13.39	75.0				
2/3 Impeller	.4	6.95	1.60	23.0	.4	4.60	1.60	34.8
	.8	8.97	3.30	36.8	.6	5.30	2.43	45.8
	1.2	10.97	5.24	48.0	.8	6.00	3.30	55.1
	1.6	12.50	7.50	60.0	1.0	6.64	4.24	63.8
	1.8	13.00	8.76	67.5	1.1	7.00	4.73	67.6
1/2 Impeller	.4	6.46	1.60	24.8	.2	3.67	.79	21.5
	.8	8.38	3.30	39.4	.4	4.46	1.60	35.9
	1.0	9.30	4.24	45.5	.6	5.25	2.43	46.3
	1.2	9.92	5.24	52.8	.8	5.80	3.30	57.0
	1.4	10.38	6.31	60.8				
1/3 Impeller	.4	6.02	1.60	26.6	.2	3.44	.79	23.0
	.6	6.80	2.43	35.8	.4	4.10	1.60	39.0
	.8	7.44	3.30	44.5	.5	4.42	2.01	45.5
	1.0	7.95	4.24	53.3				
	1.1	8.10	4.73	58.5				

RPM	Torq. Angle Deg.	Press. Ft. of Water	Suc. Ft.of Water	Total Ft.of Water	Dischg Sec. Ft. of Water	BHP	WHP	Eff. %
800	5.8	31.3	9.8	41.1	--	3.04	--	--
"	9.2	32.8	10.2	43.0	.40	4.83	1.94	40.2
"	11.0	33.3	10.4	43.7	.58	5.75	2.86	49.8
"		32.6	10.7	43.3				
"	13.5	34.7	10.5	45.2	.84	7.09	4.30	60.6
800	5.6	31.0	10.1	41.1	--	2.94	--	--
"	9.3	32.8	10.4	43.2	.40	4.88	1.95	40.0
"	11.0	33.2	10.6	43.8	.60	5.75	2.97	51.7
"		32.5	10.9	43.4				
"	13.2	34.7	10.7	45.4	.79	6.93	4.04	58.5
"	16.5	32.6	11.3	43.9	1.20	8.66	5.95	68.7
"	20.0	28.5	12.1	40.6	1.62	10.50	7.44	70.6
"	21.8	23.3	13.2	36.5	1.98	11.90	8.17	68.6
"	23.5	17.1	14.6	31.7	2.33	12.34	8.35	67.6
"	24.4	8.6	16.7	25.3	2.70	12.81	7.71	60.2
940	7.6	48.5	10.0	58.5	--	4.69	--	--
"	11.9	51.6	10.3	61.9	.39	7.34	2.73	37.2
"	14.4	53.1	10.5	63.6	.60	8.88	4.31	48.6
"	18.0	54.0	10.9	64.9	.90	11.10	6.60	59.5
"	20.8	53.0	11.6	64.6	1.20	12.83	8.76	68.3
"	25.0	48.4	12.7	61.1	1.62	15.41	11.20	72.6
"	29.0	42.5	14.3	56.8	2.05	17.90	13.17	73.5

Test April 23, 1933 - Type "E" Alteration

2/3 of periphery opening of impeller closed off by means of two bands, each 1/2 inch wide, being fastened around the circumferential openings of the impeller. Due to the discontinuity in the data the test was stopped and the pump dismantled to see if the bands had broken. They were intact and satisfactory, so the pump was assembled again and the two complete tests run April 24, 1933. The same discontinuity appears in the second set of data also, in the 800 R.P.M. run.

RPM	Torq. Angle Deg.	Press. Ft. of Water	Suc. Ft.of Water	Total Ft.of Water	Dischg Sec. Ft. of Water	BHP	WHY	Eff. %
800	5.0	32.8	6.6	39.4	--	2.62	--	--
"	9.3	35.7	7.3	43.0	.40	4.88	1.94	39.8
"	11.8	35.9	7.9	43.8	.61	6.19	3.02	48.8
"	15.1	33.8	9.0	42.8	.93	7.92	4.50	56.8
"	17.2	28.0	10.5	38.5	1.20	9.02	5.22	57.9
"	20.0	20.0	13.0	33.0	1.53	10.50	5.70	54.4
"	22.7	11.7	10.4	22.1	1.96	11.91	4.90	41.1
800	5.2	33.5	6.2	39.7	--	2.73	--	--
"	9.2	36.5	6.5	43.0	.40	4.83	1.94	40.2
"	11.8	37.6	6.4	44.0	.58	6.19	2.88	46.6
"	14.9	36.5	7.0	43.5	.90	7.82	4.42	56.5
"	17.2	32.1	7.5	39.6	1.17	9.02	5.24	58.1
"	20.5	23.6	8.4	32.0	1.57	10.78	5.68	52.8
"	22.8	12.3	10.0	22.3	1.95	11.98	4.91	41.0
940	7.0	50.4	6.2	56.6	--	4.32	--	--
"	11.4	54.1	6.5	60.6	.39	7.04	2.67	38.0
"	14.4	55.0	6.7	61.7	.57	8.89	3.98	44.8
"	19.0	55.7	6.9	62.6	.92	11.71	6.51	55.5
"	22.0	51.5	7.4	58.9	1.22	13.58	8.12	59.9
"	25.6	45.5	8.1	53.6	1.55	15.80	9.39	59.5
"	28.8	35.9	9.0	44.9	1.90	17.78	9.64	54.3
"	30.4	31.4	9.6	41.0	2.06	18.75	9.55	50.9
		16.0	11.5	27.5	2.45			

Test May 19, 1933 - Type "E" Alteration

5/6 of impeller closed off by means of two bands, each 5/8 inches wide, placed around the periphery of the vanes, leaving the 1/4 inch opening between them. The first test at 800 R.P.M. was run with the suction valve almost closed. The test was repeated with the valve fully opened. Both values are plotted and but little discrepancy exists between them.

CALIBRATION OF TORSION DYNAMOMETER

March 24, 1933

TORSION LOAD IN INCH POUNDS	INCHES OF ARC DEFLECTION	DEGREES DEFLECTION	REMARKS
1250	6.70	30.29	Length of Arc Arm = 11.07"
1027	5.53	25.00	
806	4.35	19.66	Diameter of Collar = 3.25"
589	3.17	14.33	Total Radius of Measuring Arc = 12.965"
364	1.99	9.00	
153	.83	3.75	
0	.00	.00	
0	.00	.00	
200	1.07	4.84	
422	2.24	10.12	
645	3.42	15.47	
865	4.59	20.74	
1081	5.77	26.08	
1254	6.71	30.31	

Average Value 1252 Inch Pounds Deflects 30.30 Degrees

One Degree Deflection = 41.33 Inch Lbs.

$$\text{Horse Power} = \frac{\text{Rev. Per Min.} \times 41.33 \times 2\pi \times \text{Degrees Deflection}}{12 \times 33000} = \frac{\text{RPM} \times \text{Degrees}}{1523}$$

For 800 R.P.M. $\quad \text{H.P.} = \dfrac{800 \times \text{Degrees}}{1523} = .525 \times \text{Degrees Deflection}$

For 940 R.P.M. $\quad \text{H.P.} \quad \dfrac{940 \times \text{Degrees}}{1523} = .617 \times \text{Degrees Deflection}$