Theoretical and Experimental Analysis of a
Regenerative Turbine Pump, the Sta-Rite H-7
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Abstract

Preliminary theoretical and experimental analysis was carried out on the Sta-Rite H-7 Regenerative Turbine Pump in order that a better understanding of its flow pattern and performance characteristics might be obtained. Dimensionless performance curves of head, power, and efficiency versus flow rate were obtained, and the characteristics of these curves were investigated. A theoretical analysis of the flow in the pump was worked out. In order to evaluate several unknown quantities used in the theory a series of flow angle measurements were made, and the streamline pattern was determined for a series of operating conditions.
Introduction

The regenerative turbine pump is unique in both design and operation, in that it combines many of the features of the rotary displacement class of pumps with some of the important characteristics found in centrifugal pump operation. In general, the regenerative turbine pump utilizes a rotating impeller with a set of radial teeth or vanes at the periphery to transport fluids, through an annulus chamber of some 300° from inlet to exhaust port, in a series of spiral flows by which the fluid is returned through the teeth repeatedly, producing the regenerative type of flow indicated by its name. With this regenerative flow pattern, the pump is able to maintain extremely large manometric heads at very low flow rates, thus making it ideal for use in boiler feed works, filtering and booster systems, and other hydraulic systems where the maintainance of large pressure differentials is a prime consideration.

Very little previous investigation has been reported on the regenerative turbine pump. The most complete analysis to date seems to be that reported by Yasutashi Senoo*. In Mr. Senoo's treatment of the pump, however, he carries out a two-dimensional analysis of the flow, assuming that viscous friction between the moving wall (periphery) of the impeller

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* Reports of the Research Institute of Fluid Engineering Kyushu University, Japan Vol. 5, No. 1 (1948)
and the fluid is the primary force causing the pumping action. His analysis is based on the analogy of the flow in the radial clearance space to that of two dimensional flow between a fixed and moving plate. This analysis may, in part, account for the flow action in the radial clearance space but appears to ignore the more fundamental spiral flow pattern maintained in the side and radial clearance chambers. Mr. Senoo's use of turbulent viscosity to explain the flow pattern did not seem to be compatible with the flow pattern visualized by the authors. Therefore, in addition to carrying out standard performance tests, we wished to investigate the flow by means of streamline probes. It was felt that if we could investigate the mean streamline angles and velocities at different flow rates for a series of points in the pump chamber, a much clearer foundation could be formed for added investigation of the flow and for theoretical predictions of performance. Since the back-flow loss in the chamber due to the pressure differential across the pump is quite severe, it was hoped that qualitative and quantitative investigation of the mean streamline forms might yield valuable information for use in making corrective chambering adaptations to get higher efficiencies.

On the basis of a spiral flow similar to the one described above we may use the angular momentum theorem to determine values of flow, head, and power with the use of empirical velocity coefficients to describe the flow. The
following theoretical analysis was developed in an attempt to further explain the flow:
Let: \( r_c \) = radius of outer clearance
\( r_c' \) = radius of inner clearance, \( c \) = impeller root radius of blade
\( r_1, r_2, r_3, r_4, r_5 \) = radii of mean streamline points as indicated in Fig.

Assume: \( r_1 = r_c' + \frac{b}{2} \) \( j \) \( r_2 = r_4 = \) radius of impeller tip
\( r_3 = \frac{r_c + r_2}{2} \) \( j \) \( r_5 = \frac{r_2 + r_1}{2} \)

Assume: \( C_{t1} = \alpha U_1 = \) absolute tang. velocity at rotor entrance where \( U_1 = \) wheel speed at \( r_1 \)
\( C_{t2} = \alpha U_2 = \) absolute tang. velocity leaving rotor where \( U_2 = \) wheel speed at \( r_2 \)

\( \frac{dp_1}{d\theta} = \frac{dp_2}{d\theta} = pressure \ differences \ between \ radial \ planes, \ containing \ the \ axis \ of \ rotation, \ are \ constant. \)

Angular Momentum between points \( 2 \) \& \( 4 \)

a) \( A_1 \frac{dP_1}{d\theta} r_3 = \int r_2 b C_{t2} (r_2 U_2 - r_2 C_{t4}) \)

Angular Momentum between points \( 4 \) \& \( 1 \)

b) \( A_2 \frac{dP_2}{d\theta} r_5 = \int r_2 b C_{t2} (r_2 C_{t4} - r_1 \alpha U_1) \)
but, from original assumption of pressure differences
\[
\frac{dp_1}{d\theta} = \frac{dp_2}{d\theta}
\]
or:
\[
\frac{r_2 u_2 - r_2 c t_4}{r_3 A_1} = \frac{r_2 c t_4 - r_1 \alpha u_1}{r_5 A_2}
\]
then:
\[
c t_4 = \frac{r_2 r_5 (u_2 A_2 + r_1 r_3 \alpha u_1 A_1)}{r_2 r_3 A_1 + r_2 r_3 A_2}
\]
Substituting equation c) into equation d)
\[
\frac{dp}{d\theta} = \frac{f r_2 b c r_2 (r_2 u_2 - r_1 \alpha u_1)}{r_3 A_1 + r_5 A_2}
\]
For consideration of the thru-flow in the pump:
\[
Q = c t_3 A_1 + c t_5 A_2
\]
Assuming:
\[
c t_3 = \frac{u_2 + c t_4}{2} \\
c t_5 = \frac{c t_4 + \alpha u_1}{2}
\]
\[A_0 = A_1 + A_2 = \text{available area for thru-flow with no net flow in } A_0 \text{ since the flow in } A_0 \text{ is carried through the exhaust stripper to inlet}
\]
then:
\[
2 Q = u_2 A_1 + \alpha u_1 A_2 + c t_4 A_0
\]
substituting equation c) into equation e)
\[
2 Q = u_2 A_1 + \alpha u_1 A_2 + \left(\frac{r_2 r_5 (u_2 A_2 + r_1 r_3 \alpha u_1 A_1)}{r_2 r_3 A_1 + r_2 r_3 A_2}\right) A_0
\]
Let:
\[K_1 = \frac{r_2 r_5 A_0}{r_2 r_3 A_1 + r_2 r_3 A_2} \\
K_2 = \frac{r_1 r_3 A_0}{r_2 r_3 A_1 + r_2 r_3 A_2}
\]
then:
\[2 Q = u_2 A_1 + \alpha u_1 A_2 + K_1 u_2 A_2 + K_2 \alpha u_1 A_1
\]
or:
\[
Q = \frac{u_2}{2} \left(A_1 + K_1 A_2\right) + \frac{\alpha u_1}{2} \left(A_2 + K_2 A_1\right)
\]
For the output power of flow:
\[
Q \frac{dp}{d\theta} = \left[\frac{u_2}{2} \left(A_1 + K_1 A_2\right) + \frac{\alpha u_1}{2} \left(A_2 + K_2 A_1\right)\right] \frac{f r_2 b c r_2 \left(r_2 u_2 - r_1 \alpha u_1\right)}{r_3 A_1 + r_5 A_2}
\]
or:
\[
Q \frac{dp}{d\theta} = \left(\frac{1}{2} f r_2 b c r_2\right) \left[\frac{u_2 \left(A_1 + K_1 A_2\right) + \alpha u_1 \left(A_2 + K_2 A_1\right)}{r_3 A_1 + r_5 A_2}\right] \frac{u_2 - r_1 \alpha u_1}{3 A_1 + 5 A_2}
\]
In order to arrive at the final expressions above, it was necessary to use several unknown quantities in the analysis: the absolute tangential velocity of the fluid as it enters the impeller, $\alpha\mu_1$; the radial velocity leaving the impeller, $C_r_2$; the absolute tangential velocity of the fluid as it leaves the impeller, $\omega U_2$; and the radii of the mean streamlines in the flow channel, $r_1$, $r_2$, and $r_3$. The two tangential velocities can be determined by measurements of the magnitude and direction of the flow at the points 1 and 2. The radii cannot be determined in such a straightforward manner, since their location depends on the flow pattern in the channel, and this requires rather extensive tests for its analysis. By determining the streamline patterns and by finding the velocity distribution throughout the channel the radii can be approximated, however. It should be possible to compute the velocity $C_r_2$ from a measurement of the absolute velocity of the streamlines leaving the impeller, knowing the streamline angles.
EXPERIMENTAL SETUP AND TESTING PROCEDURE

The first thing that had to be considered in setting up a test rig was whether to use air or water as the fluid in the pump. The pump is designed to handle either liquids or vapor, so the choice depended not on the ability of the pump to handle the fluid but on the relative ease with which the quantities desired could be measured, and the ease with which changes could be made in the geometry of the pump. Since the pump is designed primarily for pumping water it is sturdily built and requires, when using water, a large power input. This large power input would be a big advantage in measuring the torque if water were used for testing, since very good accuracy could be obtained with a relatively insensitive dynamometer due to the large torques involved and the greater stability of the flow due to the greater mass of the water. This, however, is the only real advantage of water over air. Air has the advantage of being readily available, and no facilities are required for disposing of it. The pressure developed in the pump is directly proportional to the density of the fluid used in it, so the head measurements for air would be about 830 times as small as those for water, and could be measured by means of standard manometer tubes. Flow measurements would not be a large problem with either fluid, since the flow rate is independent of the fluid used. The most convincing argument in favor of air, and the one which
finally overweighed the torque measuring difficulties, was the ease with which changes in the pump interior could be made with air as the fluid. Relatively weak structures could withstand the flow of air, but would not stand up in a water flow, so that by using air quick and easy changes in the pump geometry could be made. Probes for measuring the flow directions in the channel would also be easier to make for an air flow, and viewing plugs would be easier to assemble with the much lower pressures.

With the decision to use air as the test fluid, the test rig was set up. Two different groups of readings were required for purposes of analysing the pump, and facilities had to be made for measuring these. To get the performance characteristics of the pump, readings of speed, head, flow, and torque were needed. In order to check the correlation between the theoretical approach and the actual behavior of the flow some method was needed to measure the flow angles and streamline velocities in the pump channel, and to find their variation with changes in head, flow, and speed.

In the first group of readings the torque measurement was the most critical, due to the low torques which were involved using air. Preliminary estimates based on values from the Worthington Corporation performance curves for this pump, after conversion to air by means of the dimensionless parameter \( \frac{P}{\omega^3D^5} \), yielded shutoff values of 0.05 horsepower for 3600 rpm and 0.4 horsepower for 7200 rpm, which was originally set as the maximum test speed. Expressed as
torques, these are 14 in.-oz. for 3600 rpm and 56 in.-oz.
for 7200 rpm. These are values at shutoff, while at higher
flow rates for each speed the torques would be correspondingly
lower. Thus, in the design of a dynamometer to measure these
low torques, the prime consideration was to get the maximum
possible sensitivity. For a more detailed account of the
problems involved in this design and for a discussion of the
construction of the dynamometer, reference is made to the
thesis work of D. Lippman and T. Taylor on this subject.*
The final design is shown in assembly in Fig. 1, Appendix B.
This design proved to be very satisfactory in measuring the
low torques encountered during testing. In sensitivity tests
it was accurate to within .04 in.-oz., or less than 5 % of
the highest measured torque and less than 2 % of all of the
torques except those at 1200 rpm.

For flow measurements a venturi tube attached to the
intake pipe was used. A venturi was used so that maximum
pressure recovery could be obtained, and it was connected
to the intake pipe in order that the readings would not be
disturbed by perturbations in the flow due to passage through
the pump. The venturi tube used was calibrated on the intake
of a gasometer and was found to have a flow coefficient of
0.99.

Pressure was measured at intake, exhaust, and at four
pressure taps spaced at 90° intervals around the chamber,
starting at the intake port and with the last one 22.5° from

* M.E. Dept., M.I.T. Jan. 1953
the exhaust port. The arrangement of these pressure taps is shown in Figure 5, Appendix E. The pressures were measured on a manometer stand, with all of the pressures referred to the inlet pressure instead of atmospheric pressure. With the manometer stand used it was not possible to measure shutoff head at speeds above 4800 rpm, where the head was 30 inches of water.

For measuring the speed, a Strobotac was used to stop a point on the shaft connecting the dynamometer and the pump. Before each run the Strobotac was adjusted to coincide with the vibrations of the calibrating need, and was then checked by means of a hand tachometer held on the impeller end of the shaft with the pump running at the speed indicated by the Strobotac. During test runs, the pump housing end plate was taken off at 1800 rpm, 3600 rpm, and 4800 rpm and the tachometer was again used to make sure that the Strobotac was on the right speed. At low speeds the Strobotac was used on line frequency to get more accuracy, but at high speeds this was difficult to do because of the large number of harmonics in the strobe light frequency.

For measuring flow angles and streamline velocities some means of seeing into the chamber was needed. This was provided by two Plexiglas view plugs set into the pump casing 90° before the exhaust port. Through one of these plugs a radial view of the impeller can be obtained, while the other plug is set directly in front of a mirror positioned at 45° to
the line of sight in the side flow chamber, with a celluloid plate in the wall of the main flow channel allowing a view of the side of the rotor. The arrangement of these plugs is shown in Figure 4, Appendix B. Two holes (0.030") were drilled in the radially positioned view plug, one in the center and one near the edge of the plug, through which the directional and velocity probes could be inserted into the chamber. The directional probes were constructed using fine (.025) piano wire with a two-link section of small jeweler's chain soldered to the end as a swivel for the pointer, which was made of a starched thread about $\frac{1}{4}$ to $\frac{5}{16}$ long, with a wisp of yarn glued to the end as a stabilizer. The velocity probes were never used, but were to have been made of .020" seamless tubing, with the total pressure tube having an open end directed into the flow, and the static pressure tube being closed at the end, directed into the flow, and having a small hole drilled in its side to measure the static pressure. By using the pointers to get the direction of flow and then using the velocity probes to determine its magnitude it was hoped that some quantitative data could be gotten to check the theoretical approach outlined in the introduction to this thesis. Due to lack of time, however, only a qualitative survey could be made using the directional probes to measure flow angles, with no velocity measurements being taken.

Performance tests were made every 600 rpm, from 1200 rpm through 4800 rpm, at flow rates ranging from maximum flow to
shutoff for each speed. Five points were taken at each speed up through 3000 rpm, and from then on 6 points were taken at each speed. Runs were originally planned up to 7200 rpm, or twice the normal operating speed, but above 5000 rpm the impeller began to vibrate against the clearance surfaces in the pump casing, making operation at higher speeds impractical.

Before each run, the Strobotac was checked against a tachometer to make sure it was set correctly. The pump was then run up to speed, and the head and flow readings checked on the monometer board against previous approximate values to make sure the speed was correct, and not a multiple or harmonic of the speed indicated by the Strobotac. If there was any doubt, the tachometer was again used as a check. With the pump running at speed, the torque readings were taken, care being exercised to be sure there were no accelerations or decelerations of the motor which, during this period, would greatly affect the torque readings. When the torque readings were completed, the speed was held constant while the manometer readings were taken. Eight pressures were measured: pressure at inlet and exhaust of the pump; at each of the four pressure taps around the periphery of the pump; and throat pressure against atmospheric pressure in the U-tube readings for the venturi tube. Barometer readings were made at the beginning and end of each run, and air temperature readings were made at each speed.
For each speed the points measured were determined by varying the flow rate with a valve attached to the exhaust pipe, and the flow rates were approximately evenly spaced from full flow to shutoff.

The runs made to determine the flow angles were made at a different time from the performance tests and a different procedure was used during these runs. Tests were made at two speeds, 1800 rpm and 3600 rpm. In these runs the curve points were not determined by evenly spaced flow or head readings, but by the behavior of the pointer and the changes in angle which were observed while the flow rate was being changed. As far as was possible, the points were determined by evenly spaced angles, though in turbulent regions this was obviously impossible. At each point three readings were taken: $\alpha$, the angle between the pointer and the plane of the impeller, measured in the channel alongside the impeller; $\beta$, the angle the pointer made with the tangent to the impeller while in the same position as for the $\alpha$ measurements; and $\gamma$, the angle between the pointer and the plane of the impeller, measured in the radial clearance of the channel. The identification and positions of these angles are shown in Figure 4 in Appendix A. To move the probe to these positions, the wire was inserted in the off-center hole of the radially positioned plug, and the plug was rotated to give a traverse across the impeller and out to the side channel. To move the probe radially, the wire was merely pushed in or drawn out to the
desired position. The angles $\theta$ and $\phi$ were measured with a protractor placed over the face of the radially positioned plug and centered on the probe, with illumination coming from a light directed into the channel through the other plug by way of the mirror. For measuring $\beta$, the process was reversed, light coming in through the radial plug and the protractor being placed over the other plug to measure the angle by its image in the mirror. At most points there was considerable fluctuation of the pointer, and the angles could only be read to an accuracy of $\pm 5^\circ$. At flow rates and in positions where exceptional turbulence occurred, an attempt was made to get readings at the closest stable points. No torque readings were taken during the flow angle tests, but readings of head, flow, and pressure distribution were made at each point where angles were read.
Conclusions

The investigations, both theoretical and experimental, which are embodied in this paper are, for the most part, of a preliminary nature. Due to the time required for the construction and instrumentation of the test rig, a necessity for any experimental work, it was not possible to make as many tests as were desired, or to do as thorough a job as might be hoped for on the tests that were made. However, on the basis of the work done, both theoretical and experimental, a number of important conclusions can be made, and many questions have been raised in the course of the analysis and experimentation from which suggestions for future work may be drawn.

The determination of the performance curves for the pump, standardized procedure in the analysis of any machine of this type, was especially necessary in this case to determine whether air could suitably be used as the fluid medium in the proposed tests. The total head over the pump, the flow rate, and the power input were measured using the procedures described earlier, and the resultant values were converted to the dimensionless parameters $\frac{QH}{H_D^2}$, $\frac{Q}{Q_D^3}$, $\frac{P}{P_D^5}$, $+\gamma$ using the outer radius of the impeller, 5.393 inches, as the characteristic diameter of the pump. The plots of dimensionless head, dimensionless power, and efficiency versus dimensionless
flow are shown in Figures 1, 2, and 3 of Appendix A respectively. The small amount of scatter in these plots indicates that air is quite suitable for testing purposes, even for the very critical measurements of torque and efficiency. With air, all of the torque readings are very sensitive to external and frictional disturbances, especially such things as drag from the sealing disk and friction between the impeller and the clearance surfaces in the casing. By adjusting the seal and correctly positioning the impeller on the shaft these frictional forces can, however, be eliminated, or reduced to a point where they are no longer effective, and accurate readings can be made.

The head, flow, and power values from these tests, compared in dimensionless terms with the values of these same quantities taken from the Worthington Corporation Standard Centrifugal Pump Rating Curve for this type pump, gave close correlation, within \( \pm 10\% \) in most cases.

The performance curves for this pump illustrate the basic differences between the turbine-type pump and the ordinary centrifugal pump. In the turbine pump the head decreases with increasing flow, as in the centrifugal pump with backward-curved blades, but the power input also decreases with increased flow rate. This puts the maximum power input at shutoff, and the minimum power input at maximum flow. The advantage of this pump is its ability to give a very high head for its size, but the power required to do this is also
very high. Using a 15 horsepower motor to drive the pump, and pumping water, the maximum speed at shutoff would be about 2450 rpm, with a head of 550 feet of water. Extremely high heads can theoretically be obtained at higher speeds and correspondingly higher power input. This pump, for instance, would produce 2100 feet of water head at 4800 rpm shutoff, but would use 117 horsepower, based on the performance curves for air. The efficiency of the pump is very low at high heads, and even at lower heads and high flow the efficiency never exceeded 49.5%. In the plot of efficiency versus dimensionless flow it is not possible to determine the maximum efficiency of the pump because within the values plotted no maximum point is reached, although the curve seems to flatten out at the higher values of $\frac{Q}{\omega D^3}$. It is evident from this curve, though, that the efficiency never gets very high, probably not above 50%.

In the attempt to determine the flow pattern and corresponding velocities in the channel, measurements were made on the flow angles only, at positions alongside and radial to the impeller. From the partial flow pattern indicated by these points some useful qualitative conclusions can be drawn. The flow angles seem to be fairly insensitive to changes in speed and flow rate, but very sensitive to changes in head. The angle $\gamma$ in the radial channel is not very sensitive to changes in flow, speed, or head, but the angle $\beta$ is very sensitive to changes, especially changes in head. The angle $\alpha$ is an indication of the direction of flow of the
fluid in the channel either into or out of the impeller, and in all cases observed it was into the impeller, its pitch changing as $\beta$ changed. Assuming a spiral flow throughout the channel, the spiral compresses with higher head very much as a coil spring compresses under load. When running at a given speed, the spiral stays very much the same, with a slight steepening of the angles, as the flow rate is gradually cut down. As soon as the head begins to approach the shutoff value, though, the flow changes very rapidly. There is first a period when the pitch steepens until the flow is nearly radially inward, indicating that the spirals are being compressed. There is then a period of flow transition when all of the directional readings in the side channel are unstable, with the streamlines in the radial channel remaining stable and at about the same angle as before. Following this period of flow transition there is another stable period, with the flow in the side channel still going radially inward but now moving in a direction opposing the direction of rotation of the impeller. The flow in the radial channel is still in the direction of rotation, and the angles are steady except in the area nearest the side channel, where there is a mild turbulence. It is assumed that it is in this region that the spiral is being bent back and turned into a back flow due to the high head and low flow. The flow in the radial channel presumably is kept stable by the relatively small area of the passage and the inertia of the fluid being forced radially.
out of the impeller into this passage. In the side channel there is no such forced flow by the impeller and the flow is much more subject to the adverse pressure acting against it. The spiral flow, in effect, slips back along the channel until it passes into the impeller and is forced through the cycle again, thus bringing about the "regenerative" action which is assumed to be responsible for the high heads possible with this pump at and near shutoff. At shutoff the flow in the side channel and in the outer portions of the radial channel are very turbulent, and no readings can be taken for the flow angles, but the flow near the center of the radial passage is still stable.

The entire series of flow changes, from the beginning of the first unstable period to shutoff, all take place near shutoff, at 3600 rpm occurring in the interval from 90 % of shutoff head to complete shutoff. At lower speeds the interval is longer and occupies a larger percentage of the total head ; at 1800 rpm the first unstable transition point is much less noticeable than at 3600 rpm, with the flow changing more slowly from forward to backward.

As far as could be determined from the directions of the streamlines in the side passage, the flow passes into the side of the impeller all along the length of the blades, with the only flow out of the impeller occurring at the end of the blades, where it passes outward into the radial flow passage. The flow in this passage is in the plane of the
impeller at the middle of the channel, and angles out sharply a slight distance on either side of the center into the familiar spiral flow pattern.

From what has already been determined and assumed about the flow patterns and performance characteristics of the pump, several methods of further investigation can be outlined. The first thing that should be done is to get some measurements of velocities in the channel area, using the method described in the section on testing procedure. Using these velocity values, some check can be made on the theoretical approach to analysis of the pump. The behavior of the flow and its effect on the performance of the pump can be further analysed by putting several different types of shroud rings on the sides of the impeller and observing the change in the flow pattern due to this obstruction to the spiral flow. Since fluid passes into the impeller all along the blade length, a shroud would confine the flow to a definite path and should affect the performance as well as the flow pattern.

The variation of streamline angles and velocities should be further investigated for the conditions of constant head and flow at different speeds, different heads and flows at constant speeds, different heads at constant flow, and different flow rates at constant head, in order to determine more exactly which conditions most influence the flow pattern.
From measurements made on the pressure taps spaced around the pump casing it was determined that, except for the disturbed area in the region of the inlet port, the pressure distribution along the flow channel was linear. This is shown in Figure 11, Appendix A, which is a plot of the pressures around the periphery, referred to the inlet pressure, versus the angle of traverse around the periphery. For a more complete concept of the flow behavior the variation in the spiral with the gradually increasing pressure around the periphery would be useful as a possible indication of the relationship between the number of regenerative cycles and the head produced.

Some method should be found to get a higher flow rate at a given speed in order to complete the efficiency versus dimensionless flow curve and determine the maximum efficiency for the pump.

The constants $K_1$ and $K_2$ used in the theoretical analysis of the pump should serve as a guide for any geometrical changes to be made in the pump, as they give the inter-relationship of the dimensions of the pump which affect the performance and flow characteristics.

Altogether, there are a great many problems involved in the further analysis of this pump, and it is hoped that the work described in this thesis will serve as a help for any future research on this subject.
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APPENDIX A

Dimensionless Performance Curves

Characteristic Flow Patterns

Streamline Angle vs. Dimensionless Parameter Curves

Static Chamber Pressure Distribution
FIG. 1
DIMENSIONLESS PLOT OF HEAD VS. FLOW RATE

- 4800 RPM.
- 4200 RPM.
- 3600 RPM.
- 3000 RPM.
- 2400 RPM.
- 1800 RPM.
- 1200 RPM.
FIG. 2
DIMENSIONLESS PLOT OF
POWER VS. FLOW RATE

- 4800 RPM
- 4200 RPM
- 3600 RPM
- 3000 RPM
- 2400 RPM
- 1800 RPM
- 1200 RPM
FIG. 3

DIMENSIONLESS PLOT OF EFFICIENCY VS. FLOW RATE

- 4800 R.P.M.
- 4200 R.P.M.
- 3600 R.P.M.
- 3000 R.P.M.
- 2400 R.P.M.
- 1800 R.P.M.
- 1200 R.P.M.
FIG. 4

POSITIONS OF DIRECTIONAL PROBES USED TO MEASURE FLOW ANGLES
FIG 5: STREAMLINE ANGLE VS. DIMENSIONLESS FLOW
1800 R.P.M.

FIG 6: STREAMLINE ANGLE VS. DIMENSIONLESS HEAD
1800 R.P.M.
FIG. 7

FLOW PATTERN
HIGH FLOW, LOW HEAD
3600 RPM \( \frac{Q}{\omega D^3} = 4.40 \) \( \frac{H}{\omega D^3} = 8.56 \)

FIG. 8

FLOW PATTERN
LOW FLOW, HIGH HEAD
3600 RPM \( \frac{Q}{\omega D^3} = 1.23 \) \( \frac{H}{\omega D^3} = 12.93 \)
FIG. 9: STREAMLINE ANGLE VS. DIMENSIONLESS FLOW
3600 RPM

FIG. 10: STREAMLINE ANGLE VS. DIMENSIONLESS HEAD
3600 RPM
Figure 11: Static Pressure Distribution in Pump Chamber
APPENDIX B

Test Rig Assembly Drawing
Instrumentation Layout
Section of Main Pump Casing
SECTION A-A

M.E. DEPT. M.I.T.
WORTHINGTON PUMP DYNAMOMETER
T. TAYLOR - O. LIPMAN
JANUARY 19, 1953
MAIN PUMP CASING
APPENDIX C

Photos of Test Rig and Pump
EXPLANATION OF PHOTOGRAPHS

Fig. 1 Test rig, showing dynamometer, pump and pressure leads.

Fig. 2 Test rig, showing Strobotac, Variac speed control, venturi tube on inlet pipe, and manometer stand.

Fig. 3 Close-up of pump with end-plate removed, showing impeller in position.

Fig. 4 Close-up of pump with end-plate and impeller removed, showing interior openings of view plugs.

Fig. 5 External shot of view plugs, showing drilled holes for probe insertion.

Fig. 6 Viewing holes with plugs removed, showing radial and side views of impeller.

Fig. 7 Cooling device used for motor.

Fig. 8 Close-up of manometer stand during run at 4800 rpm.
   (1) Exhaust pressure
   (2)-(5) Peripheral pressure taps
   (6) Inlet pressure
   (7) Throat pressure side of venturi U-tube
   (8) Atmospheric pressure side of venturi U-tube
FIGURE 1