INVESTIGATION OF THE INSTANTANEOUS VELOCITY DISTRIBUTION IN THE VOLUTE OF A CENTRIFUGAL PUMP

Thesis by

Raymond C. Binder

In Partial Fulfillment of the Requirements
for the Degree of Doctor of Philosophy

California Institute of Technology,
Pasadena, California,

1936

SUMMARY

A technique was developed for measuring the instantaneous velocity vectors in the volute of a centrifugal pump. The development resulted in three major instruments- a special Pitot tube, a special differential gage, and a sampling valve. The special differential gage has many advantages over the ordinary type of differential gage, and has many other applications besides the one in this study. Experimental checks have demonstrated the reliability of this new technique of measurement.

Extensive measurements were made on two high efficiency centrifugal pumps of commercial design, one a double suction and the other a single suction type. Pitot stations were provided around the entire volute of the single suction pump. Tests on these pumps showed the following: 1)There was practically no instantaneous velocity variation between vanes at normal pump capacity. There was a slight variation at low and high pump capacities. 2)There was a positive radial velocity outside the shroud in nearly all cases at normal pump capacity. 3)Some double peaked radial velocity profiles were obtained in traverses across the volute at low pump discharge. The most reasonable explanation states that these peaks were due to the centrifuge action of the shroud. 4)The single suction pump test showed considerable inflow at the volute tongue, that is, flow towards the pump shaft completely across the volute. 5)At low, normal, and high discharges the flow conditions were not uniform around the volute. 6)The flow conditions at normal capacity were more uniform than at low and high capacities. 7)There were two unbalanced radial forces acting on the impeller, a static pressure force and a momentum force. The momentum force was small as compared with the unbalanced static pressure force.

ACKNOWLEDGEMENT

The writer expresses his deep appreciation to those who have aided in this investigation. Prof. R.T.Knapp has kindly given advice and help throughout this lengthy task, and Prof. von Kármán has helped towards explaining some of the results.

The writer is very grateful to the Metropolitan Water District of Southern California in many respects- for the use of the pump laboratory at the District's expense, for paying for the special apparatus, for help in some calculating and some plotting of the results, and for permission to use the experimental results for this thesis.

The writer is very grateful to the Byron Jackson Pump Co. and the Worthington Pump and Mach. Co. for permission to use the results of tests on their pumps. The Byron Jackson Co. through its Chief Engineer, Mr. A.Hollander, and the Worthington Pump and Mach. Co. through its Chief Engineer, Mr. M. Spillman, have kindly granted this permission.

TABLE OF CONTENTS

INDEX OF FIGURES

INDEX OF FIGURES (CONTINUED)

FIGURE PAGE

INTRODUCTION

There is need for more experimental data as to the flow conditions in a centrifugal pump. Because of a lack of both qualitative and quantitative information pump designers have been forced to speculate as to the flow character of the water in the impeller, volute, and other regions inside the pump. There is no doubt but what actual measurements on a standard high efficiency volute pump would be of considerable interest to the pump designer and the student of hydrodynamics.

As for previous work, theoretical studies of the impeller flow have been made by Kucharski,[1*] Busemann[2], Schultz[3], Sőrenson[4], and Uchimaru and Kito[5]. Each of these investigators assumed a perfect fluid and used the method of potential theory. Pfleiderer[6] calculated the theoretical head developed by a pump with the condition that the relative exit angle was less than the vane angle. Kucharski[1] treated mathematically the problem of an impeller with straight radial vanes, assuming an ideal fluid. Spannhake[7] pointed out that the fluid passages, formed by curved vanes of a finite length and cut off by entrance and exit circles, as found in actual practice, present many difficulties to theoretical investigation. It is well known that the actual values of the absolute velocity (in direction and magnitude) do not agree with the theoretical calculated values based on potential flow. Fischer and Thoma[8] concluded that "Practically

---

*Reference numbers refer to those in Bibliography at end of this report.

all flow conditions for an actual fluid are fundamentally different from those theoretically derived for an ideal frictionless fluid".

Prof. R.L.Daugherty[9] has made a combination experimental and theoretical analysis in his book "Centrifugal Pumps". Of particular interest is his pointing out that the vane angle and the actual relative exit angle may differ by from five to ten degrees.

Probably one of the first(if not the first) experiments to study the flow conditions inside a rotating hydraulic machine was that of Francis[10] (1851). In his test on a Tremont turbine he inserted a vane at the discharge of the runner, which gave the direction of the water leaving the wheel.

Qualitative photographic experiments have been made by Fischer and Thoma[8,11], Oertli[12], Siess[13], and others. Oertli[12] showed that the flow in a centrifugal impeller is not exactly two dimensional. The work of Fischer and Thoma was extended by photographic experiments made by Closterhalfen[14] who used a pump with a transparent case, and measured the pressure at some points along the vanes. These photographic experiments were made on special laboratory pumps specially prepared for observation purposes. The question has been raised as to whether results from such a technique can be extrapolated to a consideration of a pump of commercial design.

Yendo[15] used pressure measuring holes in the guide vanes of a turbine pump to obtain the slip coefficient. This method does not give a measurement of the magnitude of the absolute velocity.

This year Kasai[16] has reported his Pitot tube measurements in a special laboratory pump having a vortex chamber. His experiments were made from April, 1933 to March, 1934.

With the exception of Kasai, these experiments were made at low speeds, and efficiencies were low. The question has been raised as to whether results from such a technique can be extended to a consideration of a pump of commercial design operating at normal speeds. The work of this present investigation avoided the difficulties of low speed laboratory models. A technique was developed to test a standard pump of commercial design, and at a speed at which the maximum pump efficiency is reached.

The present study was started in the fall of 1931 in the Hydraulics Laboratory of this Institute. Instruments for measuring the average velocity vectors in the volute of a centrifugal pump were developed in 1932 by the present author working with the help and advice of Prof. R. T. Knapp.

An M.S. thesis[17] in May, 1933 described this technique. At that time (1933) publication of that work was held up in order to extend the technique, to improve instruments, and to wait to do the work in the new precision laboratory then being constructed at the Institute under a co-operative agreement with the Metropolitan Water District.

OBJECT OF INVESTIGATION

The previous experimental investigations have had some limitations and restrictions which this investigation sought to overcome. This present study extends this particular field of experimental knowledge and technique in the following significant respects:

1. A technique of measurement was developed to test a standard high efficiency volute pump of commercial design, thus avoiding the difficulties of special laboratory models.
2. Besides developing an improved technique for measuring average velocity vectors at a point in the volute, a technique was developed for measuring the "instantaneous" velocity vectors at a point in the volute, that is, the velocity for a time corresponding to five degrees rotation of the impeller.
3. Very extensive measurements were made at closely spaced stations completely around the volute, using both the "average" and the "instantaneous" velocity techniques. Complete sets of data were taken for different operating conditions, varying from near shutoff, through normal to high rates of discharge. With this experimental information it has been possible to analyze (a) the complete velocity distribution of the flow between the impeller and the volute, and (b) the variation in the velocity distribution in the discharge from any one impeller passage.

GENERAL METHOD

Briefly stated, the method employed was to insert a special Pitot tube across the volute. Referring to Fig. 1, a sampling slide valve was inserted in each of the two connections from the Pitot tube to the special differential gage.

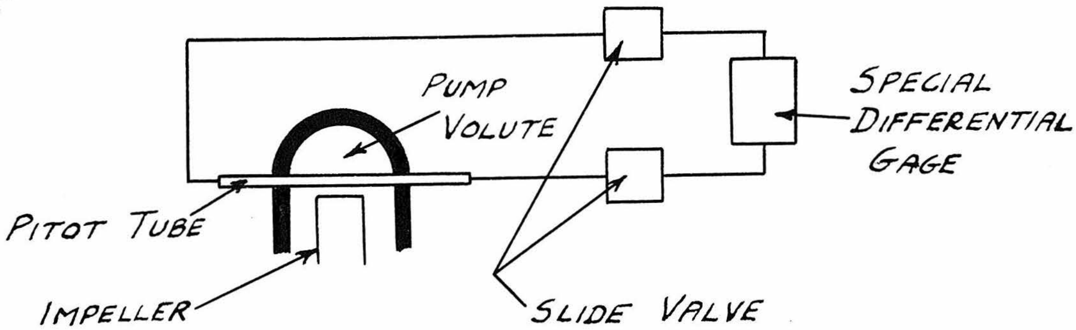


FIG. 1

These slide valves opened for a short interval of time each revolution of the pump, which resulted in a series of pressure impulses to the gage. Means were provided for shifting the phase between the pump shaft and the valve opening. A stroboscope indicated the position of the opening. Thus the velocity could be measured as any particular point of the impeller passed the Pitot tube.

In these tests, all the facilities of the Metropolitan Water District pump laboratory at the California Institute of Technology were employed. The automatic speed control was used, and accurate measurements of pump head, quantity, torque, etc., were made simultaneously with the internal velocity measurements.

This report is divided into two parts. Part One describes the technique of measurement, and treats separately each major instrument. Part Two gives the information obtained from volute surveys of two different pumps.

PART ONE - TECHNIQUE OF MEASUREMENT

THE PITOT TUBE MEASUREMENT

In any flow measurement it is not difficult to determine the true total head or dynamic pressure. The total head is obtained by placing an opening normal to the direction of flow. Since the total pressure equals the static pressure plus the velocity pressure, an accurate measurement of static pressure is necessary for an accurate determination of velocity. The main feature of the Pitot tube used in this study is that it gives an accurate measurement of static pressure in turbulent flow.

This accurate measurement can be seen by considering the pressure distribution around a small cylinder across the stream. Referring to Fig. 2, it is known that there is a critical angle with the direction of flow at which the velocity pressure has no effect. This means that, having an opening at the critical angle with the flow direction, the

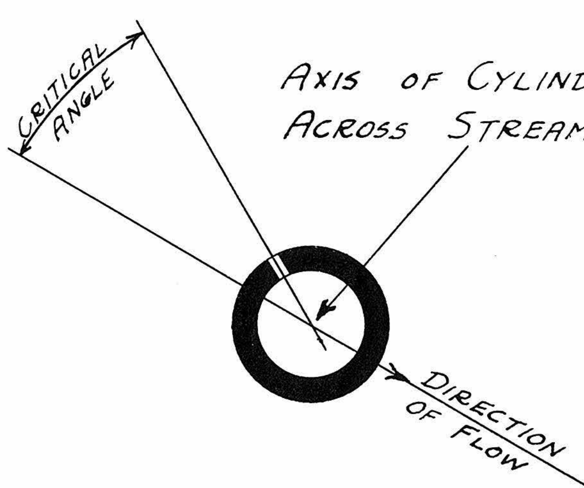


FIG. 2

pressure transmitted to a gage will be truly static and unaffected by any influence of velocity.

Various investigators have found the value of this critical angle. The rough measurements of Dryden[18] indicated that the angle was slightly less than forty degrees. Fechheimer[19] made very careful measurements with 3/16" and 1/4" diameter cylinders in a stream of air and found the critical angle to be 39¼ degrees. In 1926 Dryden, Moss, and other authorities agreed that Fechheimer's determination gave a means for the most accurate measurement of static pressure in turbulent flow.

The author has checked this critical angle and the construction of 1/4" and 3/16" diameter Pitot tubes by observing the position of the hole in a stream of known direction at a point where the static pressure was known. This check gave an angle of 39¼ degrees for the velocity range met in these pump tests. It is interesting to note that these latter measurements were made in water, but that the Reynolds' number was substantially the same as that used by Fechheimer.

A static pressure comparison was made on a pump provided with several pressure connections in the volute wall. The pressure at each wall point was measured with the laboratory balance pressure gages. Pitot tube holes were also drilled around the volute and static pressures measured with a Pitot tube built on the basis of this 39¼ degrees idea. The two independent static pressure measurements agreed

---

18. "Air Forces on Circular Cylinders", H.F. Dryden, Scientific Paper No. 394, Bureau of Standards, 1920.

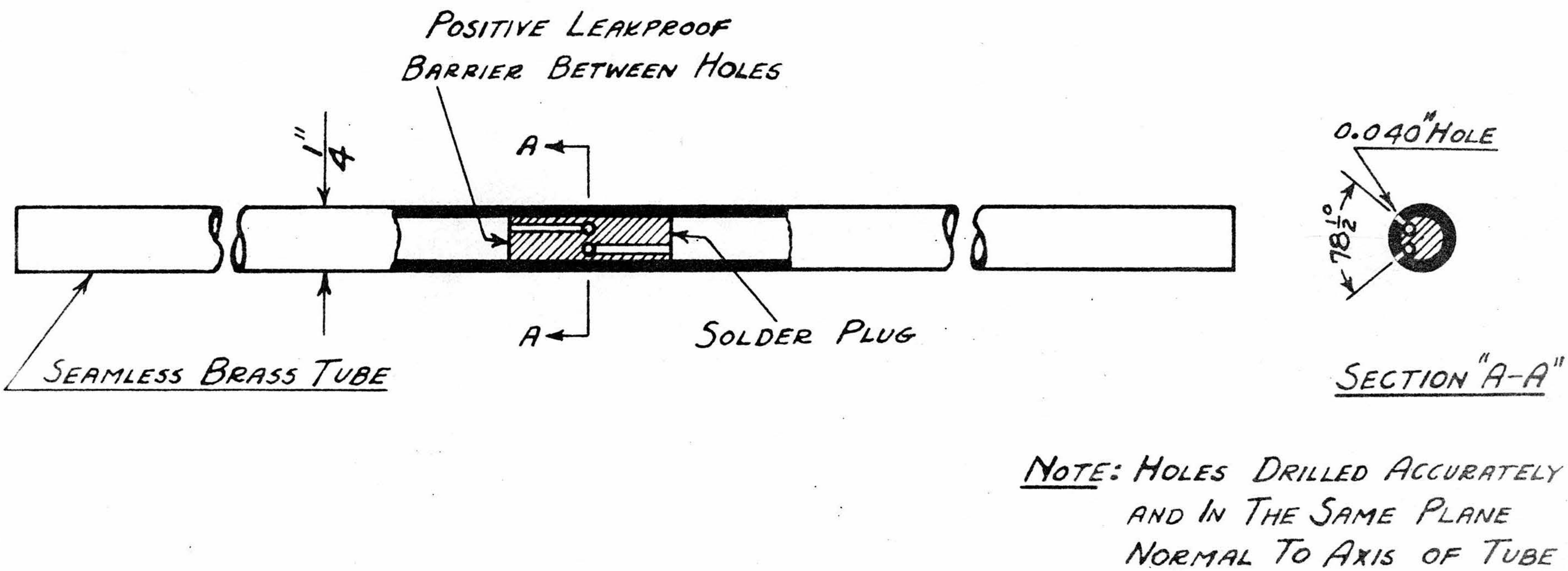
19. A.S.M.E. Transactions, 1926, Vol. 48.

closely, this checking the Pitot tube. An exact comparison was not possible because the Pitot tube holes were slightly displaced in both radial and axial directions from the static pressure connections in the volute.

Fig. 3 (Page 9) shows the construction of the Pitot tube. The small pressure openings were possible because of the use of a special differential gage (described on Page 16). In using the tube in the pump volute it was necessary to "balance" the tube. Two static holes were used, each hole being connected to one side of a differential gage. The Pitot tube in the unknown stream was rotated about its own axis until the pressures at each hole were the same, in other words, the differential pressure was zero. At this position velocity pressure had no effect on either hole, and either hole could be used to measure static pressure. The bisector of the angle between the holes gave the direction of flow. The dynamic pressure was then obtained by placing an opening normal to the direction of flow, i.e., by simply rotating one hole back into the stream 39¼ degrees. Thus, with the values of the directly measured total and static heads, the difference gave the velocity head, and the measured angle gave the direction of the velocity vector.

In order to insure that the pressure openings faced upstream the Pitot tube was first rotated 360 degrees to find the approximate position for maximum positive dynamic pressure. Then the direction was more accurately determined by "balancing" the pressures.

A Pitot diameter of 1/4" was the smallest that seemed



SPECIAL PITOT TUBE

FIG. 3

safe. It probably would have been desirable to use a smaller diameter but, at the time this work was initiated brass tubing was the only material available. It was felt that the high velocity pressures encountered might have caused an appreciable deflection of a smaller tube.

Fig. 4 (Page 11) shows the apparatus used for checking Pitot tubes. The swing spout and volume measuring tank made possible a measurement of the total rate of flow, while the long length of pipe before the jet straightened out the flow to insure accuracy of direction. The jet diameter was measured with a special micrometer jig. With the Pitot tube in the free jet various tests were made on the magnitude of velocity measurement. These are tabulated in Table 1. The small discrepancies are due to the jet setup itself.

Table 1.

Checks on Magnitude of Velocity as Indicated by Pitot Tube.

Actual Velocity Feet per Second (in Free Jet)	Velocity Indicated by Pitot Tube
26.6	26.6
37.0	37.0
41.6	41.5
45.1	45.1
48.4	48.4
53.0	52.9
55.0	54.9
57.2	57.2

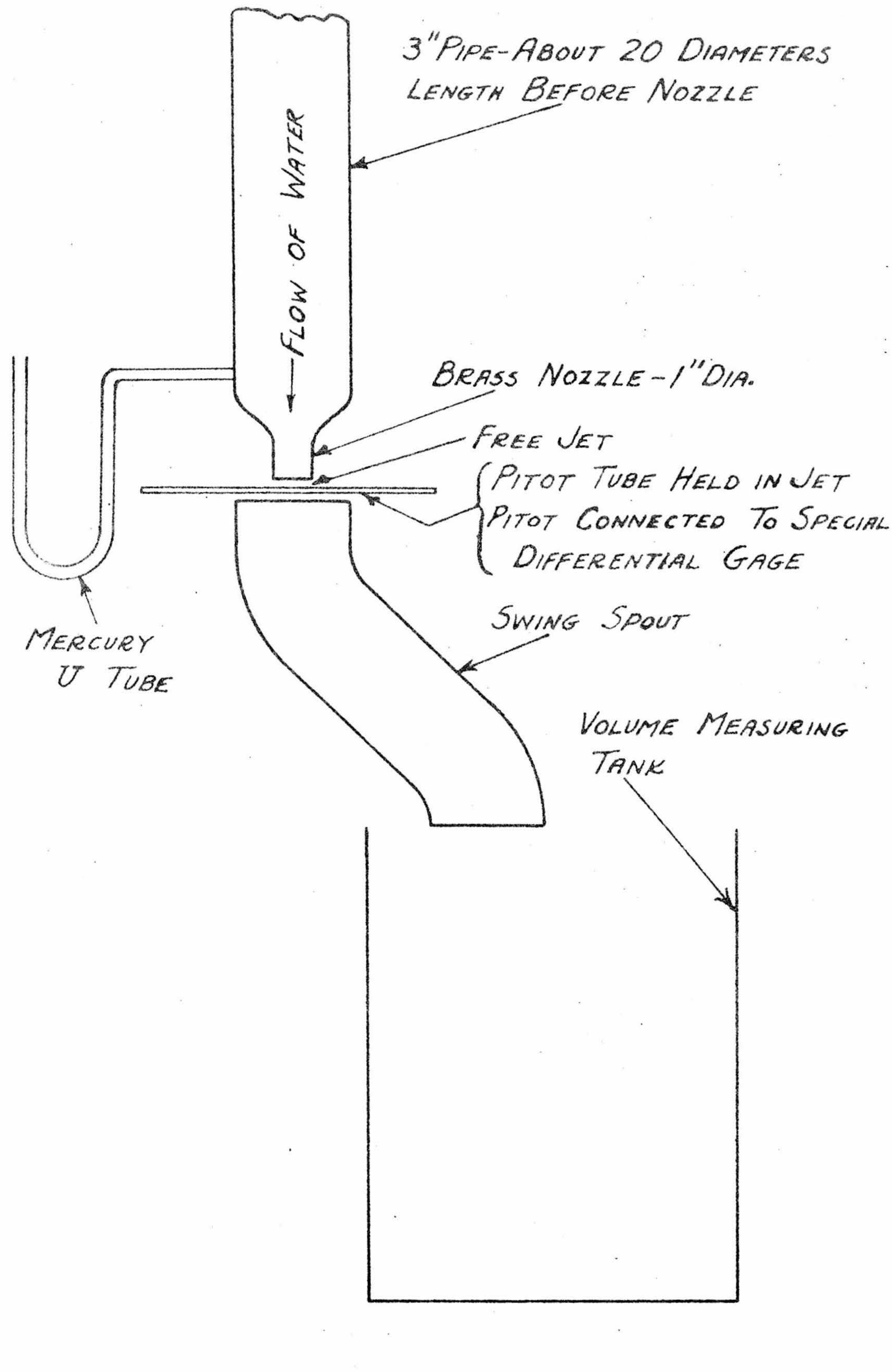


FIG. 4

DIAGRAM OF APPARATUS FOR CHECKING PITOT TUBES

PITOT TUBE WALL CORRECTION

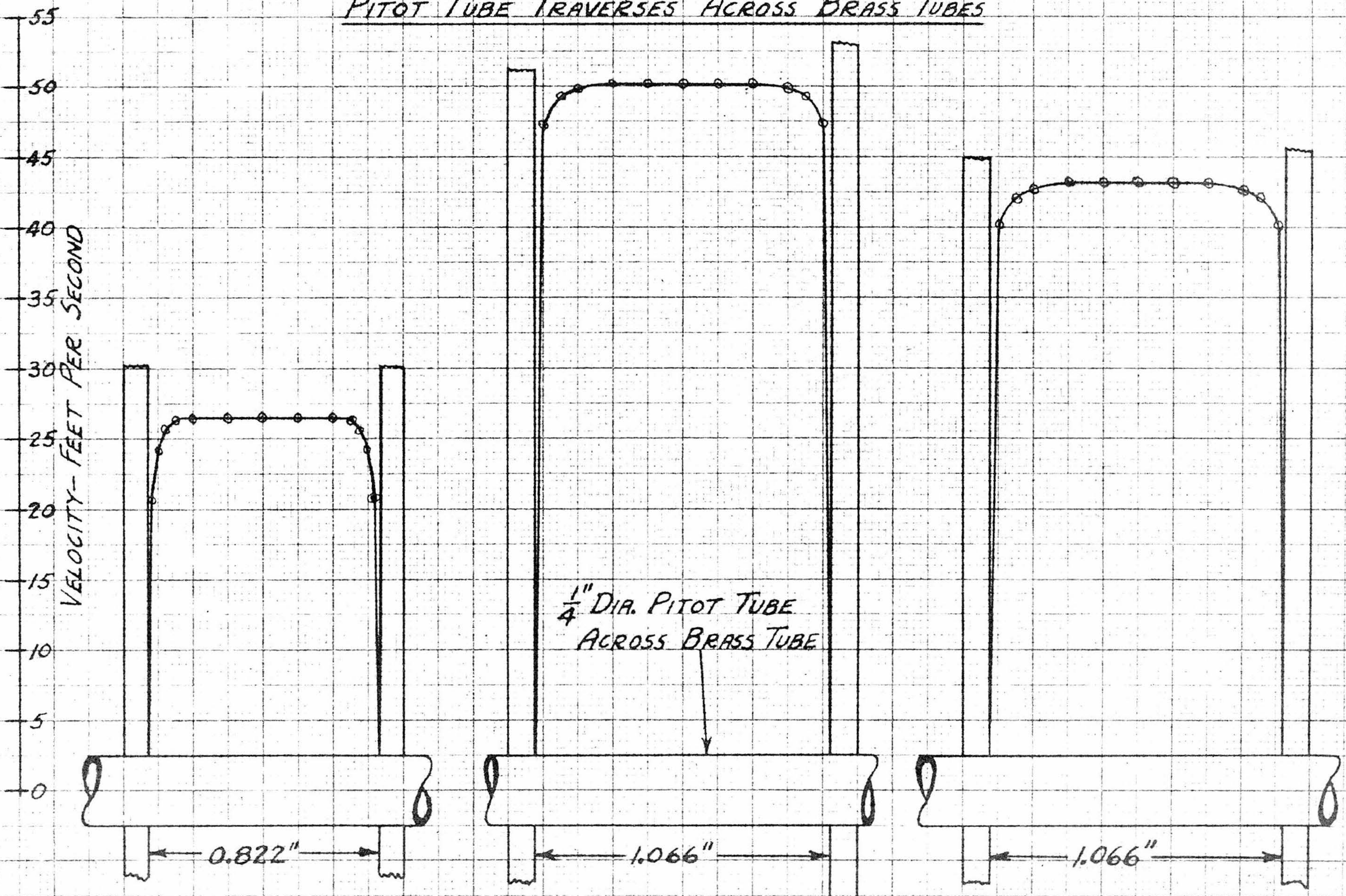
For pump traverses a correction has to be applied to compensate for the close proximity of the wall. Using the apparatus shown in Fig. 4 (Page 11), 3/16" and 1/4" diameter Pitot tubes were inserted across different sizes of brass tubes. Velocity traverses were made using the same technique as on pump traverses. Fig. 5 (Page 13) shows some of these profiles. For each case of a Pitot tube in a pipe the mean velocity from a traverse was compared with the mean velocity from simultaneous volume measurements. This plot is shown in Fig. 6 (Page 14), and covers more than the range met in pump traverses.

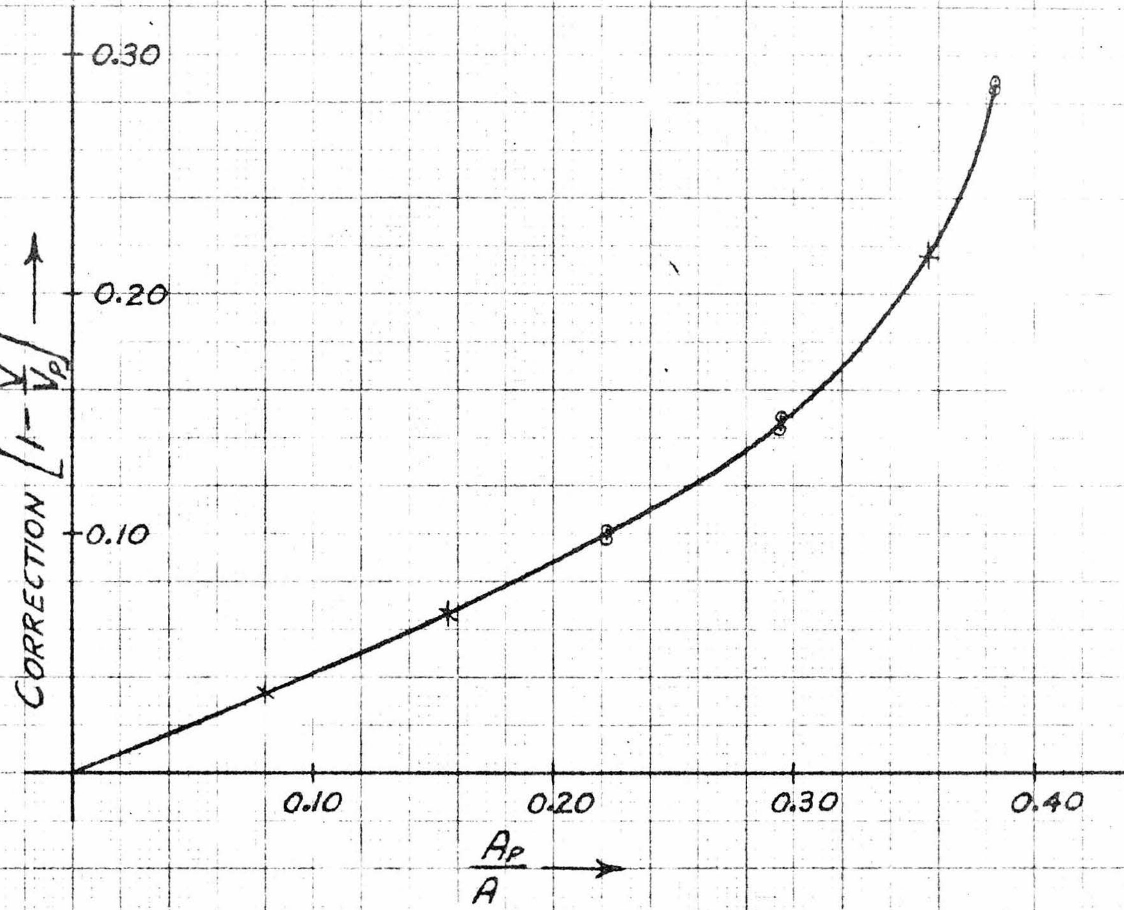
STUFFING BOXES IN VOLUTE

Fig. 7 (Page 15) shows how the Pitot was inserted in the different pump volutes. Long stuffing boxes were used to give a rigid support to the Pitot tube.

FIG. 5

PITOT TUBE TRAVERSES ACROSS BRASS TUBES



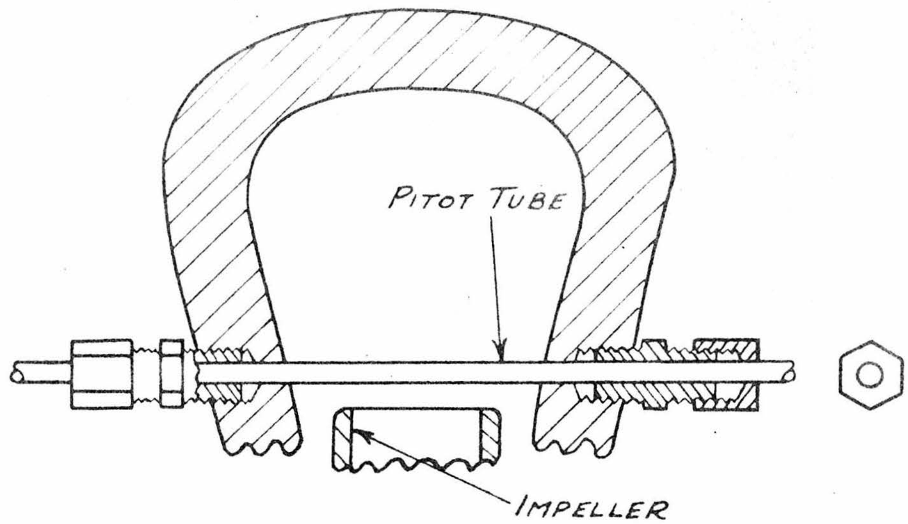


A_p = PROJECTED AREA OF PITOT TUBE
A = AREA OF PIPE WITHOUT PITOT TUBE
V_p = VELOCITY INDICATED BY PITOT TUBE
MEAN OF TRAVERSE

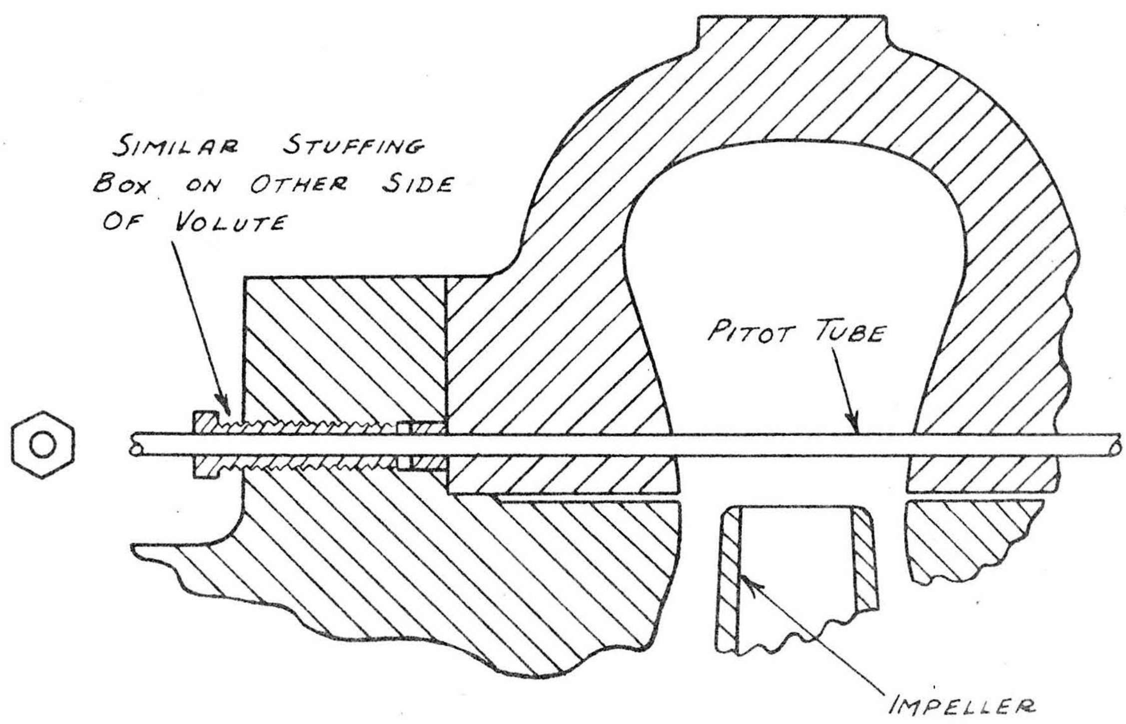
V = VELOCITY = $\frac{\text{QUANTITY PER UNIT TIME}}{A}$

PITOT TUBE WALL CORRECTION

FIG. 6



RADIAL SECTION THRU VOLUTE
OF BYRON JACKSON DOUBLE SUCTION PUMP



RADIAL SECTION THRU VOLUTE
OF SINGLE SUCTION WORTHINGTON PUMP

FIG. 7

METHODS OF INSERTING PITOT TUBE IN DIFFERENT PUMP VOLUTES

SPECIAL DIFFERENTIAL GAGE

An ordinary mercury or water U tube manometer would be out of the question on these fluctuating pressure measurements. To obtain a reading in a system using an ordinary U tube an appreciable flow is required through the Pitot pressure openings and the connecting leads, hence a reading could not be obtained in a reasonable time. This difficulty was overcome by the development of a special differential gage. Its main features are that only a very small amount of flow is required for operation, the gage is very responsive, sensitive, and accurate. Many experimenters using Pitot tubes have been limited by the use of U tube manometers. Such a manometer requires that the Pitot pressure openings be large enough to avoid excessive damping, while this special differential gage permits the use of much smaller pressure openings.

Fig. 8 (Page 17) shows the internal construction. Since the helix element is the same as is used on pressure recording instruments this differential gage can be adapted to any desired accuracy and range of pressure by a suitable selection of helix. One end of the helix element is fixed, while the other end is free to move. The free end is so connected as to cause a rotation of the stellite mirror when the free end moves. Water pressure is applied to both the inside and the outside of the Bourdon element, the whole mechanism being in water in a closed case. Thus, when the differential pressure changes, the free end of the Bourdon element rotates the mirror. The mirror arrangement magnifies this movement with the aid of an optical

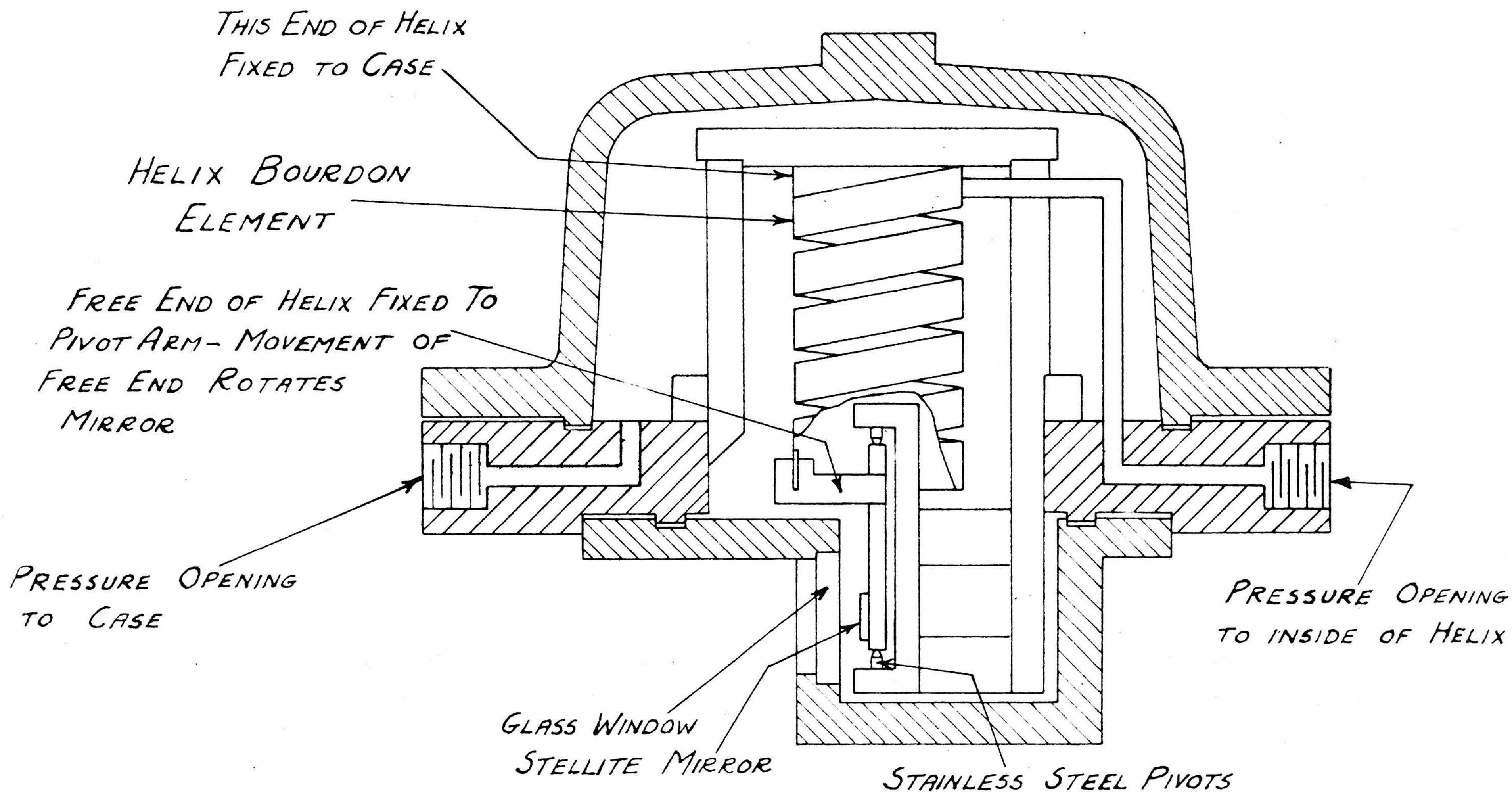


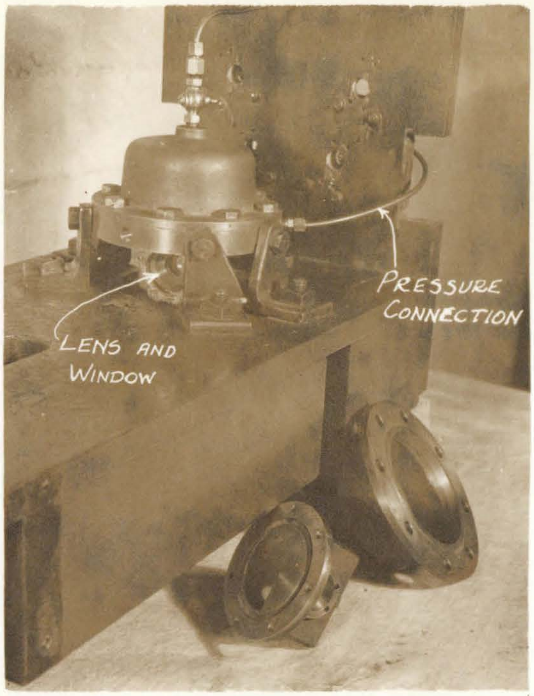
FIG. 8 SPECIAL DIFFERENTIAL GAGE

system. A light source sends a beam of light through the glass window to strike the mirror. The reflected ray was focused on a graduated scale. Some photographs are shown in Fig. 9 and 10 (Pages 19 and 20).

One interesting problem was that of the chromatic dispersion of the light image. Between the glass window and the mirror is a variable prism of water. Various mono-chromatic lamps were tried, but proved unsatisfactory. The most satisfactory solution was obtained by simply inserting a light filter directly in front of the lamp.

The gage was calibrated with a dead weight gage tester. Fig. 11 (Page 21) shows a typical straight line calibration curve. Repeated tests over long periods of time have shown that this gage holds its calibration precisely. One factor that might account for this is that of the selection of the helix. The working pressure range of the helix element is about one-third the rated pressure range. By rated pressure range is meant the range of the helix element as used in standard recording instruments. Tests have shown that the scale deflection depends solely on the differential pressure, and is independent of the absolute pressure.

One interesting feature of this gage is that it requires no time to give a pressure reading. If the cocks (one to close off one connection from the Pitot tube, and the other to open the vent) are handled quickly the light image comes to the pressure position quicker than the eye can follow it. In this gage there is no appreciable flow of water, it is practically a constant volume system. It is obvious that many other applications can be found for this type of pressure instrument.



SPECIAL DIFFERENTIAL GAGE

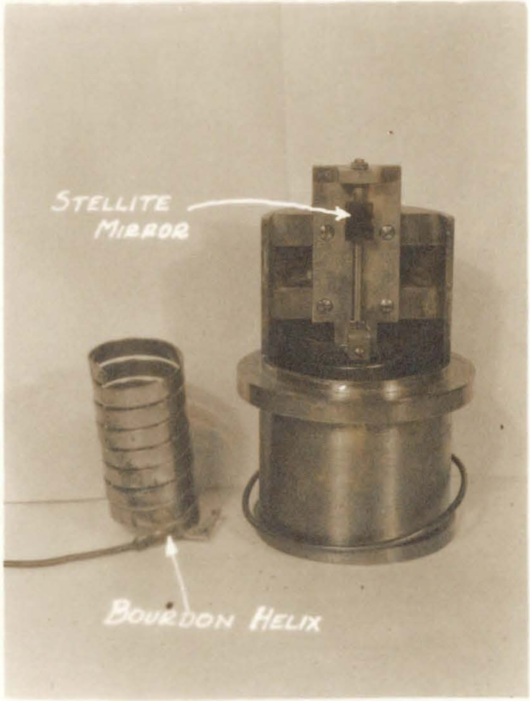
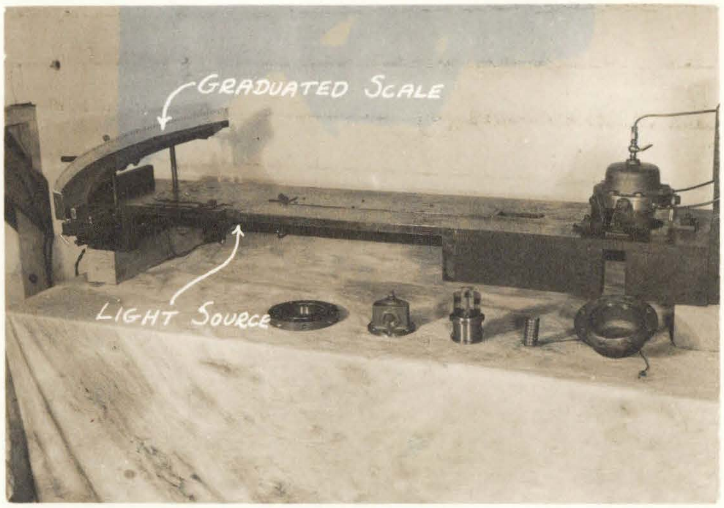
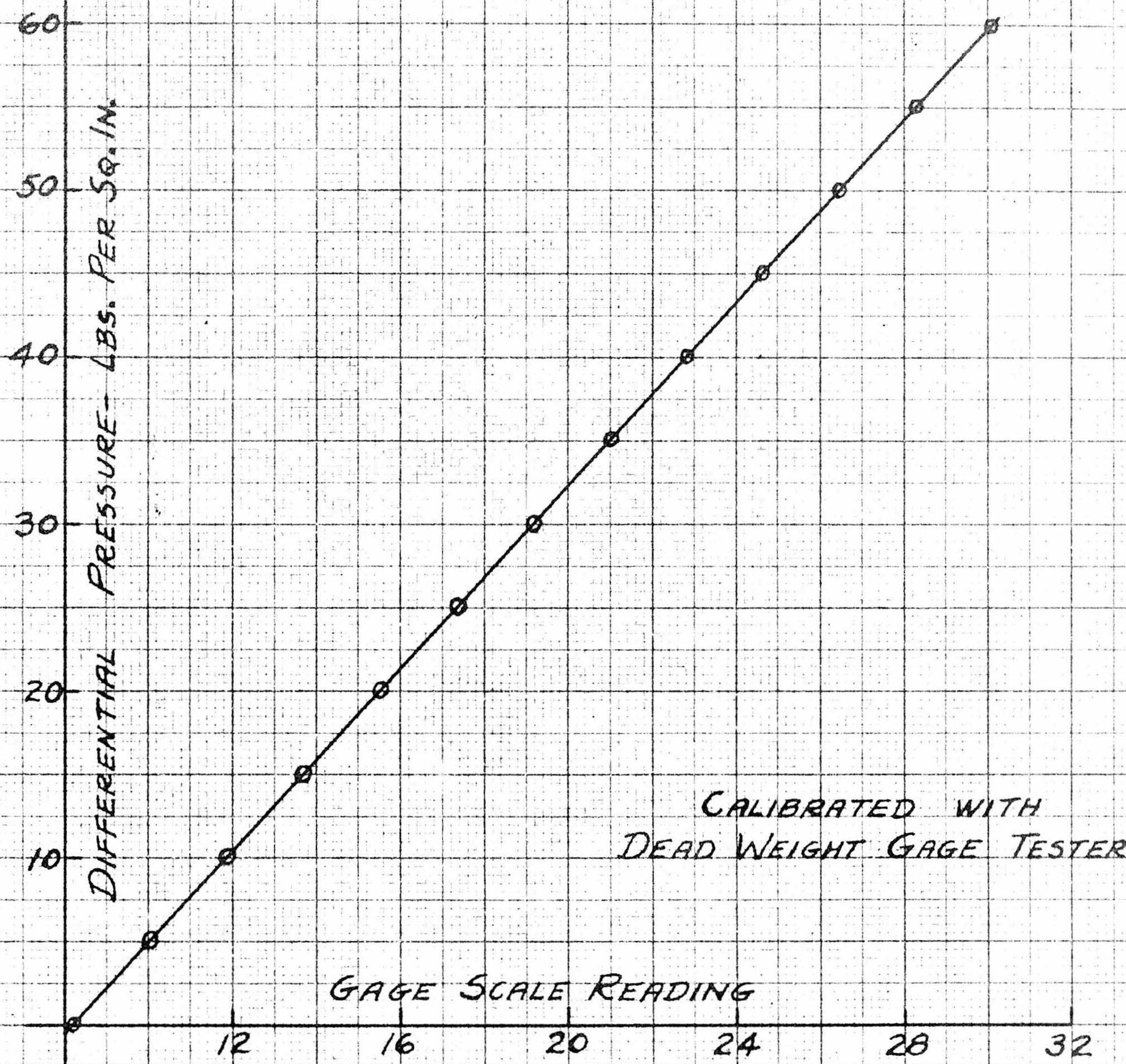


FIG. 9



SPECIAL DIFFERENTIAL GAGE

FIG. 10



TYPICAL CALIBRATION CURVE
OF SPECIAL HELIX DIFFERENTIAL GAGE

FIG. 11

SAMPLING VALVE AND PHASE SHIFTER

With the Pitot tube and the special differential gage, measurements of average velocity can be made in the pump volute. Because of the minute flow required to operate the differential gage it was possible to use a slide valve to sample the pressure transmitted from the Pitot as any particular point on the impeller passed the Pitot tube. A description followed by some photographs will show how this was accomplished.

Fig. 12 (Page 23) shows the general arrangement. The two pole generator on the dynamometer shaft drove the four pole synchronous motor at one-half pump speed. An eccentric on the motor shaft worked in a yoke to impart simple harmonic motion to the push rod driving the two valves. Each valve opened twice (back and forth) for every revolution of the synchronous motor, which meant one valve opening per revolution of the pump.

When the valves opened the commutator contact would fire the neon light at the protractor on the pump shaft (one firing per one revolution of pump). The bolts between the field and the end bells of the motor were removed, and means provided for rotating the field of the motor. Thus there was a positive mechanical-electrical connection between the pump shaft and the slide valve, and by simply rotating the field of the motor it was possible to change the phase relation between the pump shaft and the time of opening of the valve. It was possible to change the phase by 360 degrees, while the stroboscope always gave a precise indication of the position of opening of the valve.

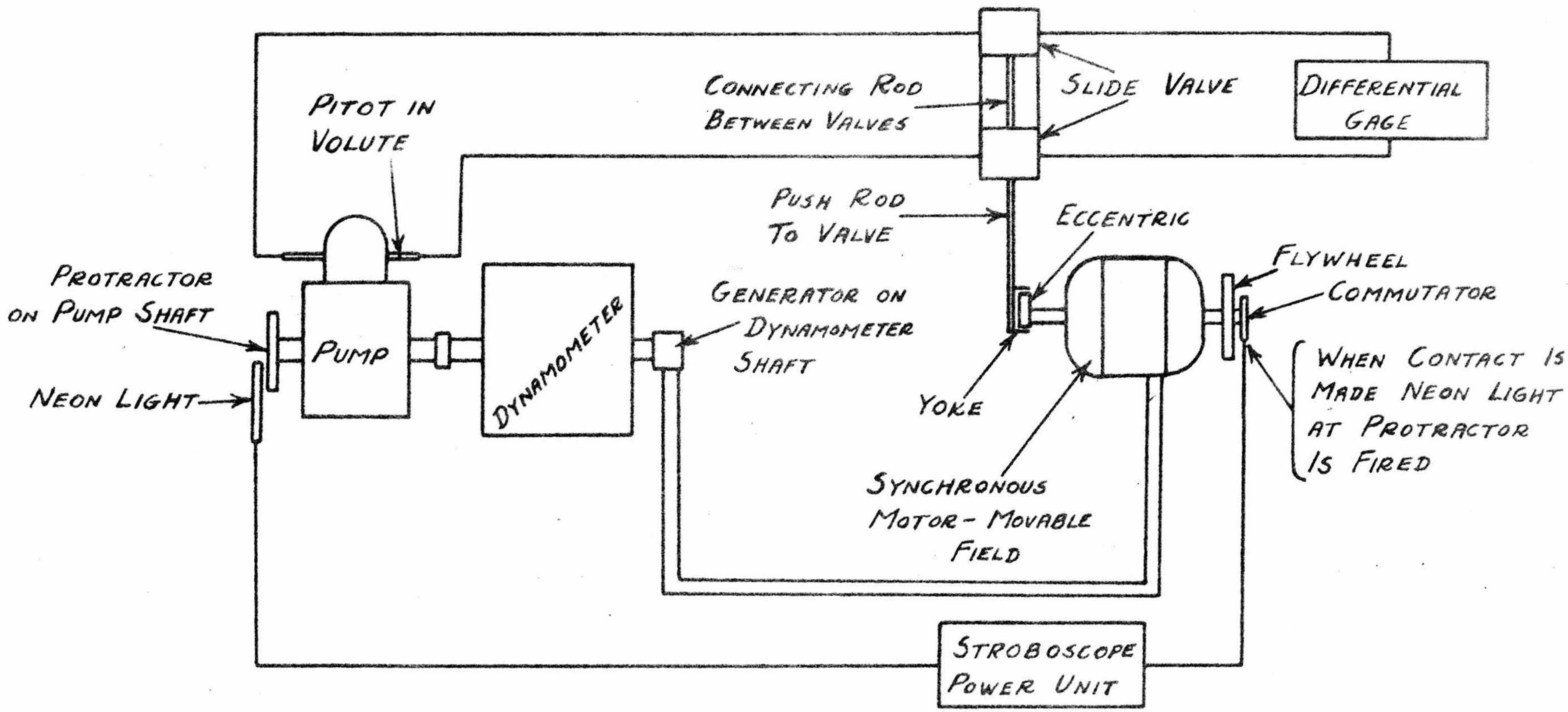


FIG. 12 ARRANGEMENT OF APPARATUS

Fig. 13 (Page 25) shows a self-explanatory sketch of one of the valves. Mr. E. R. Lockhart[20] helped work out the details of the sampling valves. Both valves are duplicates, and accurately positioned to open at exactly the same time.

Use was made of the fact that the eccentric and yoke imparted simple harmonic motion to the valve. Recalling the curve, Fig. 14, at the middle of the valve travel the velocity is a maximum while the acceleration is zero. The valve was set to open at the middle of the travel. With a slot thickness of 0.005" the time of opening corresponded to an angular rotation of five degrees of the impeller. For the present a shorter time

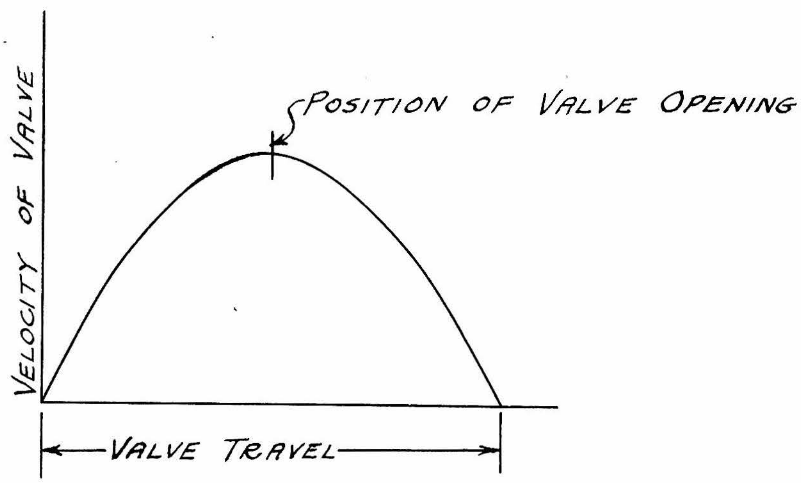


FIG. 14

of opening was not of great interest. The inertia forces of the reciprocating parts were reduced as much as possible. Operating the motor at one-half pump speed helped do this.

Each slide valve was placed in a line from a dead weight gage tester to the special differential gage. As various known pressures were applied the corresponding differential gage

20. M. S. Thesis E.R. Lockhart, 1934.

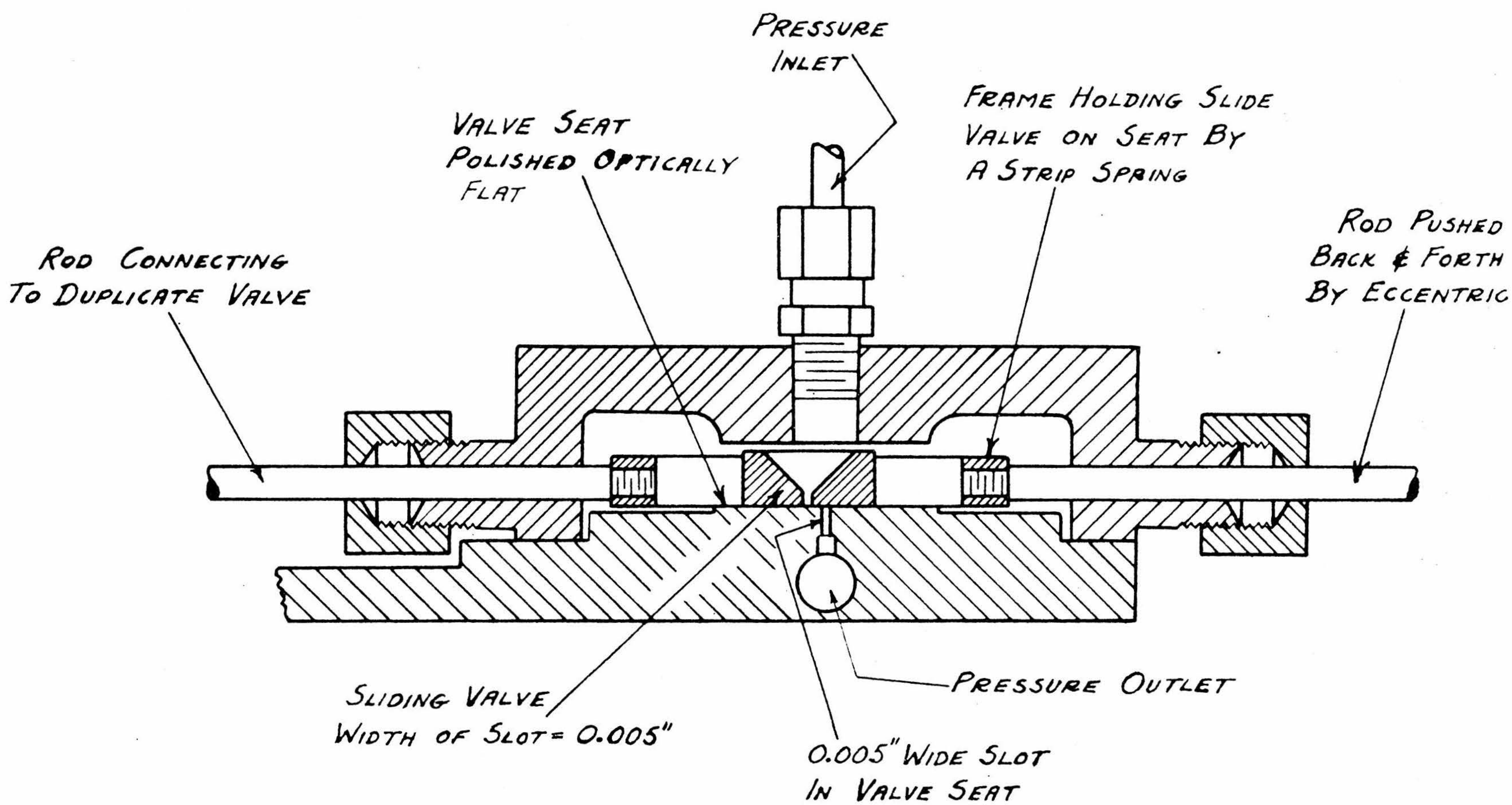


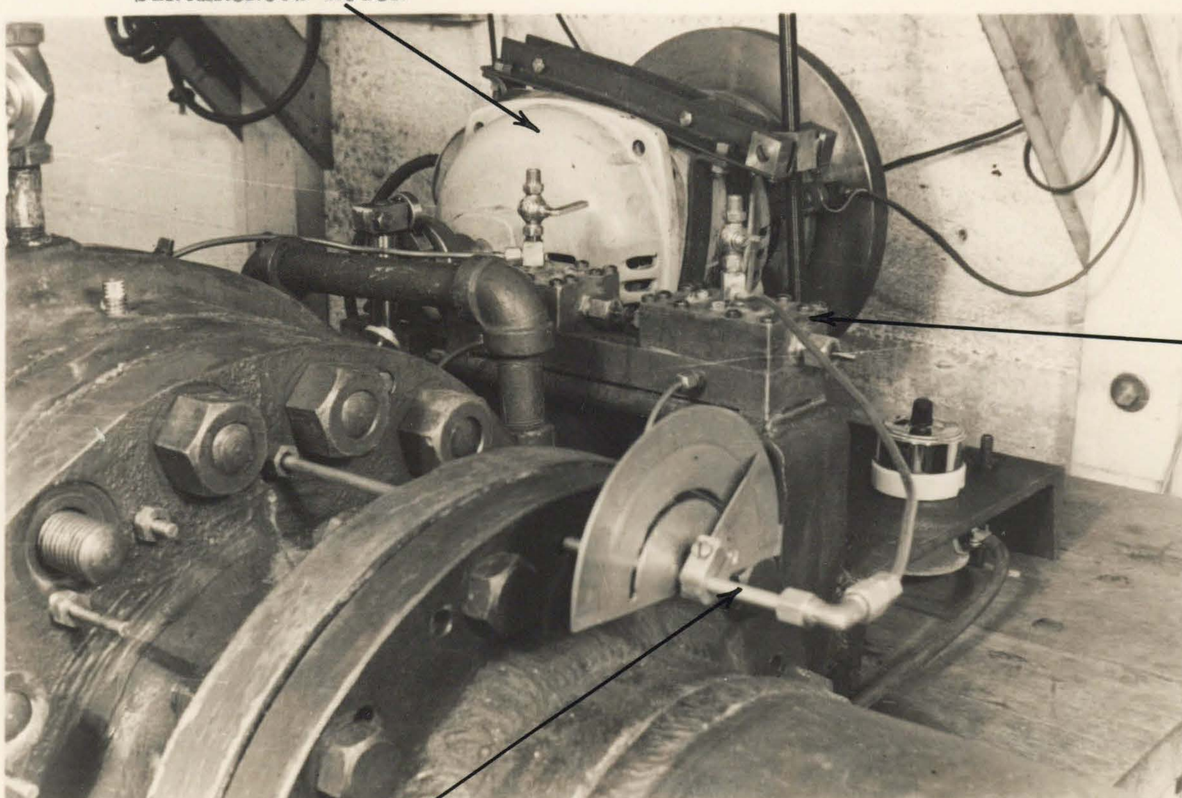
FIG. 13 CONSTRUCTION OF SLIDE VALVE

readings were noted. It was found that the motion of the valve at any speed had no effect on the pressure gage reading. This is probably due to the fact that the valve opens at the point of zero acceleration. Whether or not this is the complete explanation, it is an experimental fact that the slide valve has no effect on the pressure transmitted.

Another interesting check was made. With the apparatus installed on a pump, in place of the Pitot tube an oscillating pressure of known frequency was applied to the slide valve. The pump was run at various speeds each different from the known frequency of the applied pressure. For each case the number of "beats" per minute, as shown by the differential gage, corresponded exactly to the difference between the pump R.P.M. and the cycles per minute of the applied pressure.

Tests have shown that each slide valve when closed does not leak. If a gradually decreasing or increasing pressure is applied to a moving valve the differential gage reading changes in jerky steps - for part of the time the valve is closed and the pressure reading is held constant.

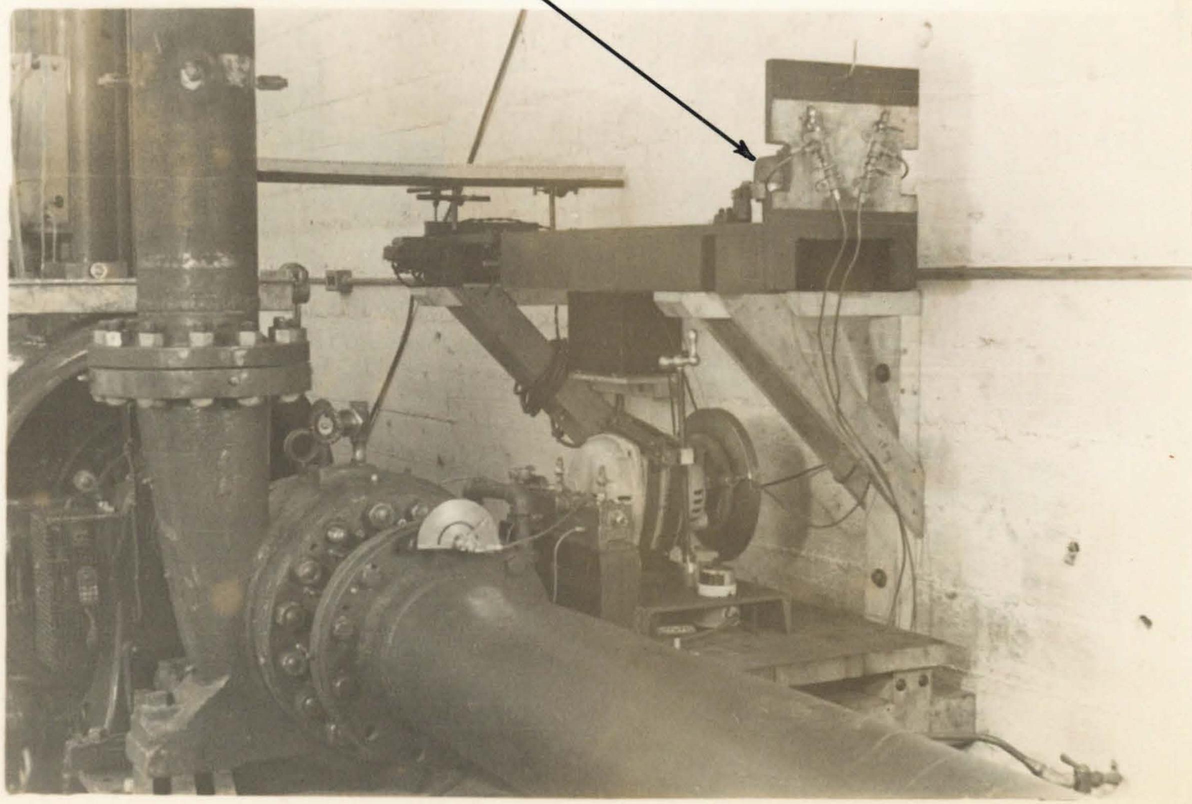
Some photographs of installations on two different pumps are shown in Fig. 15, 16, 17 and 18 (pages 27,28,29 and 30).



APPARATUS ON WORTHINGTON SINGLE SUCTION PUMP

FIG. 15

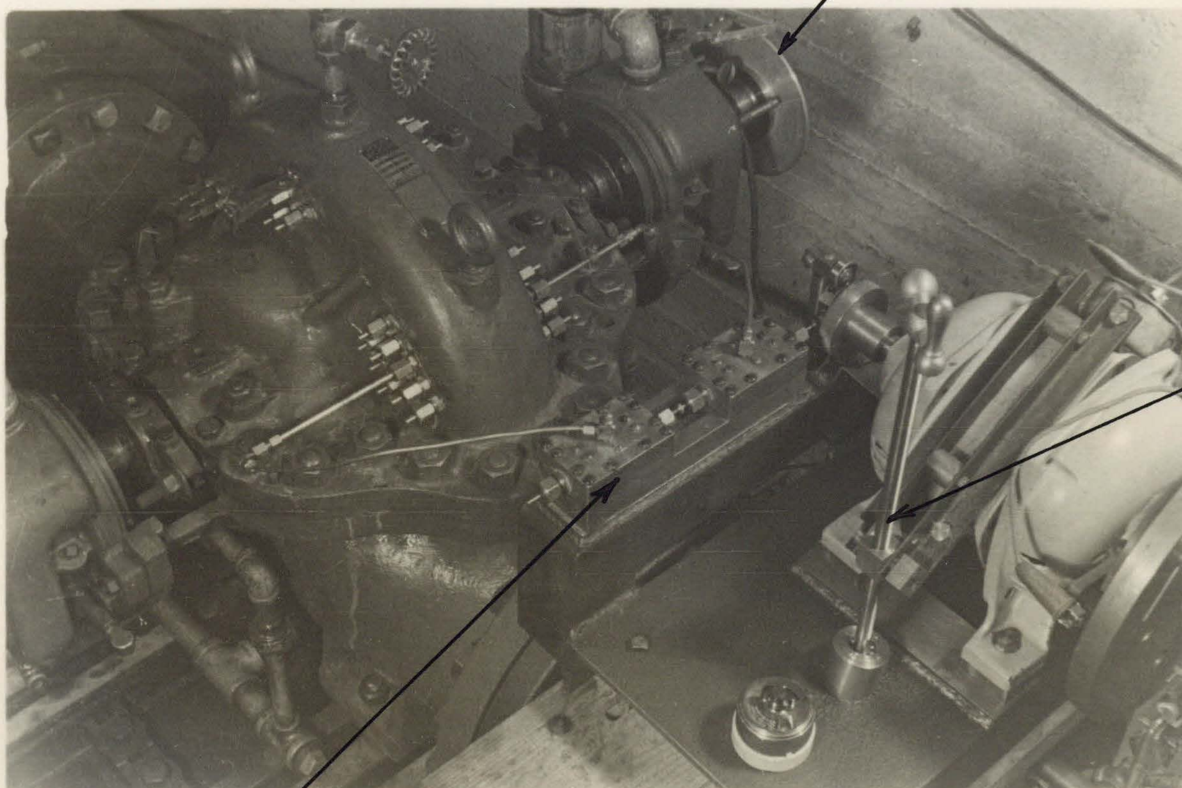
DIFFERENTIAL GAGE



BLACK BOX UNDER
DIFFERENTIAL GAGE
IS STROBOSCOPE
POWER SUPPLY

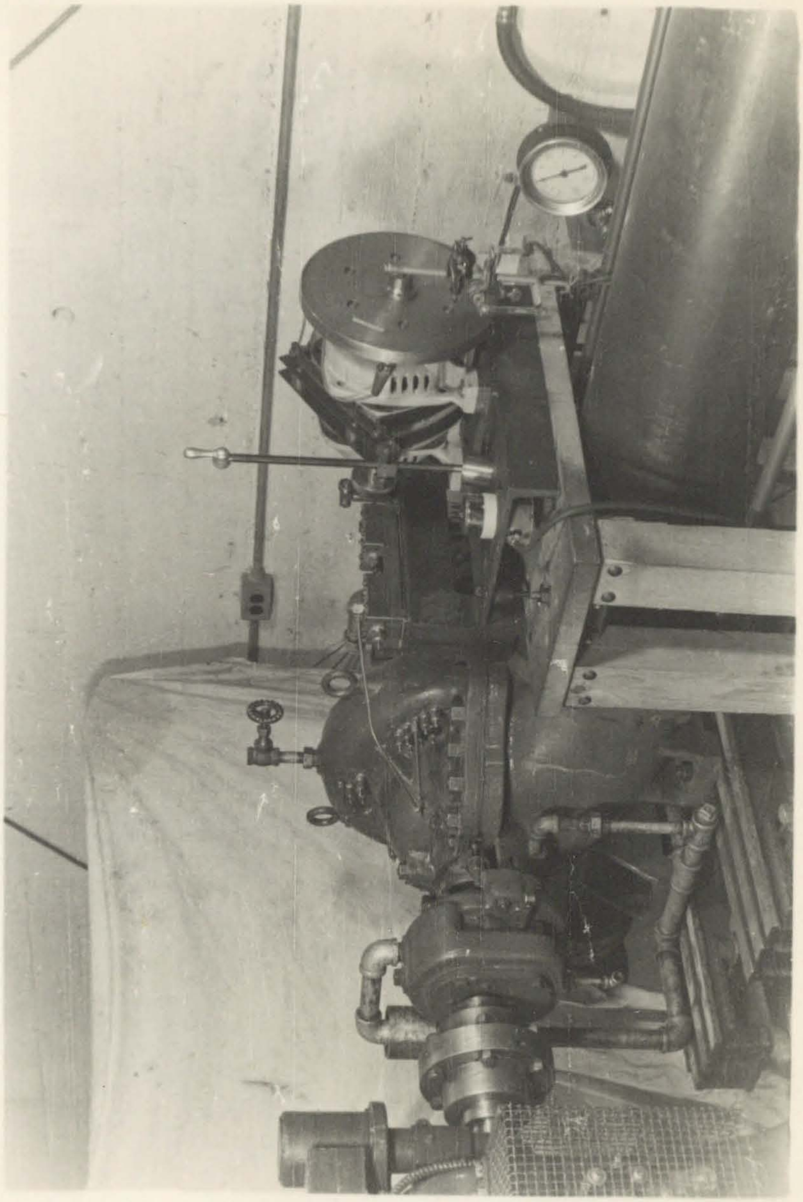
APPARATUS ON WORTHINGTON SINGLE SUCTION PUMP

FIG. 16



APPARATUS ON BYRON-JACKSON DOUBLE SUCTION PUMP

FIG. 17



APPARATUS ON BYRON-JACKSON DOUBLE SUCTION PUMP

FIG. 18

GENERAL REMARKS ON TECHNIQUE OF MEASUREMENT

This technique of instantaneous velocity measurement is possible because the system from Pitot tube to gage has no volume changes, the pressure is transmitted by the water without any flow. Pressure waves in the connecting leads might cause trouble, but the length of the leads was reduced to a minimum by placing the slide valve as close as possible to the pump.

As for accuracy of measurements, the magnitude of velocity was determined to within one-half per cent, while the angle was read to within one-fourth degree. Many readings were duplicated closer than this.

To illustrate the sensitivity of this apparatus a case will be cited. With the throttle and suction head at one setting the pump was operated without using the automatic speed control. The speed could not vary very much, no noticeable "wandering" of the venturi reading or the head (developed by pump) reading could be observed. However, the internal velocity instruments showed an appreciable "wandering" or "scatter" of readings, the apparatus being sensitive enough to detect the slight variations of the velocity vectors inside the pump. Pump traverses were made with caution and continuous rechecking to avoid any trouble of this kind.

It is to be noted that all the instruments were carefully designed, and what is reported here is the result of considerable development work. The experimental checks have been mentioned. Test results have shown other checks. All these facts demonstrate the reliability of this new technique of measurement. The following section will present the results of measurements on two high efficiency pumps, one a single suction and the other a double suction type.

PART TWO- TEST RESULTS

NOTATION USED IN MEASUREMENTS

For designating valve opening, one vane tip edge was chosen as a zero reference. Fig. 19 shows for two examples the position of the impeller when the valve opens. If the valve opened as the zero reference mark passed the Pitot tube the "phase angle" was "zero degrees". If the valve opened as some other point on the impeller passed the Pitot tube this point was referred to the zero mark as so many "degrees phase angle", where this angle is measured in the opposite sense to that of the pump rotation, i.e., the point "lags" the zero reference.

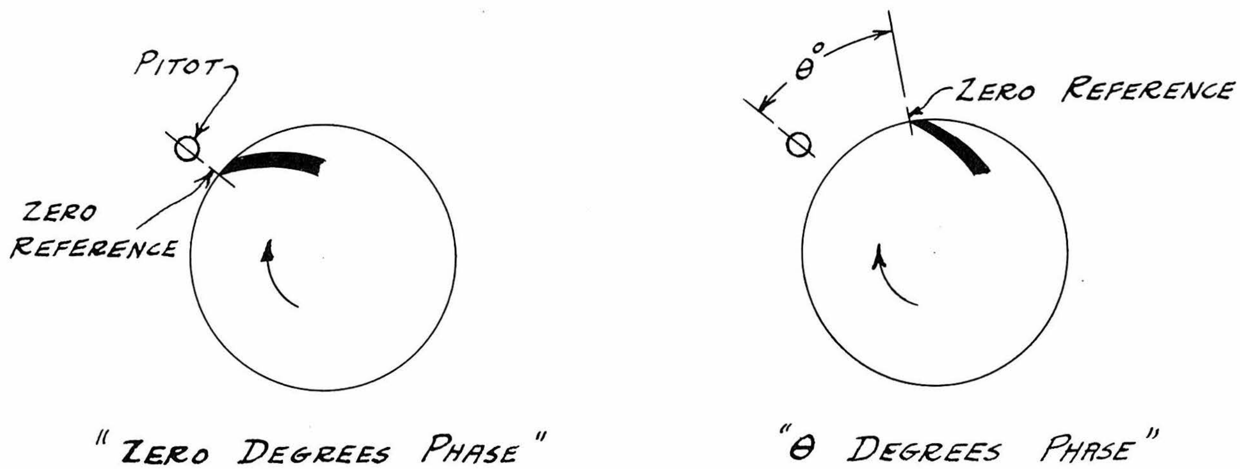


FIG. 19

For designating the axial position of measurement across the volute "center" means over the center of the impeller, while "right" or "left" refers to the side from this center. On the double suction pump "right" and "left" were used with the observer facing the pump suction flange. On the single suction pump the "right" side refers to the suction side of the pump.

INSTANTANEOUS VELOCITY MEASUREMENTS ON BYRON JACKSON 8" DOUBLE SUCTION CENTRIFUGAL PUMP

PUMP RATING AND DIMENSIONS

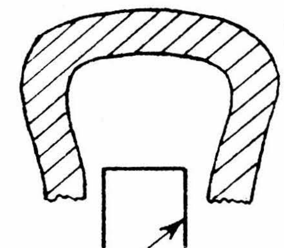
Serial No. 128407

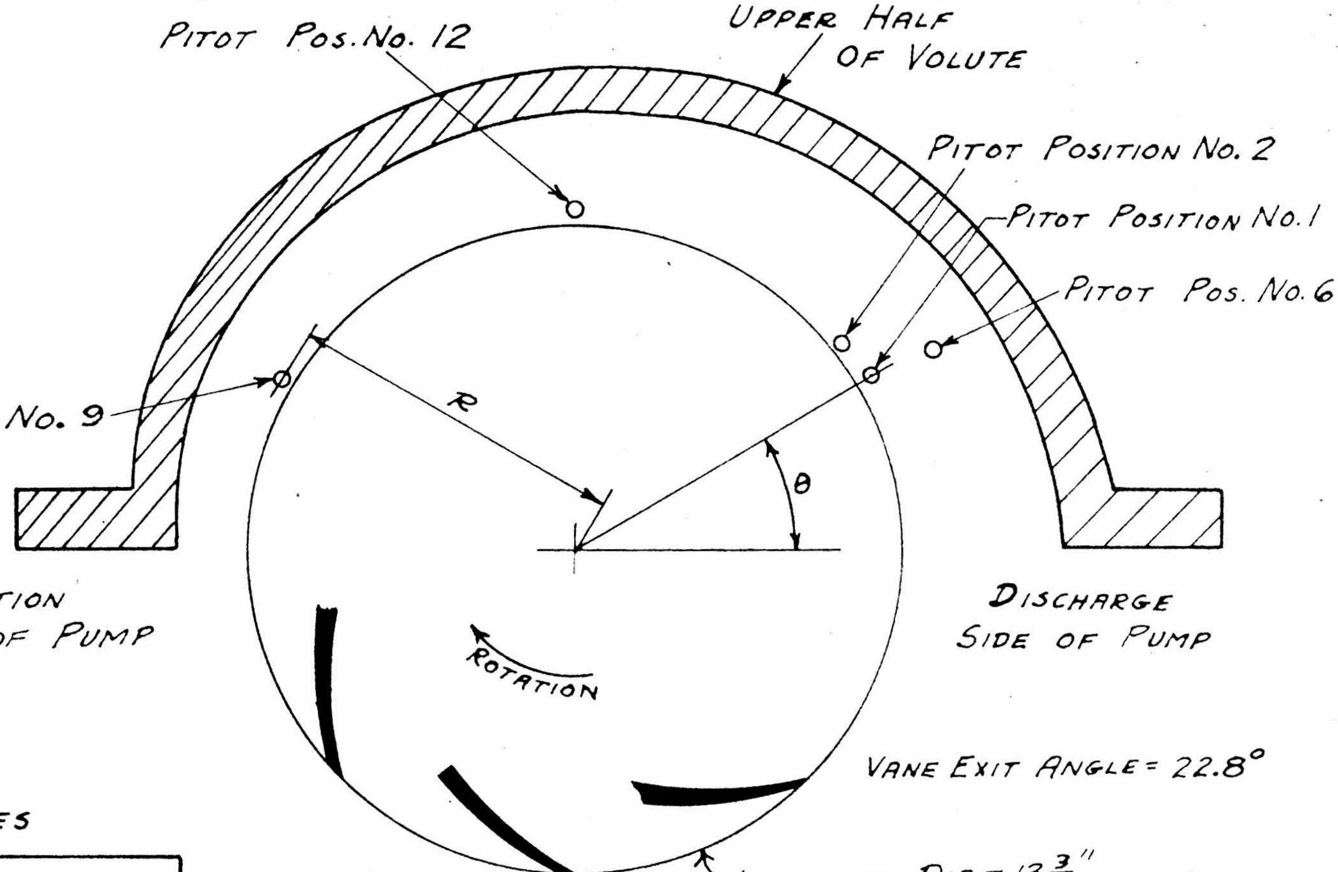
Size	8"
Type	Double Suction
Capacity	2400 G.P.M.
Total Head	360 Feet
Speed	2500 R.P.M.
Specific Speed	1400

Impeller Outside Diameter	13 3/8"
Impeller Inside Width	1 12/32", 1 13/32"
Impeller Outside Width	1 3/4"
Number of Vanes	8

All pump tests were made at 2000 R.P.M. The maximum efficiency(84.6 per cent) at 2000 R.P.M. was the same as at the rated speed. All tests were made at plus 40 feet inlet head.

Fig. 20 (Page 35) shows the spacing of the Pitot tube stations in the volute.





SPACING OF PITOT HOLES

POSITION	R RADIAL DISTANCE	θ DEGREES COUNTER CLOCKWISE	DISTANCE PITOT CENTER TO IMPELLER
1	$7\frac{1}{16}$"	30.3	3/8"
2	$6\frac{15}{16}$"	37.8	1/4"
6	$8\frac{7}{16}$"	29.3	1 3/4"
9	$7\frac{1}{16}$"	149.7	3/8"
12	$7\frac{1}{16}$"	90.0	3/8"

BYRON JACKSON
DOUBLE SUCTION PUMP
SCALE $\frac{1}{4}$ SIZE

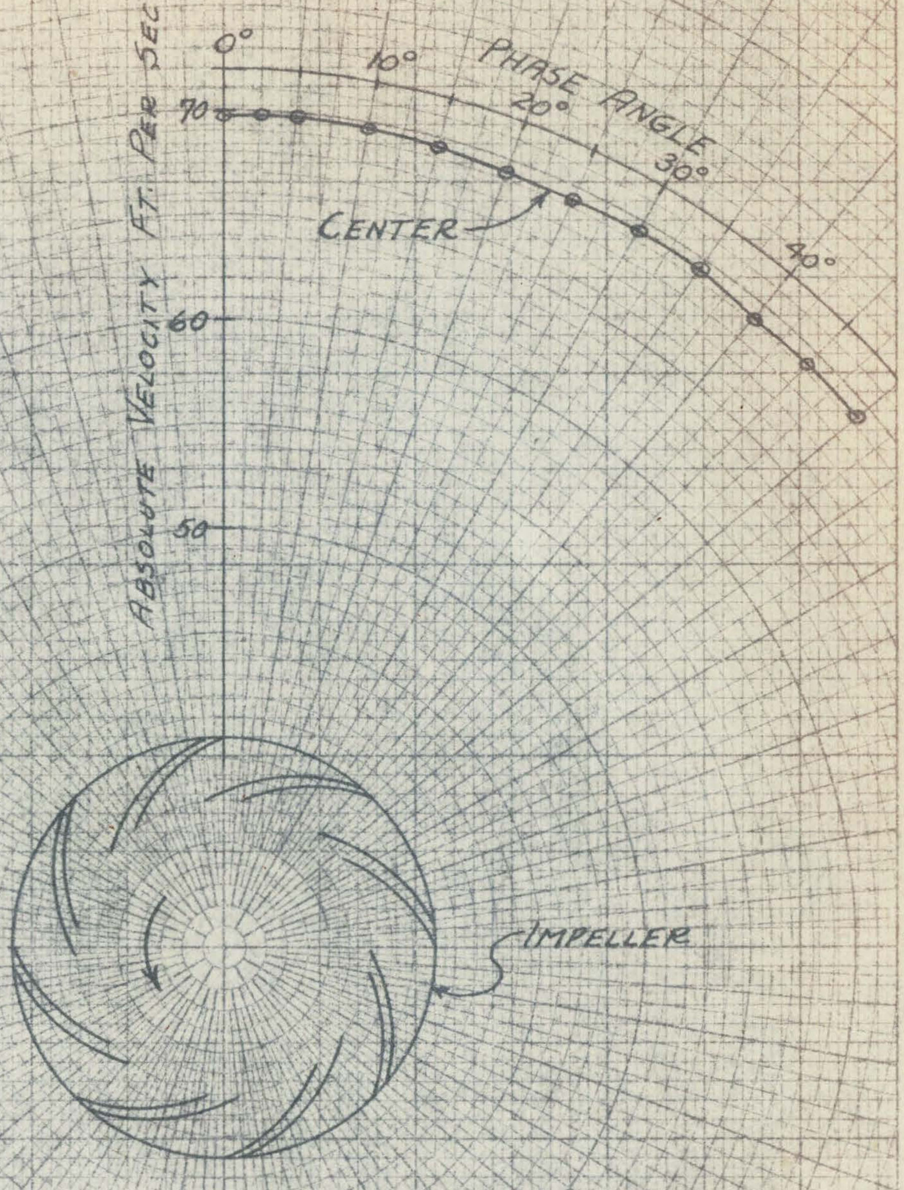
FIG. 20

INSTANTANEOUS VELOCITY DISTRIBUTION BETWEEN VANES

A very interesting question is that of the instantaneous velocity distribution between vanes. To test this, measurements were made as the phase angle was varied and the Pitot tube kept at a fixed position across the volute. Fig. 21,22,23, and 24 (pages 37,38,39, and 40) show that there was practically no velocity variation at normal pump discharges. The slight waviness in these curves might be due to variations in impeller dimensions and a slight impeller motion in the axial direction. This lack of a velocity variation was also found in a single suction pump test, and will be discussed later.

The Pitot tube was a short distance from the impeller, and it is possible that a large velocity variation does exist in a very short radial region close to the impeller periphery. The Pitot was placed as close to the impeller as seemed advisable. In one case there was 1/8" (one-half Pitot diameter) clearance between the impeller and the Pitot tube.

Suggestions have been made to consider a shorter time of valve opening. If a large velocity variation did exist in a shorter time it would be of minor importance, at least for the present, as compared to some of the other unanswered problems of internal pump flow.

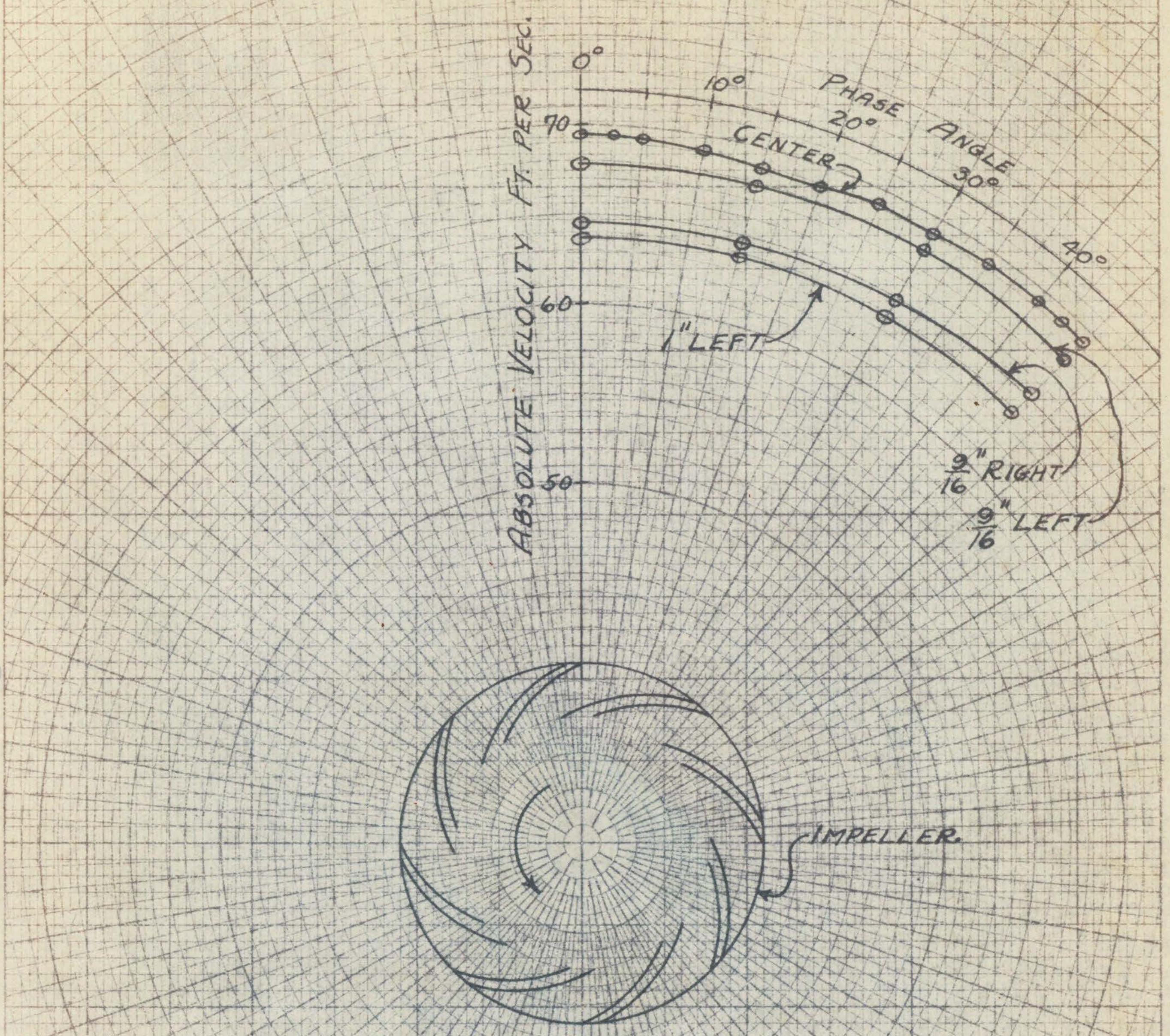


INSTANTANEOUS VELOCITY DISTRIBUTION
BETWEEN VANES

POSITION NO. 12 { 3/8" FROM IMPELLER TO PITOT CENTER

BYRON JACKSON DOUBLE SUCTION PUMP
2000 R.P.M.
NORMAL Q 4.65 CU. FT. PER SEC.

FIG. 21



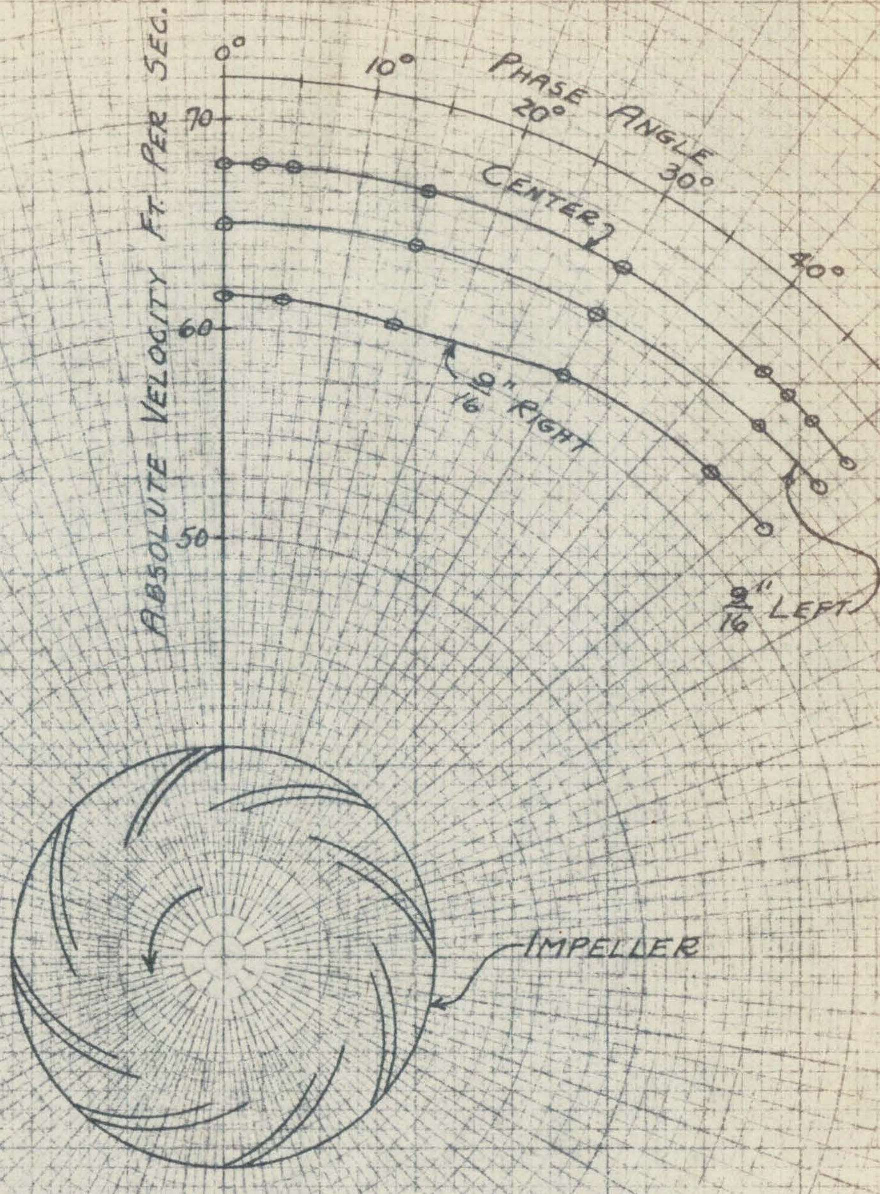
INSTANTANEOUS VELOCITY DISTRIBUTION
BETWEEN VANES

POSITION NO. 2 { 1/4" FROM IMPELLER TO PITOT CENTER

BYRON JACKSON DOUBLE SUCTION PUMP
2000 R.P.M.

NORMAL Q 4.65 CU. FT. PER SEC.

FIG. 22

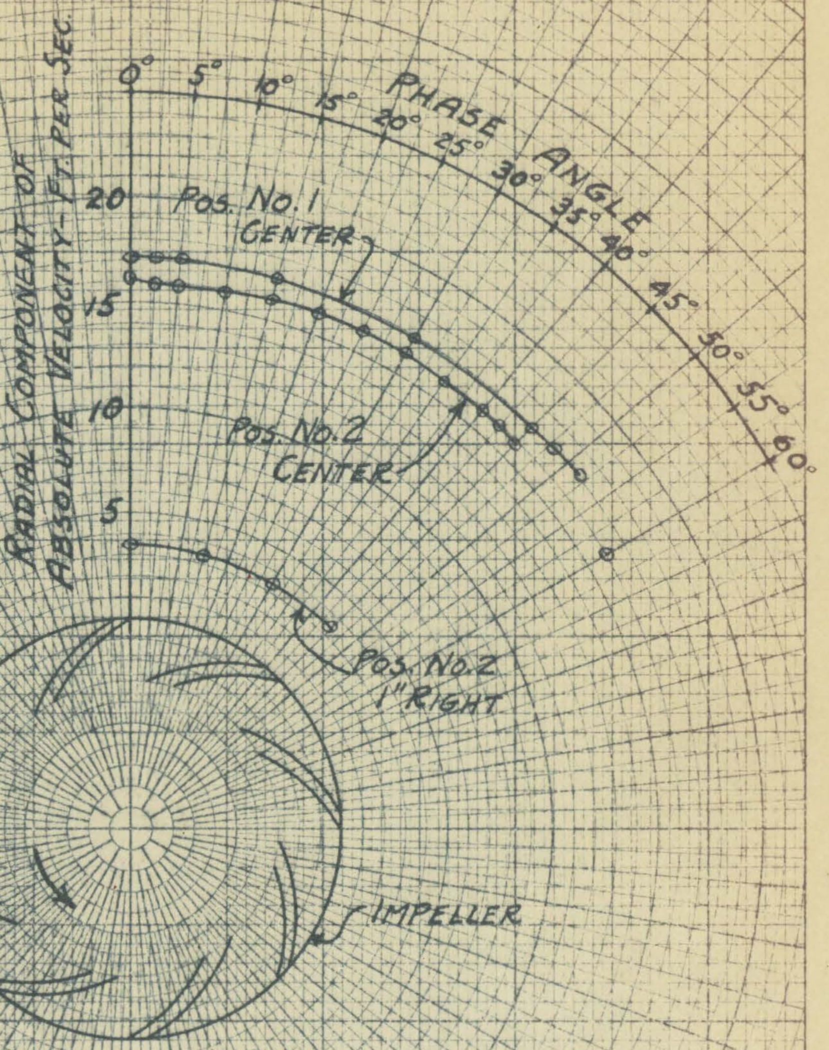


INSTANTANEOUS VELOCITY DISTRIBUTION
BETWEEN VANES

POSITION NO. 1 { 3/8" FROM IMPELLER TO PITOT CENTER

BYRON JACKSON DOUBLE SUCTION PUMP
2000 R.P.M.
NORMAL Q 4.65 CU. FT. PER SEC.

FIG. 23



INSTANTANEOUS VELOCITY MEASUREMENTS

B-J DOUBLE SUCTION PUMP

2000 R.P.M. NORMAL Q = 4.65 CU. FT. PER SEC.

POS. NO. 1 $\frac{3}{8}$" FROM PITOT CENTER TO IMPELLER

POS. NO. 2 $\frac{1}{4}$" FROM PITOT CENTER TO IMPELLER

FIG. 24

VELOCITY TRAVERSES ACROSS VOLUTE

EFFECT OF SHROUDS ON VELOCITY DISTRIBUTION

Fig. 25 and Fig. 26(pages 42) and 43) show the profiles obtained from Pitot tube traverses across the volute, each traverse being made at a constant phase angle.

One noticeable feature is that in most cases there is a positive radial velocity outside of the shroud. Pump volute cases were first built as indicated in Fig. 27, with a small clearance between the runner and case. Later, pump efficiencies were improved by allowing ample clearance between the runner shrouds and the case wall, as shown in Fig. 28. The Pitot tube

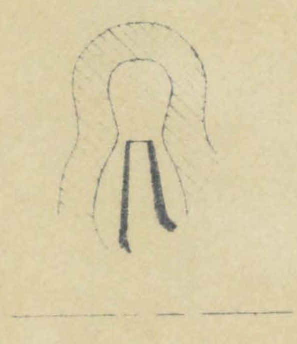


FIG. 28

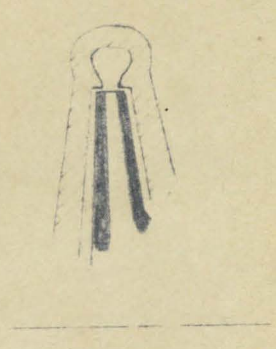


FIG. 27

measurements offer an explanation of the improved efficiencies. A clearance allows use to be made of some of the energy which otherwise would be dissipated if trapped between the shroud and casing.

Fig. 25

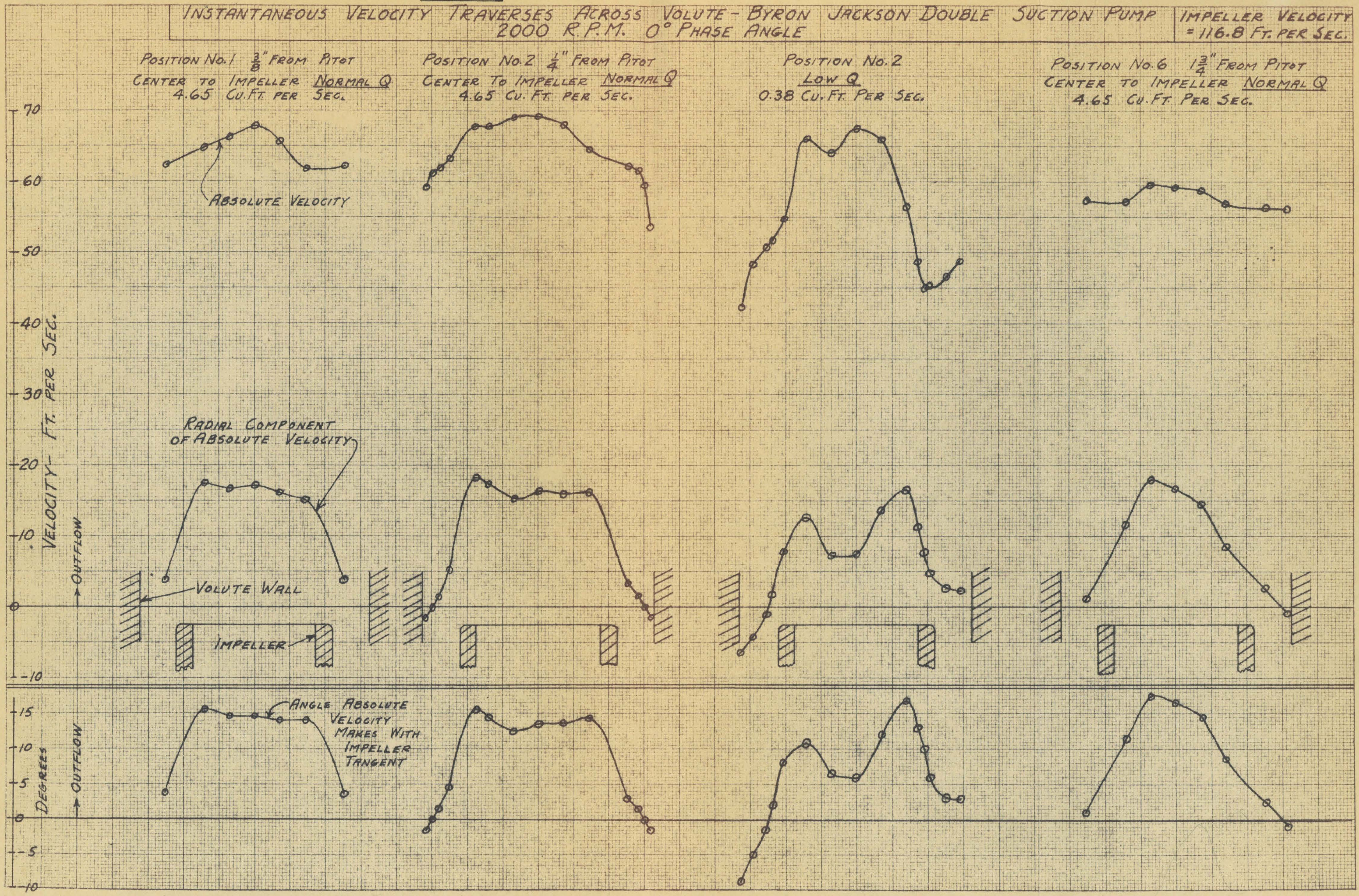
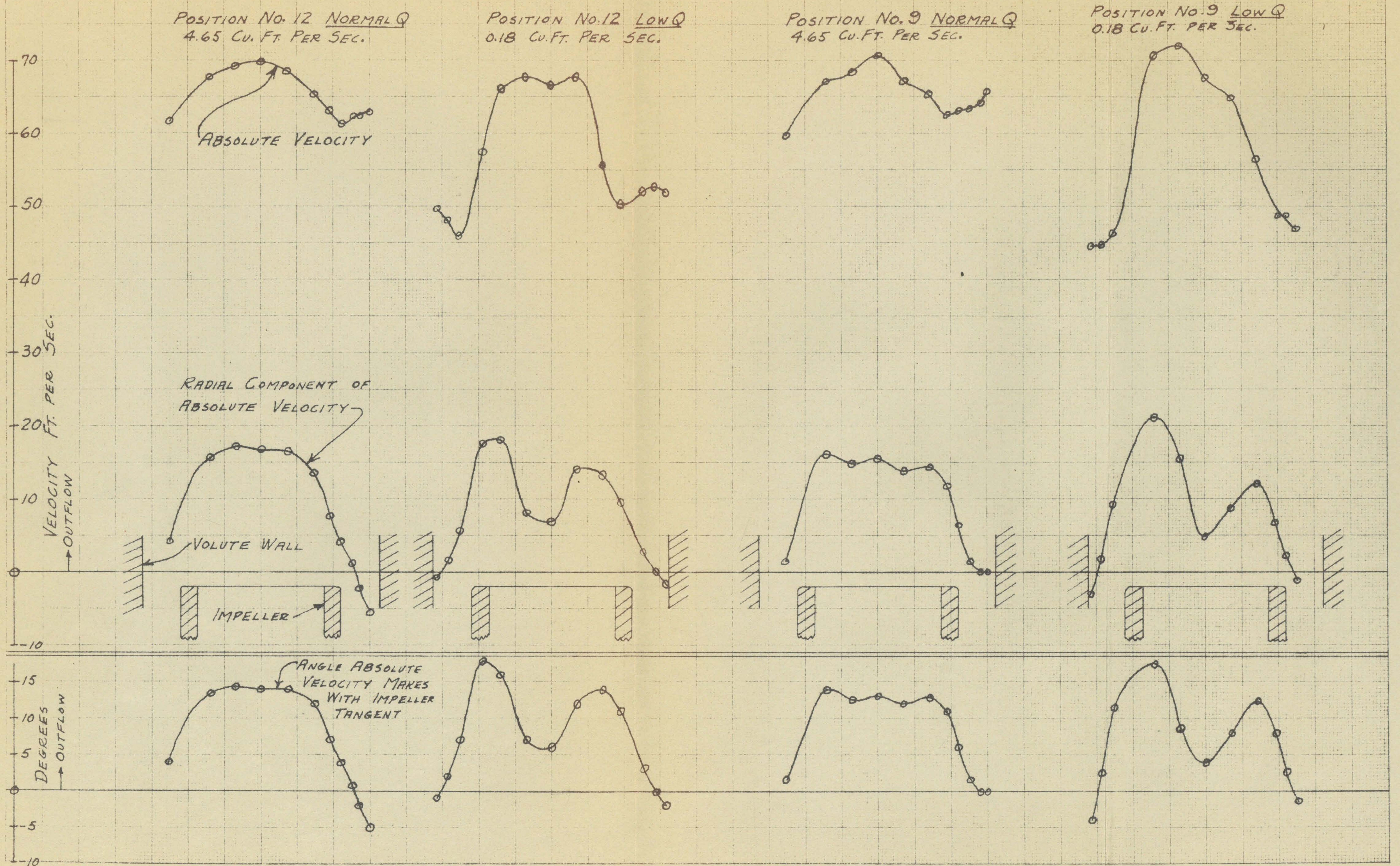

Instantaneous Velocity Traverses Across Volute - Byron Jackson Double Suction Pump
2000 R.P.M. 0° Phase Angle
Impeller Velocity = 116.8 Ft. per Sec.
Position No. 1 3/8" From Pitot Center to Impeller Normal Q 4.65 Cu. Ft. per Sec.
Position No. 2 1/4" From Pitot Center to Impeller Normal Q 4.65 Cu. Ft. per Sec.
Position No. 2 Low Q 0.38 Cu. Ft. per Sec.
Position No. 6 1 3/4" From Pitot Center to Impeller Normal Q 4.65 Cu. Ft. per Sec.
Absolute Velocity
Radial Component of Absolute Velocity
Volute Wall
Impeller
Velocity - Ft. per Sec.
Outflow
Angle Absolute Velocity Makes With Impeller Tangent
Degrees
Outflow
70
60
50
40
30
20
10
0
-10
15
10
5
0
-5
-10

Fig. 25

Instantaneous Velocity Traverses Across Volute - Byron Jackson Double Suction Pump
2000 R.P.M. 0° Phase Angle 3/8" From Pitot Center to Impeller

Impeller Velocity = 116.8 Ft. Per Sec.



Referring to the radial velocity profiles at low pump discharges, in each traverse there are two outstanding high velocity peaks. The most reasonable explanation of these peaks lies in a consideration of the centrifuge action of the shroud. Prof. von Kármán has suggested a calculation to help explain this matter. In the following calculation no claim is made to express exactly the extremely complicated conditions in a pump, but the computation serves to give the order of magnitude of several factors. Prof. von Kármán[1] has treated the problem of the frictional resistance of a rotating disc for the case of turbulent flow. He considered a smooth flat disc wetted on one side. The various momentum changes were taken into account, and the velocity distribution in the boundary layer was taken in accordance with the seventh-root law. The following expressions were derived in the treatment:

$$c_0 = 0.162\,R\omega \qquad \delta = 0.522\,R\left(\frac{\nu}{R^2\omega}\right)^{\frac{1}{5}}$$

where c_0 = maximum radial velocity in the boundary layer

R = radius of disc

ω = angular velocity of the disc

δ = thickness of the boundary layer

ν = kinematic viscosity of the fluid

The above is for that of fluid on one side of the disc. For a disc wetted on both sides it would be as shown diagramatically in Fig. 29(page 45).

---

21) "Über Laminare und Turbulente Reibung"

Z. angew. Math, Mech., vol. 1, 1921

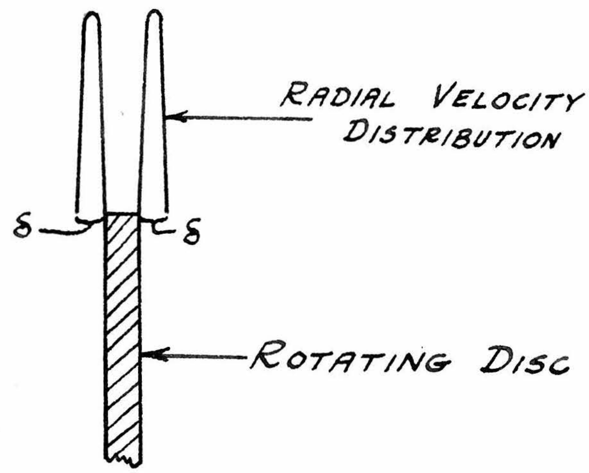


FIG. 29

Applying the above two relations to the particular case of this double suction pump test, and taking for R the radius to the periphery of the impeller, there results;

$$C_0 = 18.9 \text{ FT. PER SEC.}$$

$$\delta = 0.15''$$

It is interesting to note that some of the measured radial velocity peaks are close to this computed value of C_0 while the measured radial velocities over the center of the impeller lie between 5.0 and 7.5 feet per sec. Note also that most of the peak velocities occur at a point within the computed boundary layer thickness. These facts give a good indication that while the flow over the impeller center is being held back each shroud acts as a centrifuge to discharge a sheet of water into the volute.

The question might be raised as to why there is a lack of a "shadow" in the velocity profiles, since, referring to

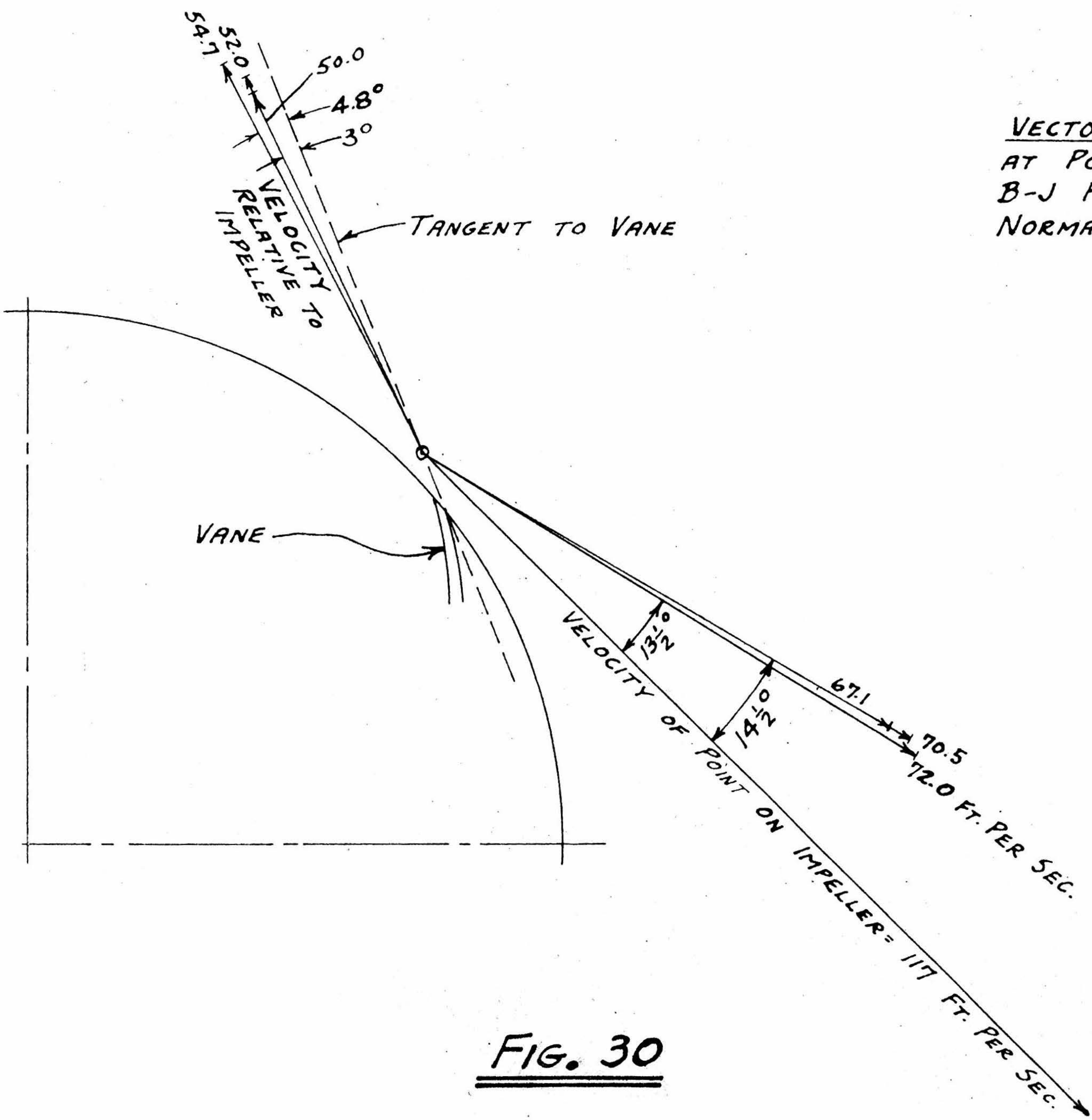
Fig. 29(page 45) directly over the shroud thickness there should be no radial flow. For this pump each shroud was about 3/16" thick. Also take into account the high value of c_θ and the fact that the pump is being throttled. In view of these factors it is quite possible that in a very short radial distance the flow from either side of the shroud could diverge, and that in this short distance these divergences could combine to give an appreciable positive velocity over the shroud thickness.

There is another question as to the source of the water which the outer face of the shroud may discharge into the volute. Another pump test at low capacity showed radial inflow (flow towards the axis of rotation) completely across the volute, in a short region at the upstream side of the tongue. From this experimental evidence it seems quite likely that at low pump capacities there is water available for the outer face of the shroud to discharge into the volute.

All these considerations serve to offer the most reasonable explanation of the unusual radial velocity profiles found at low pump discharges.

Fig. 30(page 47) show the various velocity vectors at Position No. 2 which is the nearest Pitot station to the impeller.

VECTORS FROM MEASUREMENTS
AT POS. NO. 2 OVER IMPELLER
B-J PUMP 2000 R.P.M.
NORMAL Q = 4.65 CU. FT. PER SEC.



72.0 CENTER
70.5 9/16" LEFT
67.1 9/16" RIGHT

FIG. 30

PITOT TUBE MEASUREMENTS AT DIFFERENT PUMP DISCHARGES

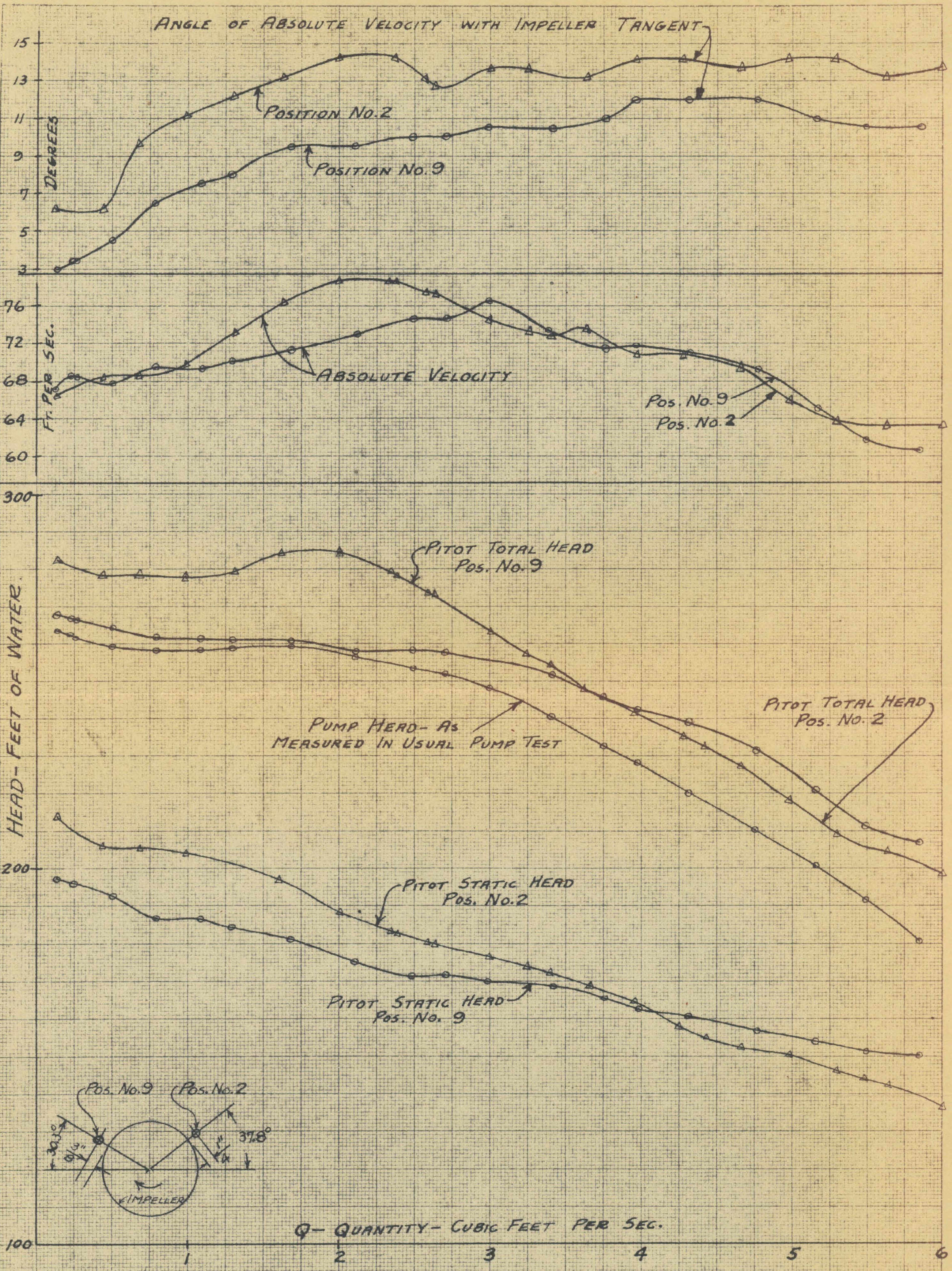
With the Pitot tube at one position in the center of the volute, the pump capacity was varied. The results are shown in Fig. 31(page 49).

One important item of considerable practical interest is shown by the Pitot static pressures at the two stations nearly 110 degrees apart. At low flows there is an appreciable difference in static pressure showing evidence of an unbalanced force on the impeller. At low flows the pump is a good fatigue testing machine, and this unbalanced force may be a serious matter. It may be enough to deflect the impeller to cause wearing of the seal rings, increase clearances, drop efficiencies, and in severe cases to break the pump shaft. Note that at high pump flows the unbalanced force changes direction.

COMPARISON WITH MOMENTUM LAW

If it is assumed that a "free vortex" exists in the volute, the angular momentum will be constant and the product RV is constant, where R is the radius to some point in the volute, and V is the tangential velocity at that point. Fig. 32(page 50) shows a close agreement for one case.

With this double suction pump the investigation was limited because it was not possible to provide Pitot stations around the entire volute. However, in a single suction pump it was possible to provide stations around the entire volute, and so the following section gives the results of a more complete study.



PITOT TUBE MEASUREMENTS
INSTANTANEOUS VALUES
BYRON JACKSON DOUBLE SUCTION PUMP
2000 R.P.M. 15° PHASE ANGLE
CENTER POSITION
NORMAL Q = 4.65
+40 FOOT SUCTION

FIG. 31

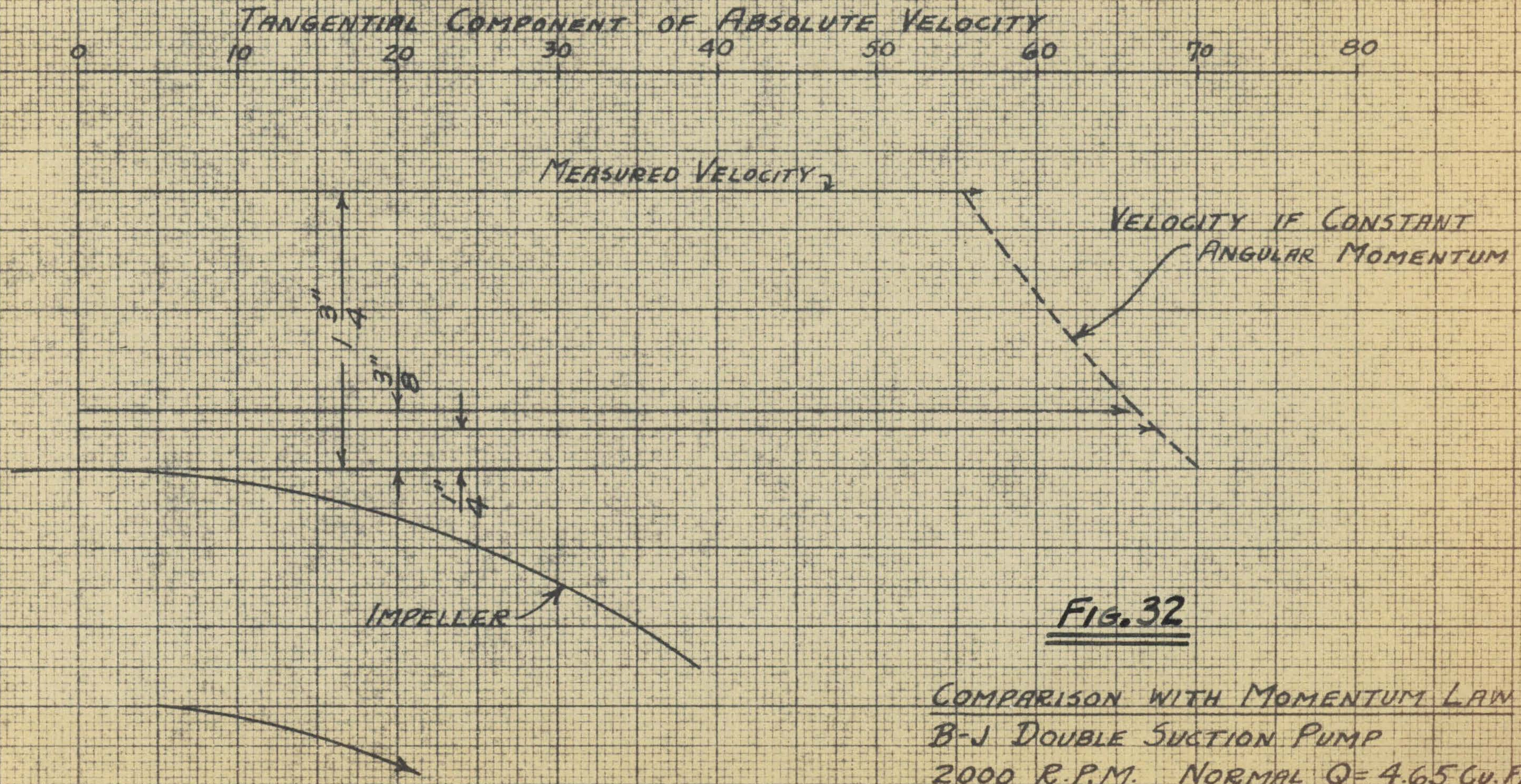


FIG. 32

COMPARISON WITH MOMENTUM LAW
B-J DOUBLE SUCTION PUMP
2000 R.P.M. NORMAL Q = 4.65 CU. FT. PER SEC.
CENTER POSITION PHASE ANGLE 0°
IMPELLER VELOCITY = 116.8 FT. PER SEC.

INSTANTANEOUS VELOCITY MEASUREMENTS

ON WORTHINGTON 7" SINGLE SUCTION CENTRIFUGAL PUMP

PUMP RATING AND DIMENSIONS

Serial No. 887789

Size	7"
Type	Single Suction
Capacity	2400 G.P.M.
Head	360 Feet
Speed	2900 R.P.M.
Specific Speed	1720

Impeller Outside Diameter	$12\frac{1}{2}$"
Impeller Inside Width	1 7/32", 1 15/64"
Impeller Outside Width	1 17/32"
Number of Vanes	7

All pump runs were made at 2500 R.P.M. The maximum efficiency (88.6%) at 2500 R.P.M. was the same as at the rated speed. All tests were made at plus 40 feet inlet head.

Fig. 33(page 52) shows the spacing of the Pitot tube stations in the volute.

Extensive tests were made at three different pump discharges- "normal", "low", and "high". "Normal" pump discharge is that at the point of maximum pump efficiency. "Low" refers to a discharge less than "normal", and "high" refers to a discharge greater than "normal".

DIMENSIONS OF PITOT LOCATIONS

POSITION NO.	"R" RADIAL DISTANCE	"θ" DEGREES	DISTANCE PITOT CENTER TO IMPELLER
A	6.70"	25.5	0.45"
B	6.68"	49.8	0.43"
C	6.68"	67.1	0.43"
D	6.66"	90.2	0.41"
E	6.67"	135.4	0.42"
F	6.68"	157.6	0.43"
G	6.69"	224.9	0.44"
H	6.68"	269.1	0.43"
I	6.72"	295.4	0.47"
J	6.71"	314.5	0.46"
K	6.71"	334.1	0.46"
L	6.71"	354.8	0.46"

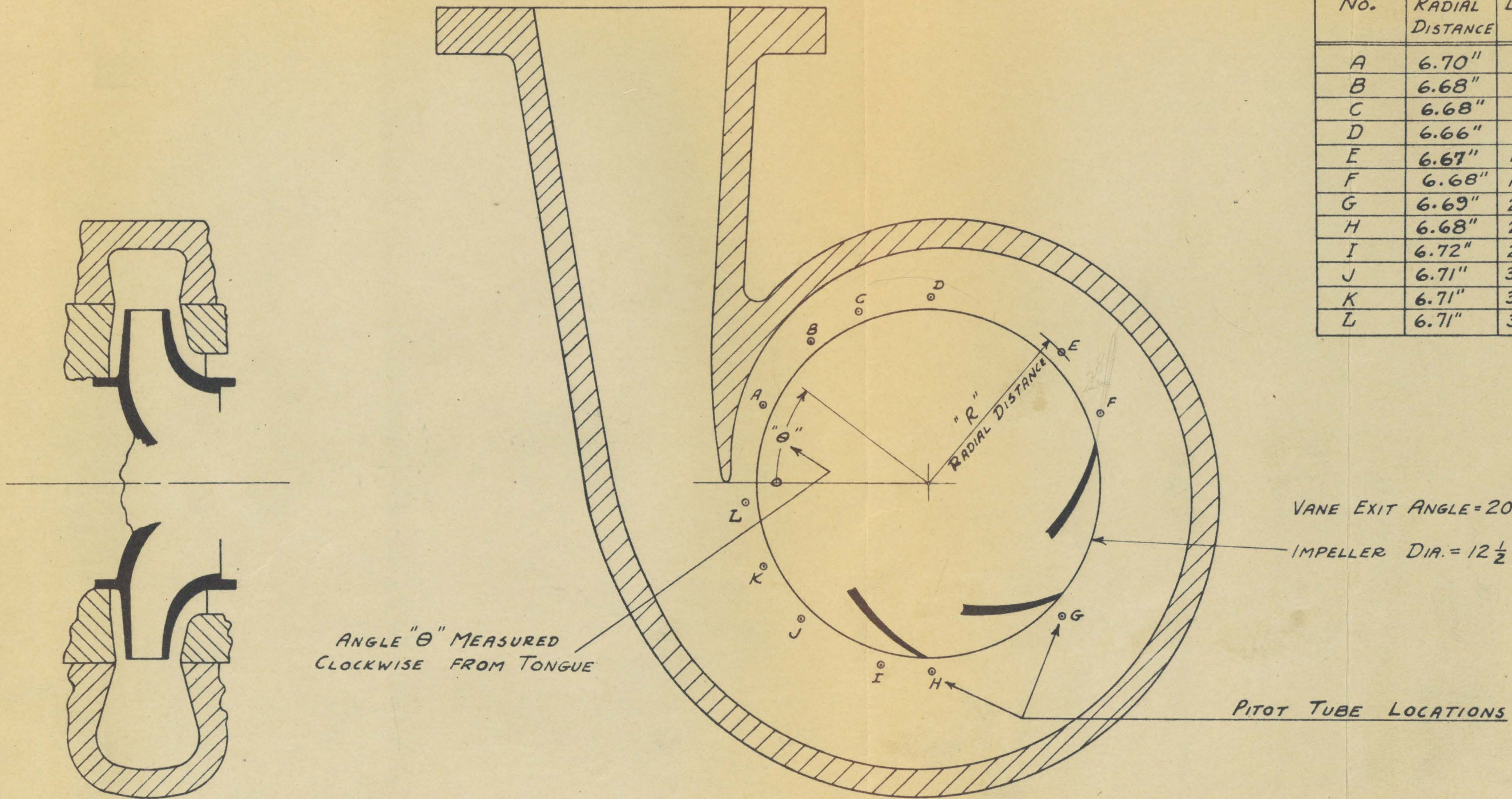


FIG. 33

LOCATION OF PITOT HOLES

WORTHINGTON SINGLE SUCTION PUMP
SCALE: $\frac{1}{4}$ SIZE

INSTANTANEOUS VELOCITY DISTRIBUTION BETWEEN VANES

Fig. 34 to 39 inclusive (pages 54 to 59 inclusive) show the results of measurements as the Pitot tube was kept at a fixed position across the volute and the phase angle varied. Note carefully the coordinate scales used.

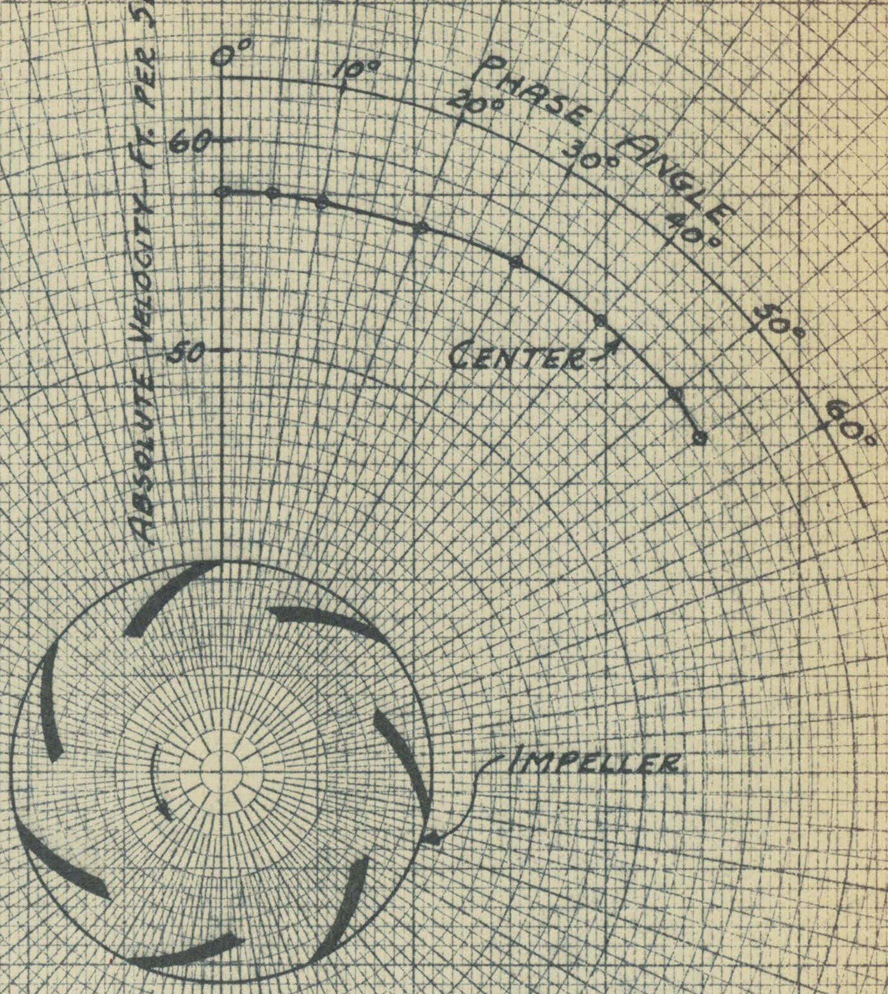
Fig. 34 and 35 show that there was practically no velocity variation at normal pump discharge. This was the same as found on the double suction pump (page 36).

Fig. 36 and 37 shows a slight change in velocity with phase angle at low capacity. The flow is not exactly periodic. At low pump capacity the velocity head fluctuations were much greater than at normal capacity (the head fluctuations were not great enough to seriously affect the accuracy of the measurements). The qualitative experiments of Fischer and Thoma[1] have shown that at very low pump capacities " the flow conditions were characterized by two different criteria: 1) The outward flow was no longer stable and varying conditions of reverse flow appeared. 2) At any given time different flow conditions were to be found in the different water passages". Reverse flow here refers to the dead-water zone composed of whirls and eddies, the zone being on the low pressure side of the vane. Thus the present measurements agree to some extent with the findings of Fischer and Thoma. However, the present measurements show only a slight velocity variation at low pump capacity, and any comparison should take into account the remarks made in the Introduction of this paper.

Fig. 38 and 39 show a slight change in velocity with phase angle at high pump discharge.

---

8) A.S.M.E. Transactions, 1932

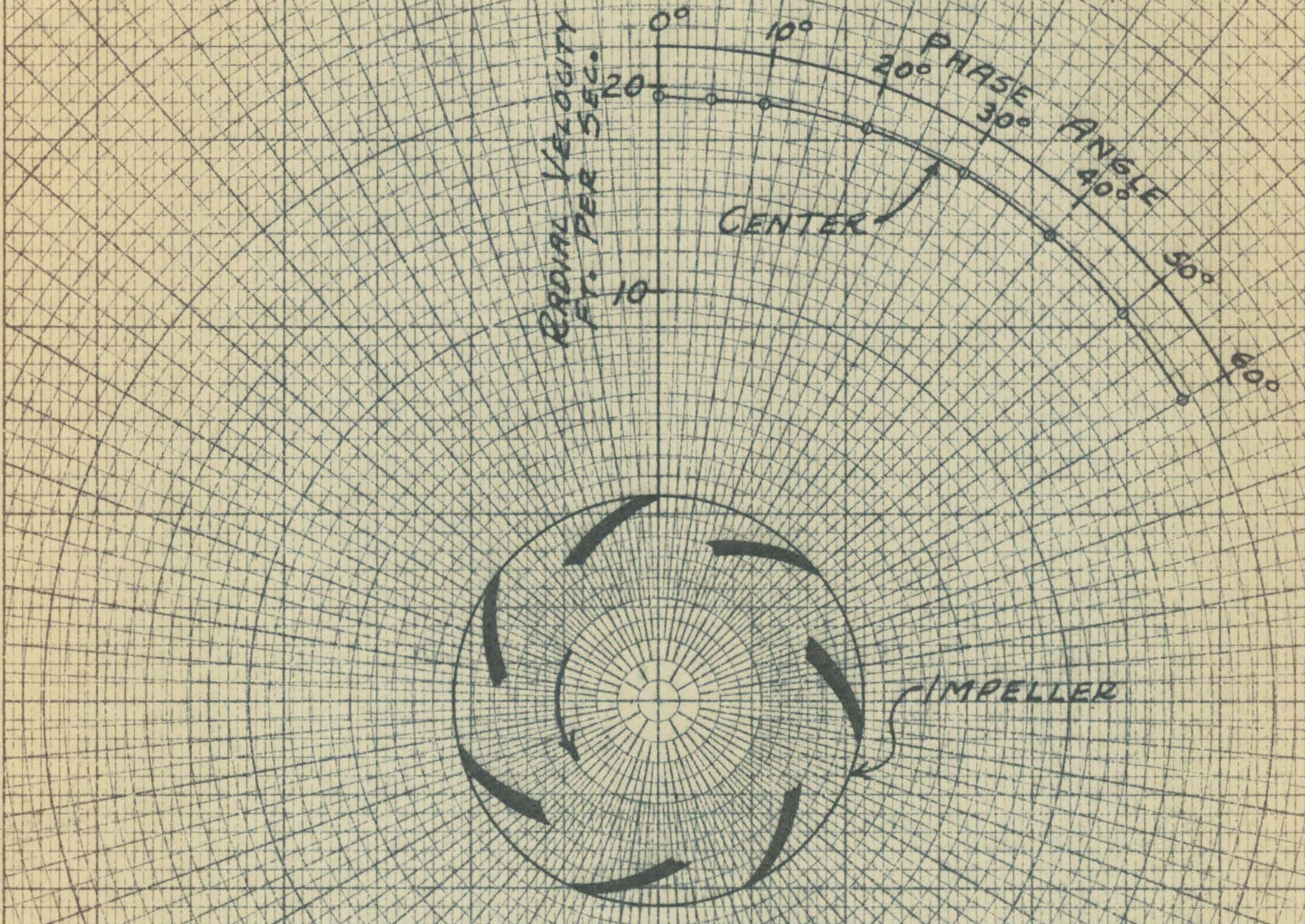


INSTANTANEOUS VELOCITY DISTRIBUTION
BETWEEN VANES.

POSITION NO. 6 { 0.46" FROM IMPELLER TO PITOT CENTER

WORTHINGTON SINGLE SUCTION PUMP
2500 R.P.M.
NORMAL Q 5.29 CU. FT. PER SEC.

FIG. 34



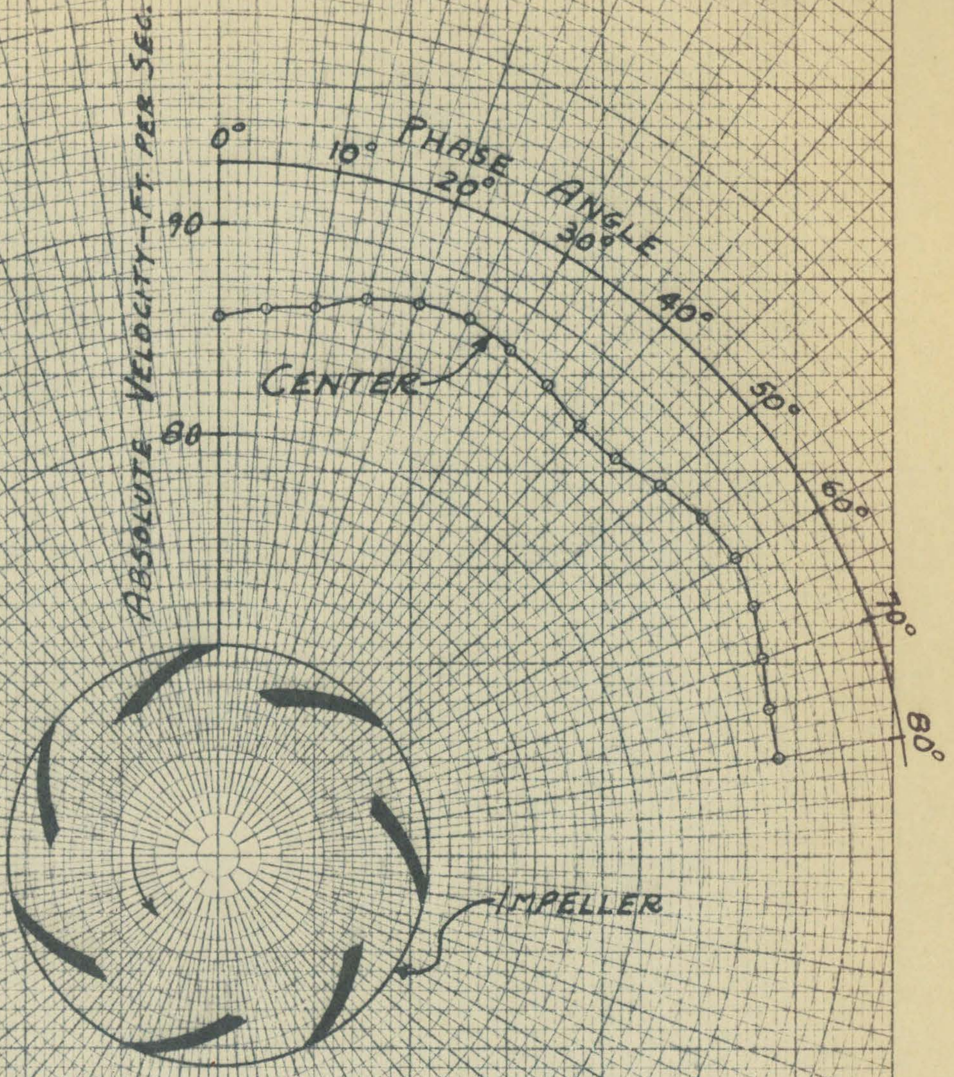
INSTANTANEOUS VELOCITY DISTRIBUTION
BETWEEN VANES

POSITION NO. L { 0.46" FROM IMPELLER TO PITOT CENTER

WORTHINGTON SINGLE SUCTION PUMP
2500 R.P.M.

NORMAL Q 5.29 CU. FT. PER SEC.

FIG. 35



INSTANTANEOUS VELOCITY DISTRIBUTION BETWEEN VANES

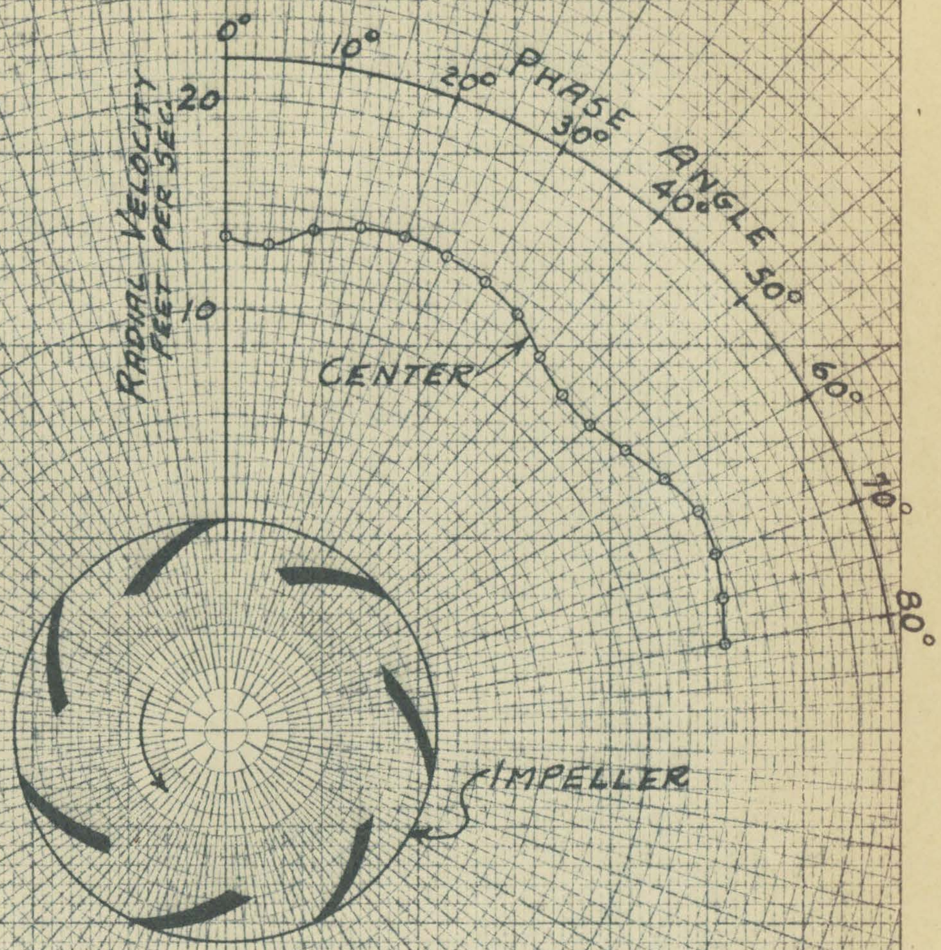
POSITION NO. A {0.45" FROM IMPELLER TO PITOT CENTER

WORTHINGTON SINGLE SUCTION PUMP

2500 R.P.M.

LOW Q 0.996 CU. FT. PER SEC.

FIG. 36



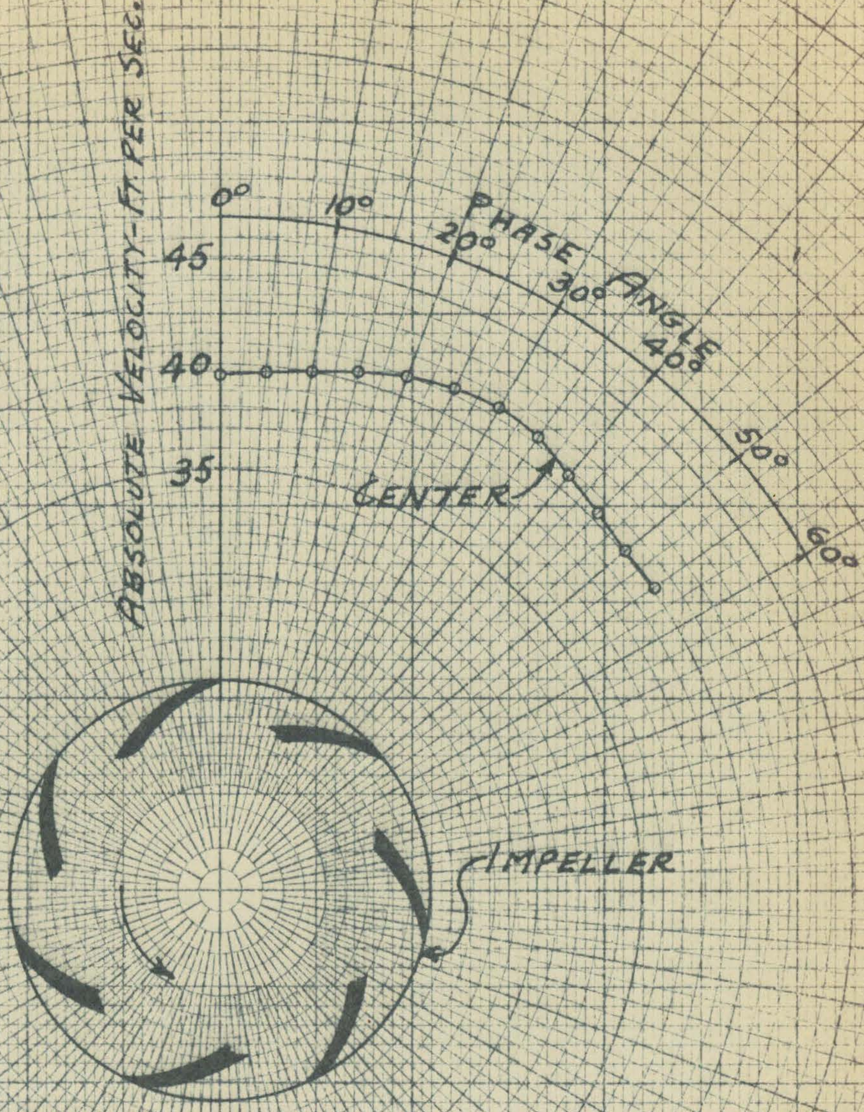
INSTANTANEOUS VELOCITY DISTRIBUTION BETWEEN VANES

POSITION NO. A {0.45" FROM IMPELLER TO PITOT CENTER

WORTHINGTON SINGLE SUCTION PUMP
2500 R.P.M.

LOW Q 0.996 CU. FT. PER SEC.

FIG. 37



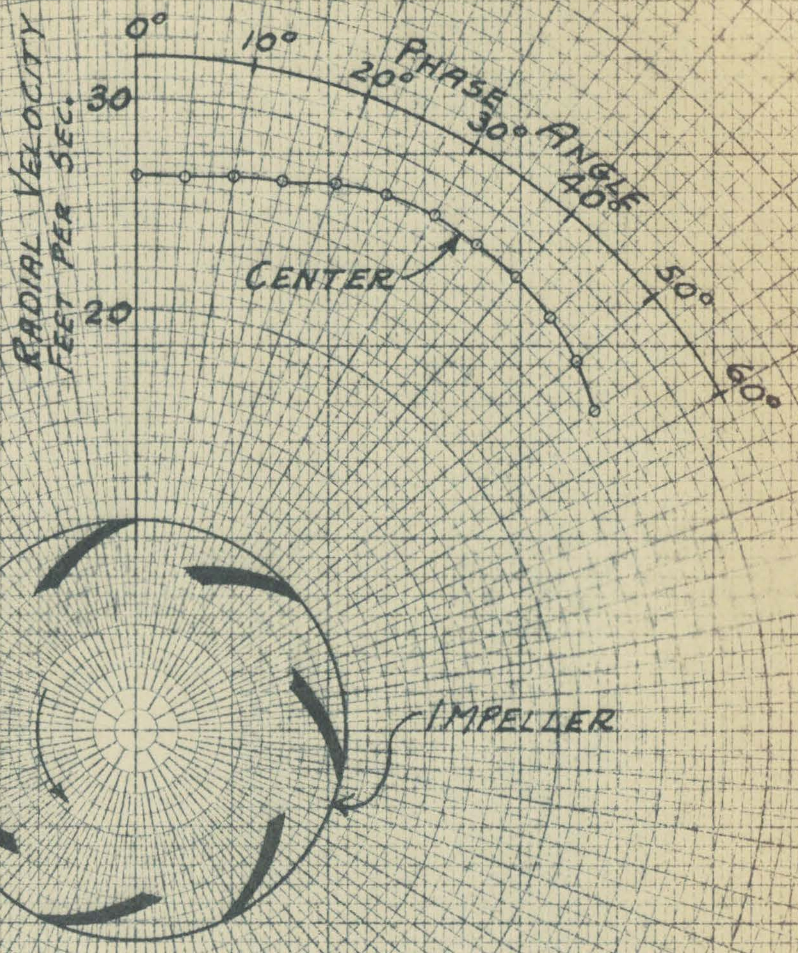
INSTANTANEOUS VELOCITY DISTRIBUTION
BETWEEN VANES

POSITION NO. A {0.45" FROM IMPELLER TO PITOT CENTER

WORTHINGTON SINGLE SUCTION PUMP
2500 R.P.M.

HIGH Q 7.51 CU. FT. PER SEC.

FIG. 38



Instantaneous Velocity Distribution
Between Vanes

Position No. A {0.45" from Impeller to Pitot Center

Worthington Single Suction Pump

2500 R.P.M.

High Q 7.51 Cu. Ft. per Sec.

Fig. 39

VELOCITY TRAVERSES ACROSS VOLUTE

A) NORMAL PUMP CAPACITY

Fig. 40 and Fig. 41 (pages 66 and 67) show the profiles obtained from Pitot tube traverses across the volute, each traverse being made at a constant phase angle. Recalling the notation discussed on page 31, the right side of the impeller center is the inlet side of the pump. Because of a projection on the pump at the time of measurement Position I is not a complete traverse.

In the familar discussion of double suction pump vs. single suction pump it is sometimes claimed that the double suction pump gives a better or more symmetrical velocity distribution. A comparison of Fig. 40 and Fig. 41 with Fig. 25 and Fig. 26(pages 42 and 43) shows that the absolute velocity profiles of the double suction pump are slightly more symmetrical than those of the single suction pump. For a majority of the traverses in the single suction pump the maximum absolute velocity is to the right of the impeller center, while the double suction pump profiles show the maximum absolute velocities at the impeller center.

These Pitot measurements show one general result and that is that the flow conditions are not uniform around the volute. Fig. 40 and Fig. 41 show that the profiles differ in character and magnitude. Measurements at other pump capacities yield another general result and that is that the flow conditions at normal discharge are more uniform than at low and high capacities. This latter result is quite reasonable when account is taken of the fact that at normal capacity the pump has reached its maximum efficiency, and therefore the flow conditions should be the most favorable.

This lack of uniformity of flow conditions is indicated in Fig. 42(page 68), a plot of average radial velocity vs. angle around the volute. For each Pitot station in the volute the average radial velocity was found from a complete traverse across the volute. Integration of this curve and a consideration of the area gives a net quantity of flow which should agree with the flow registered by the venturi meter. Fig. 42 shows that the agreement is not exact, but the agreement is quite close taking into account the considerable separation of some of the Pitot stations. This agreement serves as a check upon the reliability of the Pitot tube measurements. More Pitot stations could not have been provided because of projecting studs and obstructions on the pump.

VELOCITY TRAVERSES ACROSS VOLUTE

B) LOW PUMP CAPACITY

Fig. 43 and Fig. 44(pages 69 and 70) show the profiles obtained from Pitot tube traverses across the volute at low pump discharge.

The profiles differ very much in both character and magnitude, and show that the flow conditions at low capacity are very much less uniform than at normal capacity. There is a more or less progressive change in the profiles from station to station. Note particularly the cases of inflow, that is, flow towards the axis of rotation. At Position L there is inflow completely across the volute. With the flow throttled, there is probably a dead water zone in the diffuser, and thus the impeller outflow(in other parts of the volute) is forced back in towards the impeller at the tongue.

Fig. 45(page 71) shows the average radial velocity distribution around the volute. Between Positions L and A there is a very appreciable, abrupt change in the average radial velocity, and naturally a question exists as to what curve to draw between Positions L and A. The one given seems the most reasonable and conservative. On the basis of this curve the quantity of flow indicated by the Pitot agrees very closely with the flow registered by the venturi meter. This close agreement in these erratic flow conditions speaks well for this technique of measurement.

Referring to the radial velocity profiles in Fig. 43 and Fig. 44, in many of the traverses are found two outstanding high velocity peaks, as was observed in the double suction pump tests (page 44). Applying the two relations discussed on page 44 to the particular case of this single suction pump test there results:

$$C_0 = 22.1 \text{ FEET PER SEC.}$$

$$\delta = 0.14''$$

where C_0= maximum radial velocity in the boundary layer

δ= thickness of the boundary layer

Here, as in the double suction pump test, one notes that some of the measured radial velocity peaks(in the cases of double peak profiles) are close to this computed value of C_0. Note also, as was observed in the double suction pump test, that many of the peak velocities occur at a point within the computed boundary layer thickness. Undoubtedly other factors besides the shroud are affecting the impeller flow, but the present point is that the most reasonable explanation of the high velocity peaks states that these peaks are due to the centrifuge action of the shroud. The inflow at such stations as Position L could easily furnish water for the outer face of each shroud to discharge into the volute.

VELOCITY TRAVERSES ACROSS VOLUTE

C) HIGH PUMP CAPACITY

Fig. 46 and Fig. 47(pages 72 and 73) show the profiles obtained from Pitot tube traverses at high pump capacity. The traverses are not complete, but the measurements are useful to some extent in a comparison with the results of tests at normal and low pump capacities.

There is a definite progessive change in the absolute velocity profiles from Position A to Position F. Some of the radial velocity profiles are smoother and more symmetrical than those at normal capacity, while others are as distorted as those at low capacity.

Fig. 48(page 74) show that the radial velocity distribution around the volute is much less uniform than that at normal pump discharge.

UNBALANCED RADIAL FORCES ACTING ON IMPELLER

Fig. 49(page 75) shows the static pressure distribution around the volute as given by the Pitot tube. The static pressure plotted is that developed by the pump, and thus does not include the inlet pressure. The dotted horizontal lines indicate the mean static pressure value for each pump capacity.

The static pressure shows the same general trend as was observed with other characteristics of the pump flow. Fig. 49 shows that at normal capacity the static pressure is much more uniform around the volute than that at high and low capacities. Furthermore, the static pressure is not exactly uniform at normal capacity. Lack of uniformity indicates an unbalanced static pressure force.

Fig. 50(page 76) shows both the unbalanced static pressure and the momentum forces acting on the impeller. In the absence of other definite information the width of the impeller was taken as the area over which the static pressure acts. Because of a lack of uniform velocity distribution around the volute there is a momentum force acting on the impeller. Fig. 50 shows that the momentum force is small in comparison with the static pressure force. It is interesting to note two features about the static pressure force, the momentum force, and the resultant force- 1) each force is larger for low and high discharge than at normal discharge, 2) each force changes direction as the pump discharge is increased from low to high.

POWER BALANCE AROUND VOLUTE

Fig. 51(page ~~76~~ 77) shows a power balance around the volute for low and high pump discharges. The Pitot horse-power was computed from Pitot tube measurements of total head and radial velocity. The total head is that total head developed by the pump and therefore does not include the inlet head.

The area under each curve in Fig. 51 gives the total horse-power at the imaginary cylindrical surface which passes through the Pitot stations and which extends completely across the volute. For low capacity the inflow(towards the axis of rotation) accounts for the negative power values.

The shaft input horse-power is the power input at the pump shaft. The difference between the shaft input horse-power and the Pitot horse-power is labelled "impeller loss" for the sake of convenience. Actually this so-labelled "impeller loss" includes more than the loss in the impeller proper.

The power distribution is more uniform at normal capacity than at low capacity, and further, the power distribution is not exactly uniform at normal capacity.

FIG. 42
AVER. RADIAL VELOCITY
NORMAL Q

FIG. 42

AVERAGE RADIAL VELOCITY (AVERAGE OF TRAVERSES)
VERSUS
ANGLE AROUND VOLUTE

WORTHINGTON SINGLE SUCTION PUMP - 2500 R.P.M.

IMPELLER PERIPHERAL VELOCITY = 136.3 FT. PER SEC. PUMP HEAD = 274 FEET

NORMAL Q { 5.297 CU. FT. PER SEC. (VENTURI MEASUREMENT)
5.61 CU. FT. PER SEC. (PITOT TUBE MEASUREMENTS)

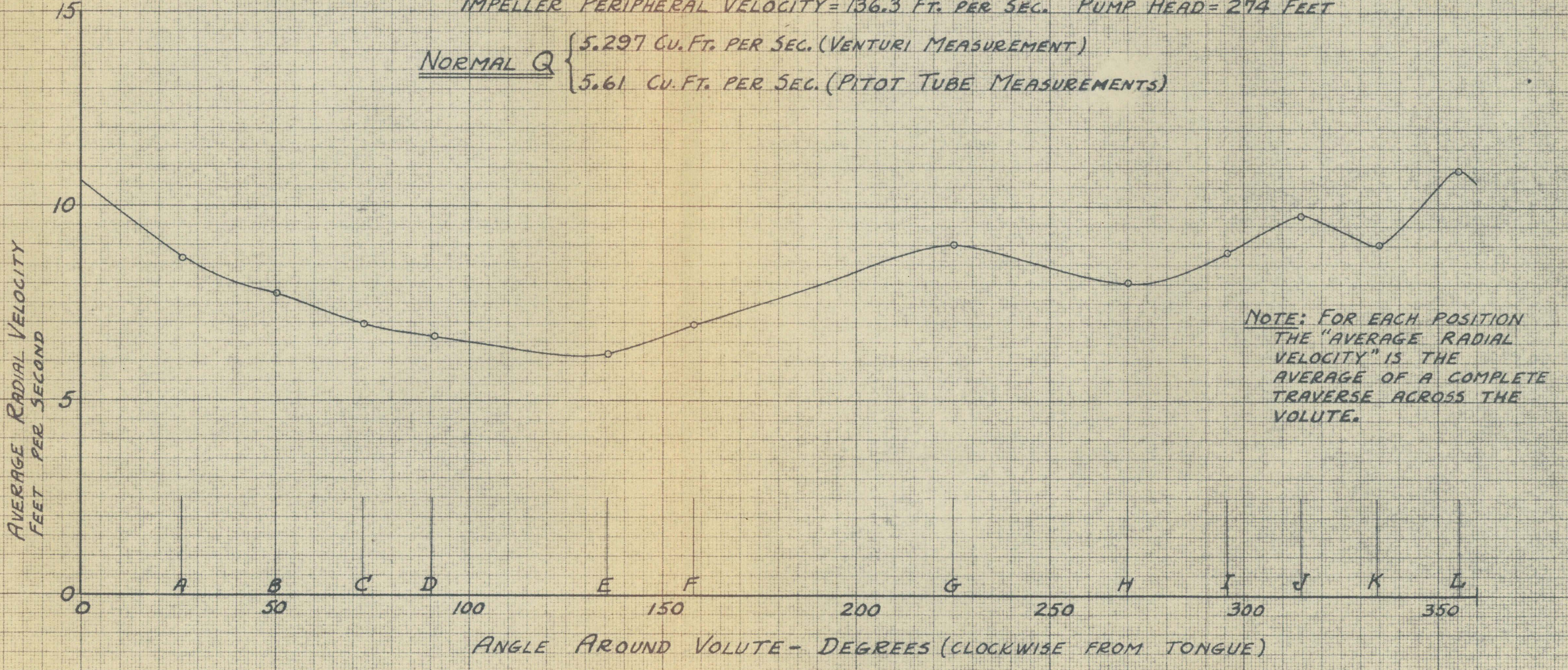


FIG. 45

AVER. RADIAL VELOCITY

LOW Q

FIG. 45

AVERAGE RADIAL VELOCITY (AVERAGE OF TRAVERSES)

VERSUS

ANGLE AROUND VOLUTE

WORTHINGTON SINGLE SUCTION PUMP - 2500 R.P.M.

IMPELLER PERIPHERAL VELOCITY = 136.3 FT. PER SEC. PUMP HEAD = 336 FEET

LOW Q — 0.996 CU. FT. PER SEC. (VENTURI MEASUREMENT)
1.03 CU. FT. PER SEC. (PITOT TUBE MEASURENT)

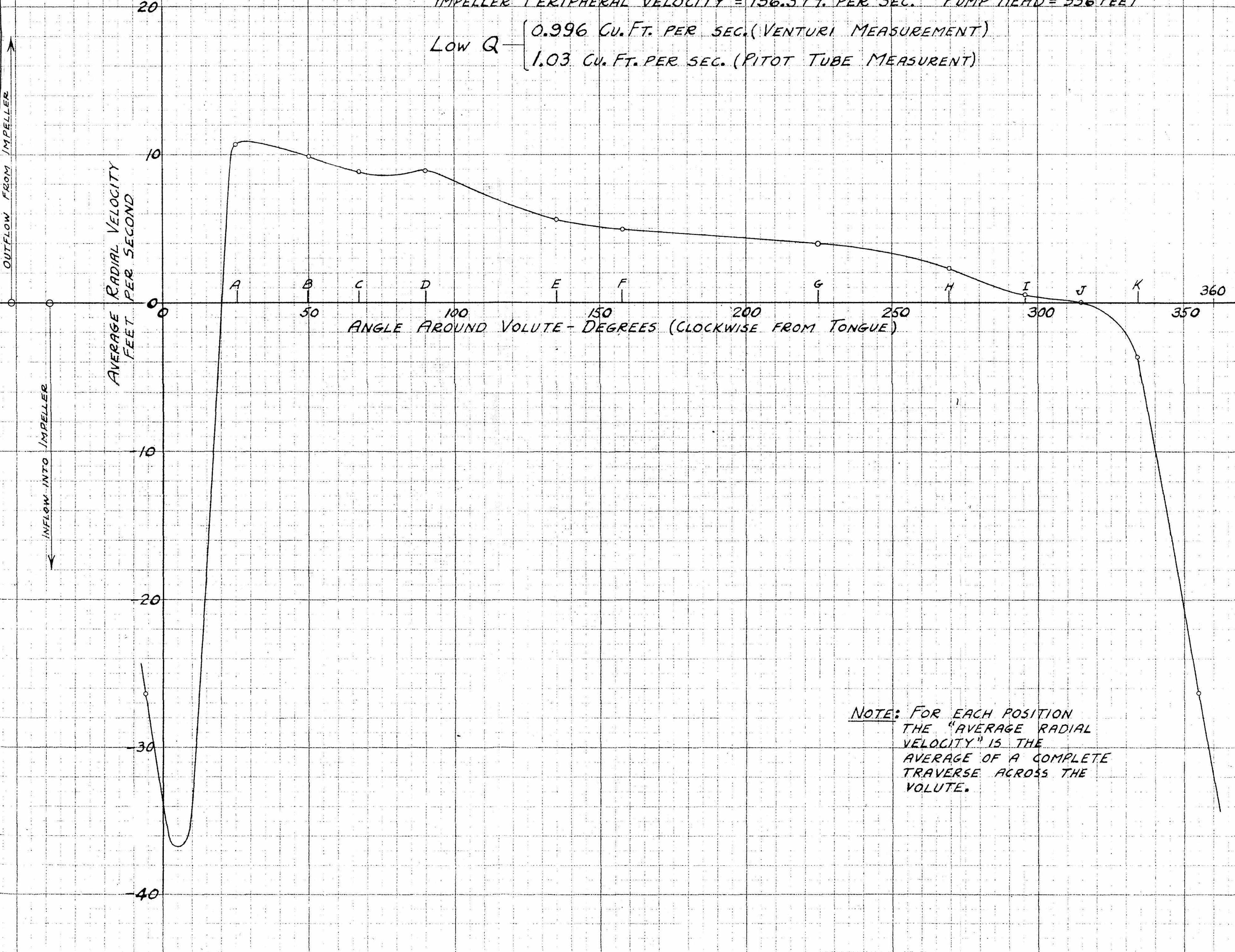


FIG. 48

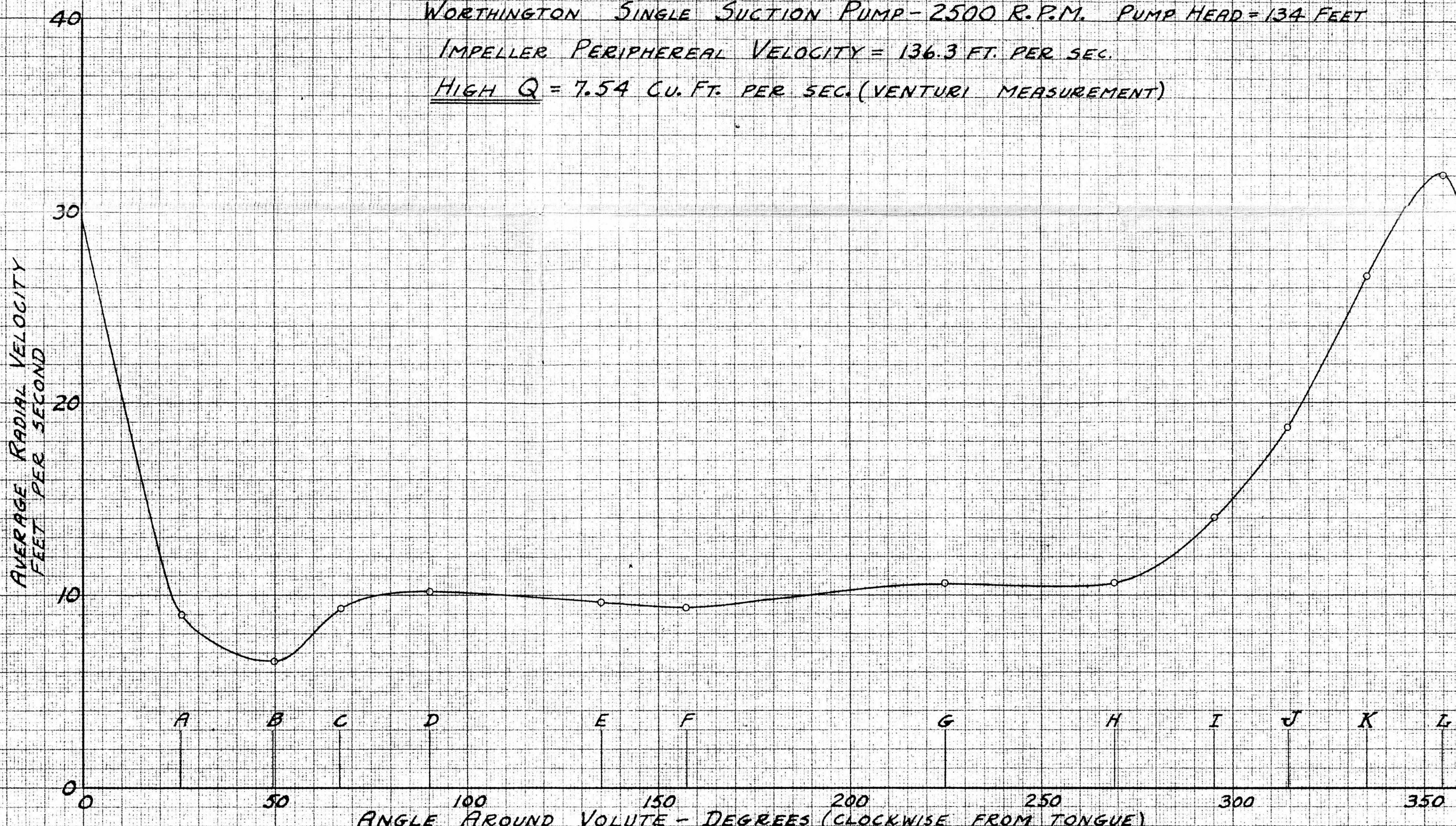
AVER. RADIAL VELOCITY

HIGH Q

FIG. 48

AVERAGE RADIAL VELOCITY (AVERAGE OF EXTRAPOLATED TRAVERSE)
VERSUS
ANGLE AROUND VOLUTE

WORTHINGTON SINGLE SUCTION PUMP - 2500 R.P.M. PUMP HEAD = 134 FEET
IMPELLER PERIPHEREAL VELOCITY = 136.3 FT. PER SEC.
HIGH Q = 7.54 CU. FT. PER SEC. (VENTURI MEASUREMENT)



NOTE: FOR SOME POSITIONS AT THIS PUMP DISCHARGE INCOMPLETE TRAVERSES WERE EXTRAPOLATED. FOR EACH POSITION THE "AVERAGE RADIAL VELOCITY" IS THE AVERAGE OF A COMPLETE TRAVERSE ACROSS THE VOLUTE.
INTEGRATION OF THESE TRAVERSES GIVES A NET DISCHARGE OF 8.94 CU. FT. PER SEC.

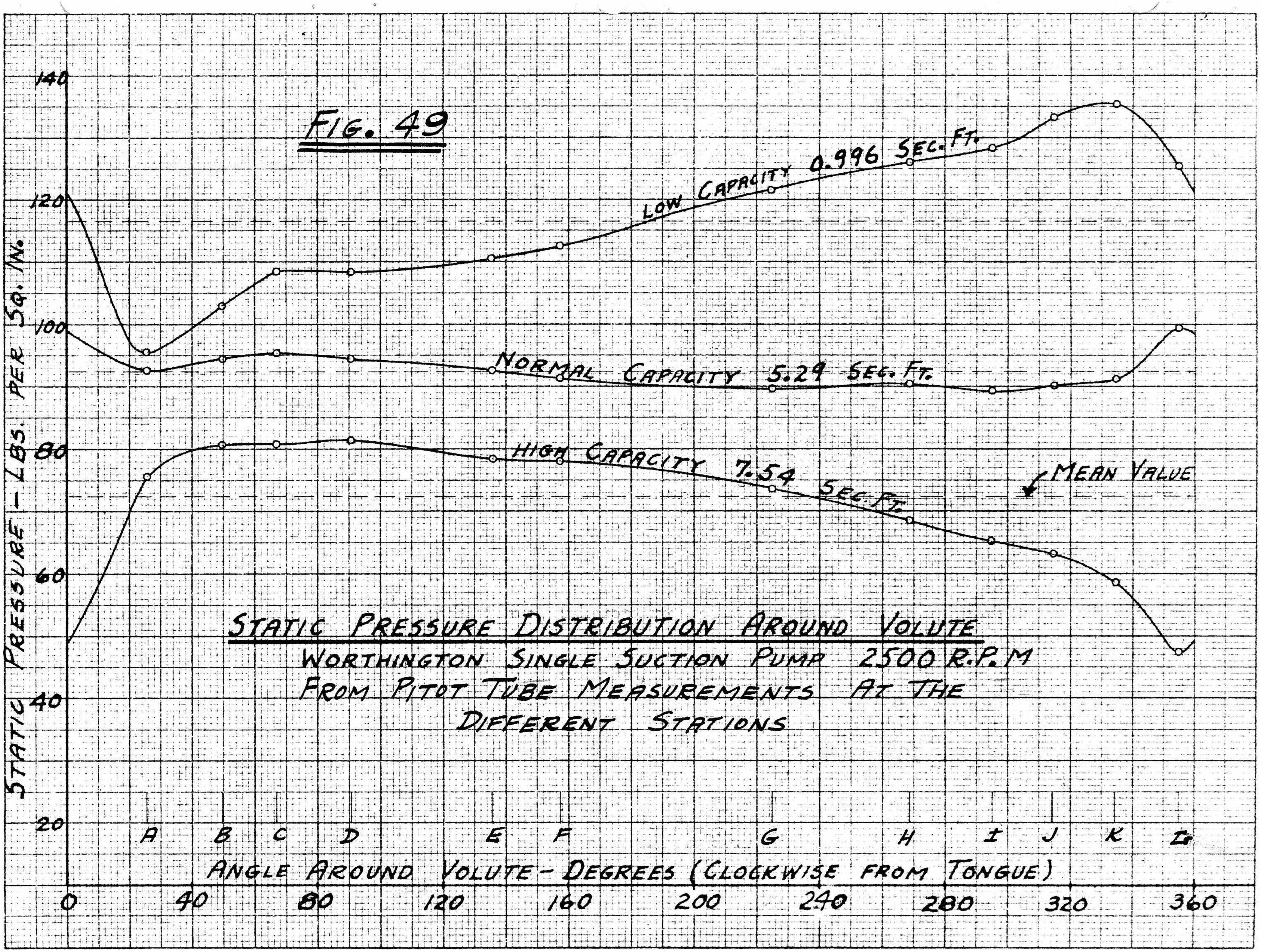
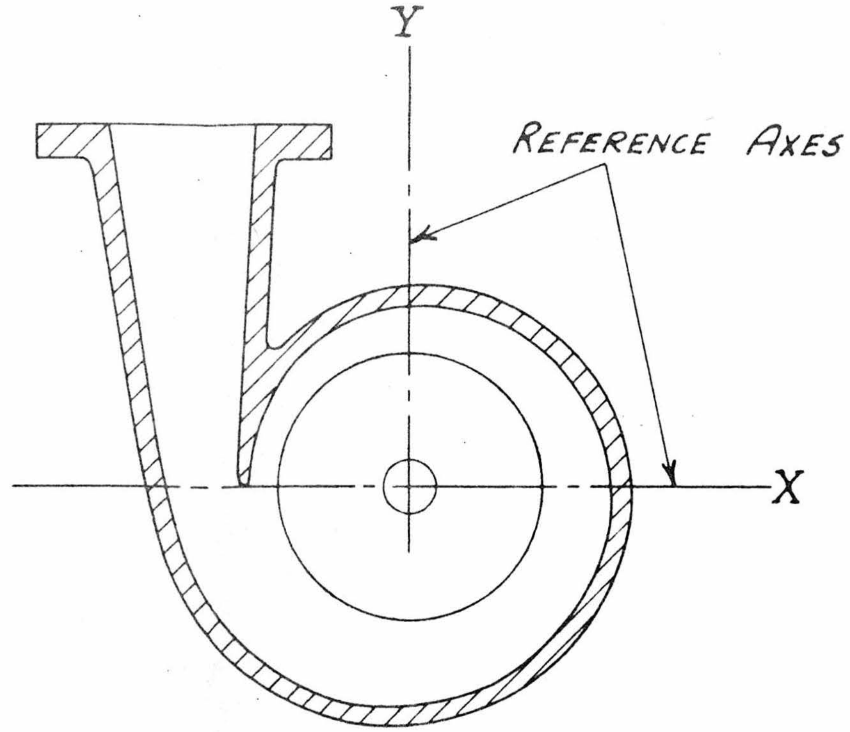
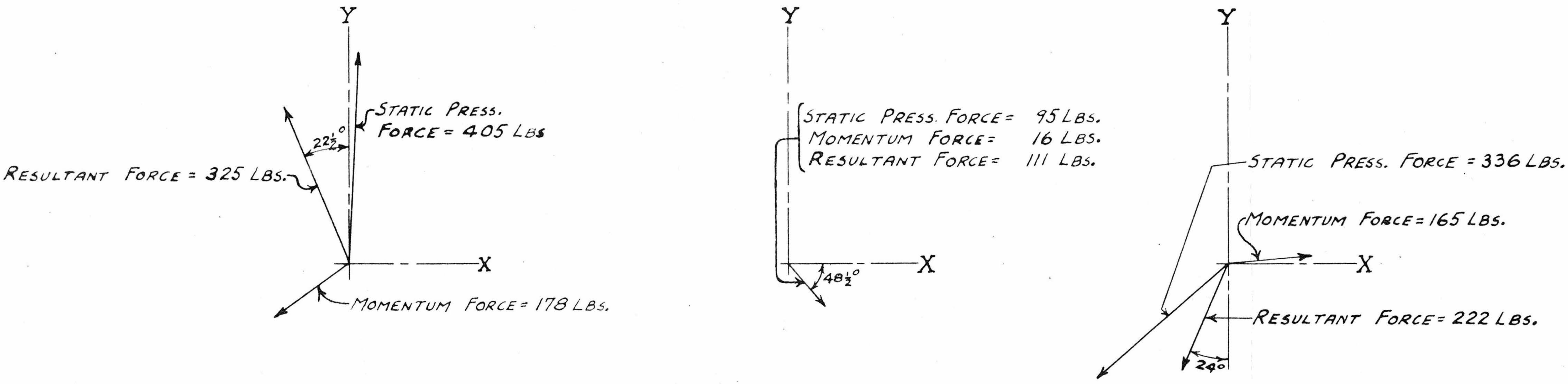


Fig. 49
Static Pressure Distribution Around Volute
Worthington Single Suction Pump 2500 R.P.M
From Pitot Tube Measurements At The
Different Stations
Low Capacity 0.996 Sec. Ft.
Normal Capacity 5.29 Sec. Ft.
High Capacity 7.54 Sec. Ft.
Mean Value
Static Pressure - Lbs. Per Sq. In.
Angle Around Volute - Degrees (Clockwise from Tongue)
A B C D E F G H I J K
0 40 80 120 160 200 240 280 320 360
20 40 60 80 100 120 140



NOTE: "STATIC PRESSURE FORCE" REFERS TO THE UNBALANCED STATIC PRESSURE ACTING ON AREA OF IMPELLER WIDTH ($1\frac{17}{32}$")



LOW CAPACITY
0.996 SEC. FT.

NORMAL CAPACITY
5.29 SEC. FT.

HIGH CAPACITY
7.54 SEC. FT.

FIG. 50 UNBALANCED FORCES ACTING ON IMPELLER (RADIAL FORCES)

WORTHINGTON SINGLE SUCTION PUMP
2500 R.P.M.

FIG. 51

POWER BALANCE AROUND

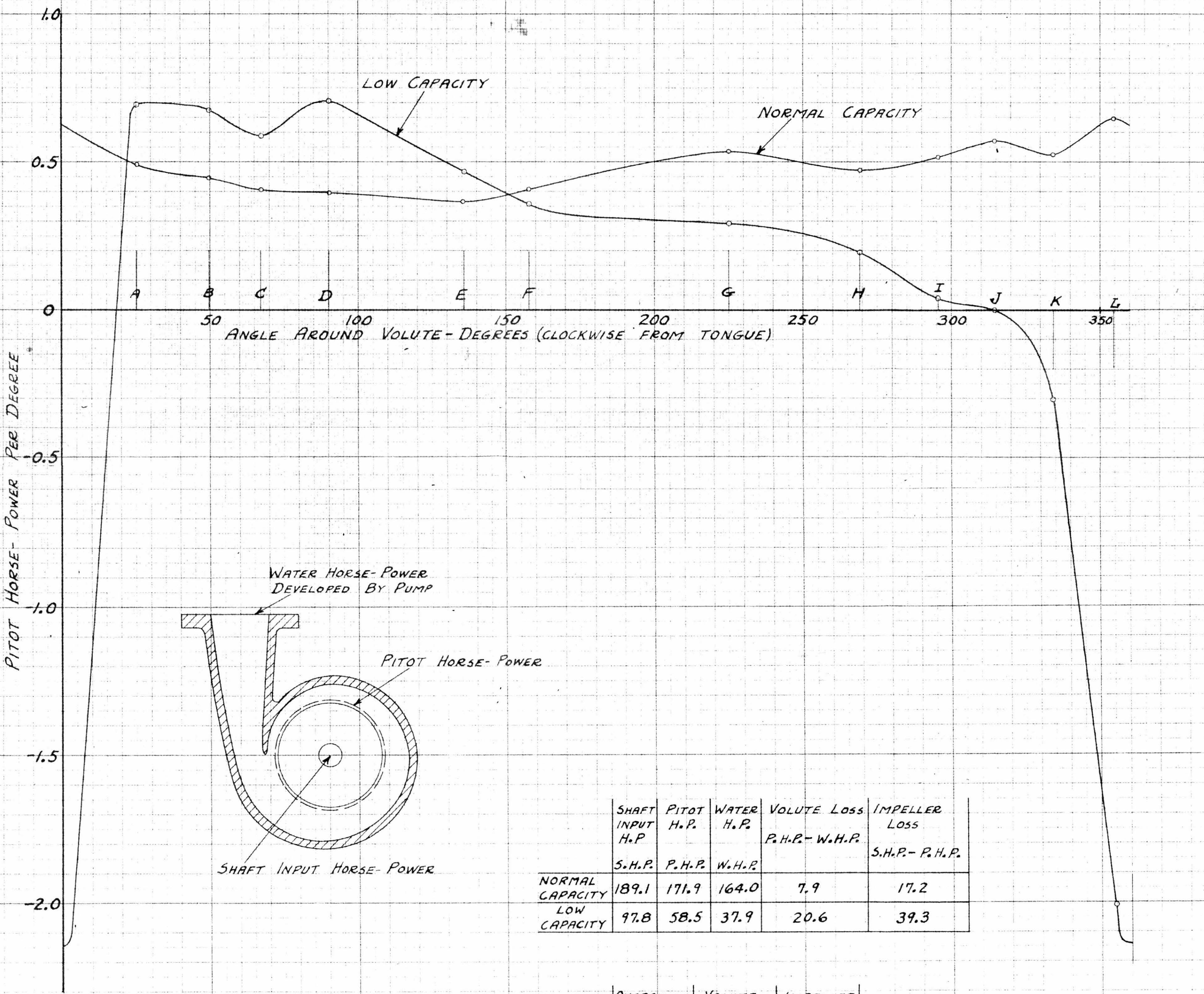
VOLUTE

Fig. 51

Power Balance Around Volute

Worthington Single Suction Pump - 2500 R.P.M.

"Pitot Horse-Power" refers to the power at an imaginary cylindrical surface passing through the pitot tube stations in the volute. This surface extends completely across the volute.



	Shaft Input H.P. S.H.P.	Pitot H.P. P.H.P.	Water H.P. W.H.P.	Volute Loss P.H.P.- W.H.P.	Impeller Loss S.H.P.- P.H.P.
Normal Capacity	189.1	171.9	164.0	7.9	17.2
Low Capacity	97.8	58.5	37.9	20.6	39.3

	Overall Efficiency $\frac{W.H.P.}{S.H.P.}$	Volute Efficiency $\frac{W.H.P}{P.H.P.}$	Impeller Efficiency $\frac{P.H.P.}{S.H.P.}$
Normal Capacity	86.7%	95.4%	90.9%
Low Capacity	38.7%	64.8%	59.8%

Normal Capacity = 5.29 Cu. Ft. per Sec. Pump Head = 274 Feet

Low Capacity = 0.996 Cu. Ft. per Sec. Pump Head = 336 Feet

DIRECTION OF VELOCITY VECTORS

Fig. 52 shows the notation used in designating the direction of the velocity vectors.

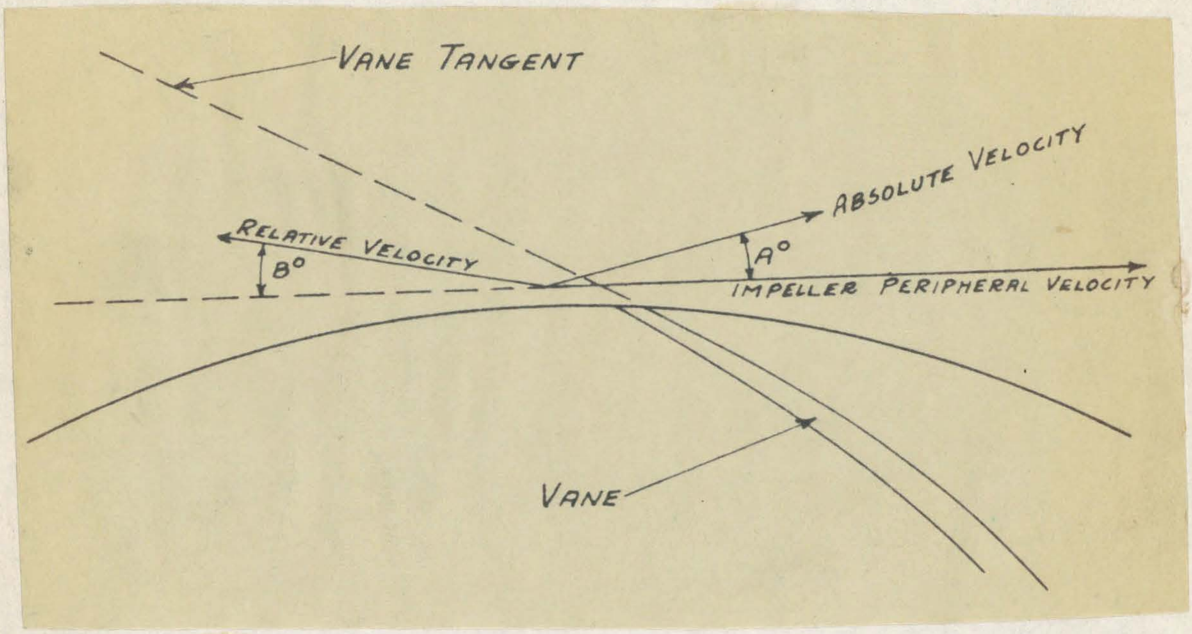


FIG. 52

Fig. 53 shows the average direction of the absolute velocity at different Pitot stations. On Fig. 40,41,43,44,46, and 47 are plotted the measured angles from traverses across the volute. Average angle "A" as shown on Fig. 53 is the average of a curve of these measured angles across the inner width of the impeller.

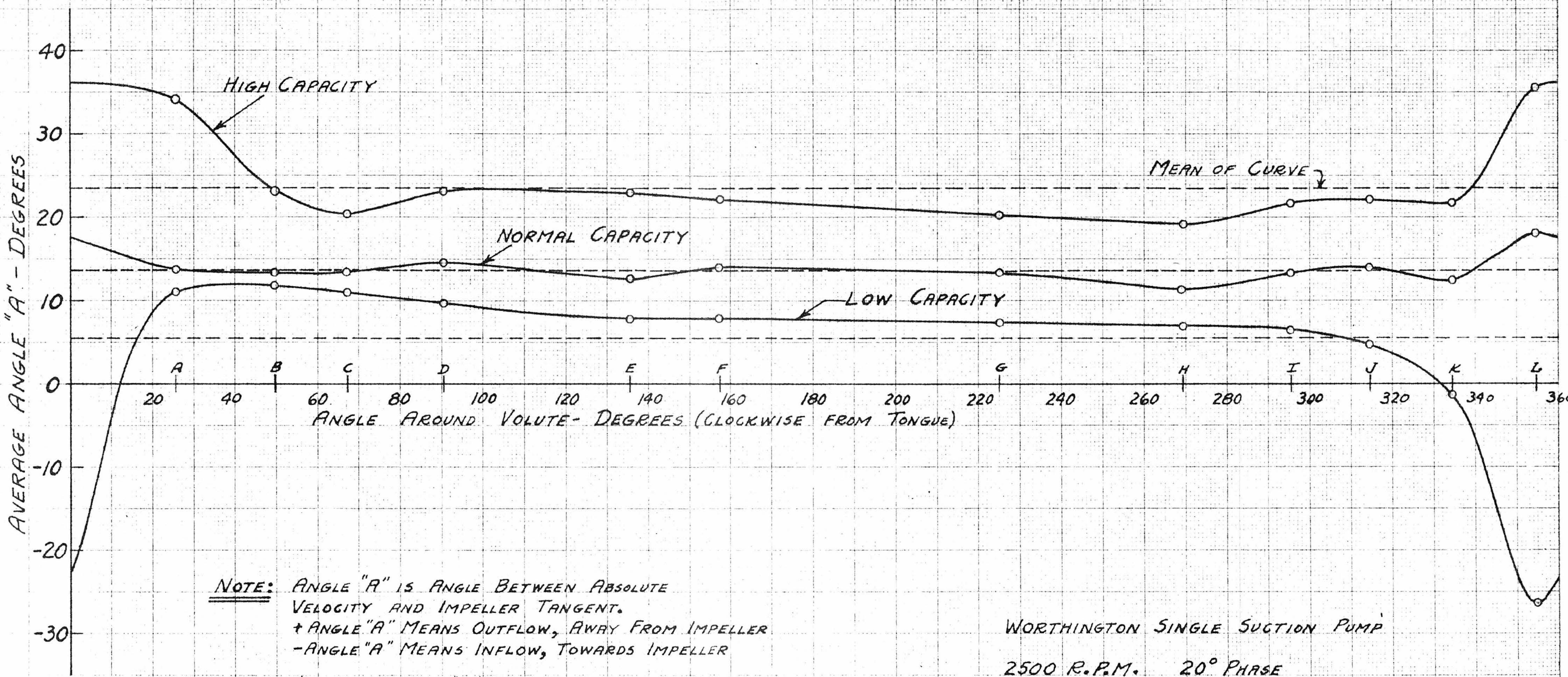
At normal pump capacity the average angle "A" is slightly non-uniform, while the direction is much more uniform than at high and low pump capacities.

Fig. 54 shows the average direction of the relative exit velocity vectors at the different Pitot stations. All points but one show a relative exit angle less than the vane exit angle, that is, the relative exit velocity is not following the vane tangent. Considering this matter of relative exit directions, there is a remarkable agreement between the observed values and the conclusions reached by Prof. R.L.Daugherty in his book,"Centrifugal Pumps". On page 81 of his book he states (arrived at by inductive reasoning) that the difference between the vane angle and the actual relative exit angle "...may differ by from five to ten degrees...". Fig. 54 shows that for normal capacity the difference in angles was nine degrees.

An unpublished calculation made by Prof. Daugherty showed a close agreement. Using his theory, he made calculations for a pump whose specific speed was very close to that of this Worthington pump. The calculated relative exit angles for low, normal, and high capacities agreed closely with the observed values shown on Fig. 54. The range of agreement was something like one degree. This agreement between theory and experiment speaks well for both entirely independent methods of attacking this problem of pump flow.

AVERAGE ANGLE BETWEEN ABSOLUTE VELOCITY AND IMPELLER TANGENT
AT
DIFFERENT PITOT STATIONS AROUND VOLUTE

FIG. 53



NOTE: ANGLE "A" IS ANGLE BETWEEN ABSOLUTE VELOCITY AND IMPELLER TANGENT.
+ ANGLE "A" MEANS OUTFLOW, AWAY FROM IMPELLER
- ANGLE "A" MEANS INFLOW, TOWARDS IMPELLER

AVERAGE ANGLE "A" IS A MEAN OF A CURVE OF MEASURED ANGLES ACROSS THE INNER WIDTH OF IMPELLER.

WORTHINGTON SINGLE SUCTION PUMP

2500 R.P.M. 20° PHASE

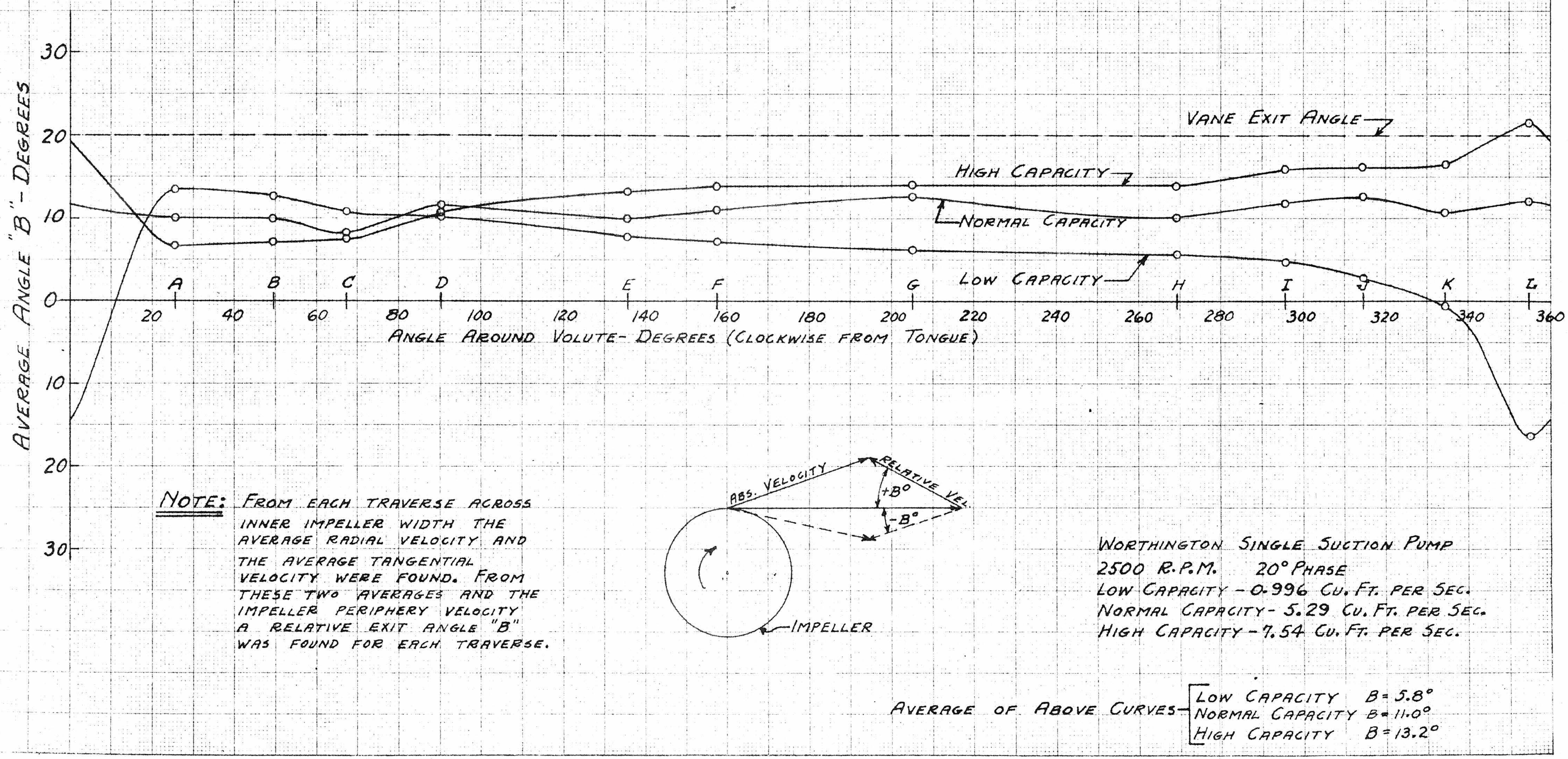
LOW CAPACITY - 0.996 CU. FT. PER SEC.

NORMAL CAPACITY - 5.29 CU. FT. PER SEC.

HIGH CAPACITY - 7.54 CU. FT. PER SEC.

DIRECTION OF AVERAGE RELATIVE EXIT VELOCITY AT DIFFERENT PITOT STATIONS AROUND VOLUTE

FIG. 54



BIBLIOGRAPHY

1. Kucharski, "Strömungen einer reibungsfreien Flüssigkeit bei Rotation fester Korper", Munich and Berlin, 1918.
2. Busemann, "Das Förderhohenverhältnis radialer Kreiselpumpen mit logarithmisch-spiraligen Schaufelen", Ztsch. f. angew. Math. u. Mech., Bd. 8, 1928, s. 372.
3. Schultz, "Das Förderhöhenverhältnis der Kreiselpumpen für die ideal und wirkliche Flüssigkeit", Forsch. Arb. Ing.-Wes. Ht. 307.
4. Sörenson, "Pontentialströmung durch rotierende Kreiselräder", Ztschr. f. angew. Math. u. Mech., Bd. 7, 1927, s. 89.
5. Uchimura and Kitô, "On potential flow of water through centrifugal impeller", J. of Fac. of Engineering, Tôkyô Imp. Univ., Vol. 19, No. 8, 1931.
6. Pfleiderer, "Die Kreiselpumpen", Berlin, 1924.
7. Spannhake, "Hydraulische Probleme", Metteilungen des Institute für Strömungungsmaschinen der Technischen Hochschule Karlsruhe, Heft 1, 1930.
8. Fischer and Thoma, A.S.M.E. Transactions, 1932.
9. R. L. Daugherty, "Centrifugal Pumps", 1915.
10. Francis, "Lowell Hydraulic Experiments", 1909.
11. Thoma, "Die Versuchsanstalt für Wasserturbinen in Gotha", 1918.
12. Oertli, "Untersuchung der Wasserströmung durch rotierendes Zellenkreiselrad", Zurich, Dessert., 1923.

BIBLIOGRAPHY Cont'd.

13. Stiess, "Über die Relative-Stromung in einem Pumpen-Laufrad von Grossem Radien-Verhältnis", Mitt. d. Inst. f. Stromungsm. d. Tech. Hochs. Karlsruhe, Ht. 3, 1933, s. 77.

14. Closterhalfen, "Hilfsmittel zur Beobachtung und Messung an umlaufenden Kreiselrädern", Forschung, Berlin, January, and February, 1931.

15. Yendo, "Experimental Researches of Turbine Pumps", Rep. of the Yokohama Tech. College, No. 1, 1930.

16. Kasai, "On the Exit Velocity and Slip Coefficient of Flow at the Outlet of the Centrifugal Pump Impeller, with Some References to the Characteristics of the Pump". Kyushu Imperial Univ. Memoirs, Japan, 1936.

17. R. C. Binder, M.S. Thesis, California Institute, 1933.

18. "Air Forces on Circular Cylinders", Dryden - Scientific Paper No. 394, Bureau of Standards, 1920.

19. Fechheimer, "Measurement of Static Pressure", A.S.M.E. Transactions, 1926.

20. E. R. Lockhart, M. S. Thesis, California Institute, 1934.

21. Th. von Kármán, "Über Laminare und Turbulente Reiburg", Z. angew. Math., Mech., vol. 1, 1921.

A. Closterhalfen, "Versuche an einer Schaukreiselpumpe" Forschung, Berlin, July 1931