CAVITATION PHENOMENA

AND

FLOW IN DIVERGING TUBES

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Flow in Diverging Tubes.

by

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CONTENTS

Preliminary.

The first part of this thesis is devoted to a general description and a discussion of the results of a series of experiments carried out by the author and Mr. L. O. Cooper in the Hydraulic Laboratories of McGill University.

When a new hydraulic turbine or one of unusual proportions is being developed, the present day practice is to build a model of the turbine for operation in a testing flume, and to use the results from this to design the full-sized unit. It has been noticed that tests performed on large turbines built homologous with existing models generally show higher efficiencies than those obtained with the models.

The original intention of the investigation was to test a series of precisely similar diverging tubes in order to determine the effect of the draft tube on the efficiency of a model turbine and its prototype. However, after testing two such similar tubes and finding no difference in performance, it was deemed unnecessary to continue the investigation further along these lines. The remainder of the available time was devoted to an investigation of the performance of one of the diverging tubes which was modified, as will be described later, to allow the water to flow between two parallel surfaces before being discharged. This, as will be seen later, corresponds in principle to flow in certain types of draft tubes.

The second part contains a general resumé of the available literature pertaining to the cavitation problem as met with in engineering practice.

Acknowledgment.

The author wishes to acknowledge his indebtedness to Professor Ernest Brown, of the Department of Applied Mechanics and Hydraulics at McGill University, whose valuable suggestions throughout the experimental work always proved helpful; and to thank Mr. S. D. McNab, Superintendent of the Strength of Materials Laboratory at McGill University, for giving the author the benefit of his years of experience in testing work of various kinds.

PART ONE

Earlier Experiments.

A common form of diverging tube used for a variety of purposes, frequently to increase the discharge, is one consisting of a converging part or nozzle which in turn discharges into a diverging tube or cone, the outlet of which is usually submerged. The principle underlying the action of this tube is by no means new, having been known to the ancient Romans who sometimes applied diverging tubes to the public pipe lines, thereby drawing off more water than they paid for. Again, in New York City during the early part of the 19th Century, this fraudulent artifice is said to have been in practice when the price charged for water was in proportion to the area of the delivery pipe.

Diverging tubes are employed whenever the minimizing of the lost head due to contraction or expansion of a stream of fluid is an advantage, a few applications being such hydraulic devices as the Venturi meter, jet pumps, the expansion or reduction from one pipe to another of different size and the draft tube of a hydraulic turbine.

Near the end of the 18th Century Venturi performed experiments on diverging tubes and found that the rates of discharge varied with the angle of divergence, the best angle (angle of cone) being about 4.5°. Again, in 1875, James Brownlee[1] made tests of nozzles with diverging outlets in order to determine the action of the water and frictional resistance or loss of energy when flowing at various velocities. In 1917, Mr. Fred B. Seely[2] carried out experiments at the Experimental Engineering Station of the University of Illinois to obtain data on the effect on the discharge coefficient of placing a converging inlet and a diverging mouthpiece on a straight pipe, and also to determine the variation in the loss of head with different angles of divergence in the mouthpieces.

Although draft tubes have been used in water power plants for some time, it is only within recent years that any attempt has been made to secure an efficient design. The draft tube enables the kinetic energy or velocity head, as

[1] See Reference 1, Bibliography.
[2] See Reference 2, Bibliography.

it is called, of the water discharged from the runner of a turbine to be utilized. Inefficient tubes used in connection with the old type of low specific speed runner did not seriously reduce the overall efficiency of the turbine, as, with this type of runner, the energy to be regained in the tube is comparatively small. The low head power developments which are now being considered have made it necessary to investigate a method of securing all the draft head possible. Since a low head plant requires a runner of high specific speed, and therefore a high exit velocity from the runner, it is evident that, for economical operation, a draft tube of high efficiency is required. This necessity resulted in the development of several new types of draft tubes designed on basic hydraulic principles. In general, draft tubes for hydraulic turbines have a variety of shapes, both straight and curved. The straight draft tube may have its sides forming the frustum of a cone or it may be trumpet-shaped. Two recently developed types are the Moody spreading draft tube and the White hydraucone regainer. The Moody tube, which is trumpet-shaped, is usually, although not always, provided with a central cone. The hydraucone regainer, which may be either straight-diverging or trumpet-shaped, provides for the outlet flow of water to take place.before discharge between two parallel or approximately parallel surfaces. The sketches in Fig. 1 show the general shapes of a few of the more common types of draft tubes.

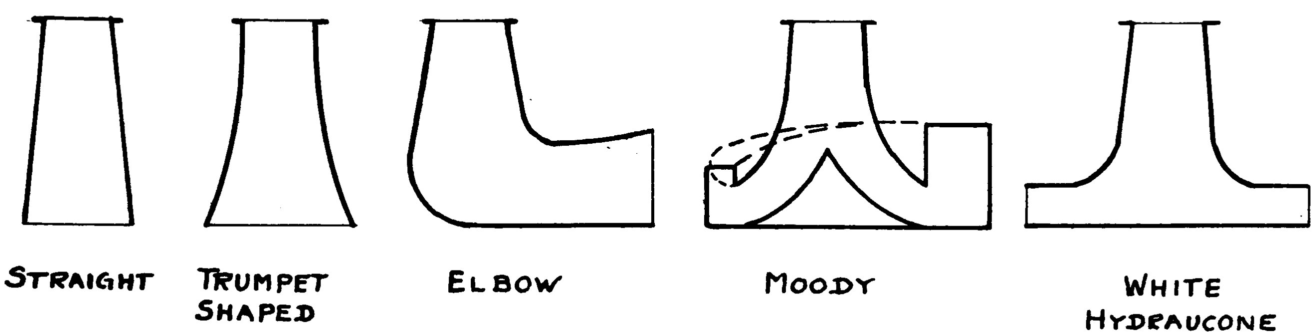

Fig. 1

Many tests have been made on draft tubes without runners and the results obtained, although not applying except perhaps in a relative sense when the tube is operating in connection with a water-wheel, are of great value to the designer. A very slight change, however, in the conditions under which the flow of water in the tube takes place, may cause, in some cases, a large difference in the performance of the tube. Therefore there is always danger in extending the use of experimental data to apply to conditions of flow different from those under which the data were obtained. Thus it is only by the careful correlation of experimental data, obtained under various conditions, that any rules or code of draft tube design may be established.

In 1921 Mr. George E. Lyon[1] investigated the flow in several draft tubes in order to determine the distribution of velocity at different cross sections. A

[1] See Reference 3, Bibliography.

Pitot-tube, introduced at different points in its length through the side of the draft tube being tested, was employed to measure the velocity across the section. The tests indicated that the velocity at the throat of the tube before entering the flare was very nearly uniform, falling off considerably at the walls, but that further down the tube the uniform velocity which persisted at the throat disappeared, the curve of velocity becoming peaked at the centre. It was found that this peak persisted down the tube flattening out somewhat at the mouth.

In the same year Mr. W. M. White[1] experimented on several types of draft tubes in order to determine their relative efficiencies when used in connection with a hydraulic turbine. In the experiments, model draft tubes were tested with the same runner and setting. Coincident with the model tests, experiments were made without the use of a runner on many smaller draft tubes to determine their performances before building larger models. Included among the smaller model tests were many on the type of draft tube developed by Mr. White called the hydraucone regainer, the principal variable factors in the experiments being the distance between the outlet plates and the radius of the curved section of the regainer.

Tests on small model draft tubes were carried out recently by Professor A. H. Gibson[2] and Mr. Schofield Labrow,

[1] See Reference 4, Bibliography.

[2] See Reference 6, Bibliography.

the experiments being performed without a runner. The object was to determine the efficiencies of the different types of tubes when used to regain kinetic energy. Experiments were made with straight conical tubes, trumpet-shaped tubes and short elbow tubes and tests similar to those of Mr. White were performed on the hydraucone regainer.

Fundamental Formulae.

In engineering practice the flow of water encountered is usually of a turbulent nature. Bernoulli's theorem or the general equation of energy used so often in hydraulics applies only when the particles of water move with uniform stream-line motion. Although this condition of flow seldom occurs, satisfactory analysis may often be made by using an average velocity and introducing empirical constants. In the following analysis stream-line motion is assumed to take place in a perfect fluid, that is, a fluid incompressible and devoid of internal friction. The assumption is also made that all the particles in a cross-section of the fluid have the same velocity, the cross-section being taken normal to the direction of flow. Although these assumptions correspond to an ideal type of flow, experience has shown that formulae based on them give results remarkably close to actual conditions obtained, thus justifying their use. The following nomenclature will be used in deriving the fundamental formulae:

Q = Discharge (cu.ft. per sec.)

a = Area of stream section considered (sq. ft.)

v = Mean velocity in the stream section (ft. per sec.)

P = Mean pressure at any point in the section considered (lbs per sq. ft.)

ω = Specific volume of the fluid considered (lbs. per cu. ft.)

g = The gravity constant

Bernoulli's Theorem.

Consider an infinitesimal stream or tube in a moving body of water shown in Fig. 2.

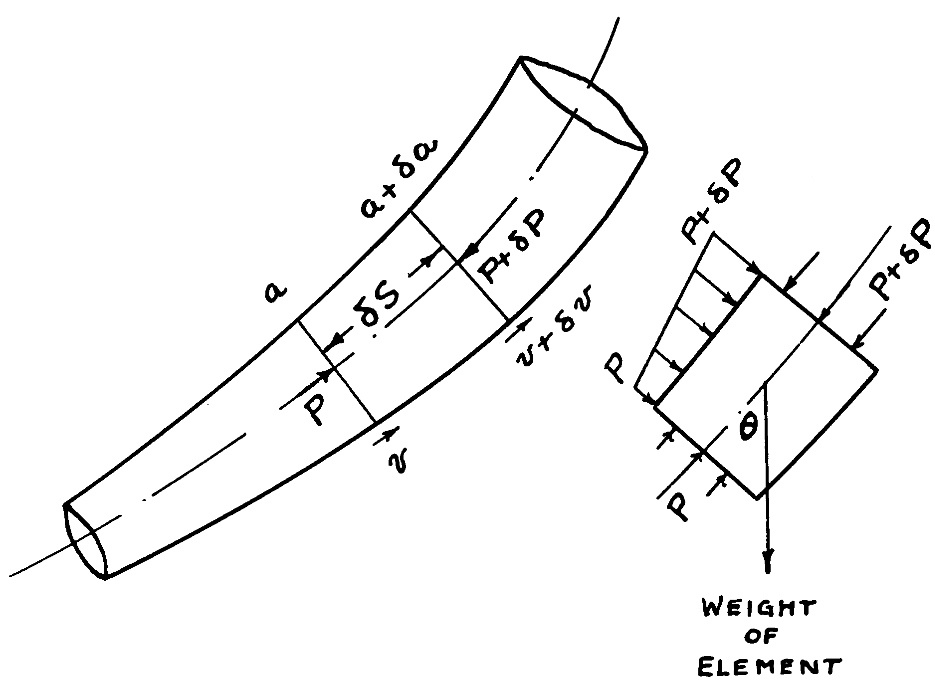

Fig. 2

Suppose that in an element of the stream of length δS ft., a receives an increment δa , v an increment δv and P a corresponding increment δP. Examining the forces on this element it is seen that the pressure on the curved surface varies from P at one end to $P+\delta P$ at the other. The average pressure on this surface may be written as $P+K\delta P$ where K is less than unity.

The net force along the axis of the element will be

$$Pa-(P+\delta P)(a+\delta a)+(P+K\delta P)\delta a-\omega\left(a+\frac{\delta a}{2}\right)\delta Z$$

where $\delta Z=\delta S\cos\theta=$ difference in elevation of the ends of the element measured in ft.

From the laws of mechanics it is known that

Force = Mass x Acceleration

Therefore,

$$Pa-(P+\delta P)(a+\delta a)+(P+K\delta P)\delta a-\omega\left(a+\frac{\delta a}{2}\right)\delta Z=\frac{\omega\left(a+\frac{\delta a}{2}\right)}{g}\delta S\,\frac{\delta v}{\delta t}$$

$\frac{\delta v}{\delta t}$ being the acceleration or the rate of change of the velocity with respect to the time.

By expanding and neglecting the second order of the small quantities this equation reduces to

$$-a\delta P-\omega a\,\delta Z=\frac{\omega a}{g}\delta S\frac{\delta v}{\delta t}$$

Therefore $\frac{\delta P}{\omega}+\delta Z+\frac{1}{g}\delta S\frac{\delta v}{\delta S}\cdot\frac{\delta S}{\delta t}=0$ as $v=\frac{\delta S}{\delta t}$

whence by integration

$$\frac{P}{\omega}+Z+\frac{v^2}{2g}=\text{a constant}$$

Thus the total energy possessed by a particle of water is the sum of the energies due to its pressure, velocity and elevation and, neglecting losses, this total energy remains the same regardless of the movement of the particle. This statement is known as the general energy equation or

Bernoulli's theorem.

By applying Bernoulli's theorem it is seen that a fluid discharging from an orifice, under a head h , has a theoretical velocity v equal to the velocity acquired by a body falling freely in vacuo through a vertical distance h . This statement is known as Torricelli's law and the expression $\frac{v^2}{2g}$ is termed the "velocity head".

Sudden Enlargement Losses.

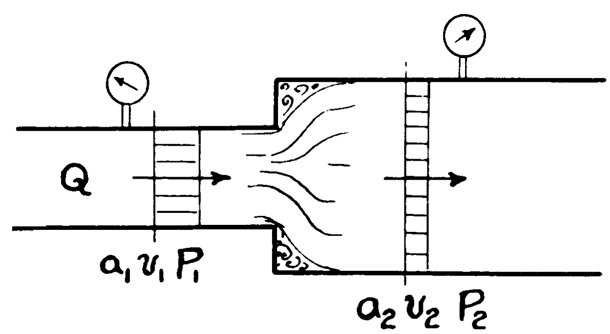

Fig. 3

Suppose a quantity of fluid Q to be flowing through a pipe which is suddenly joined to a larger one as in Fig. 3. Denote the velocity, pressure and area at the section just before the enlargement by the suffix 1 and just after by the suffix 2. An application of Bernoulli's theorem and Newton's second law of motion shows that the loss at sudden enlargement equals $\frac{(v_1 - v_2)^2}{2g}$ ft.

Efficiency of Diverging Tubes.

The efficiency of a diverging tube may be considered from two viewpoints:

(a) the viewpoint of the actual loss in the tube in relation to the loss at sudden enlargement.

(b) the viewpoint of conversion of kinetic energy into pressure energy, usually referred to as regained energy.

The efficiency based on the former will be referred to as "the efficiency" and on the latter as "the efficiency of kinetic energy regain" or, briefly, as "the efficiency of regain". The fundamental theory of each method of computing the efficiency will now be considered.

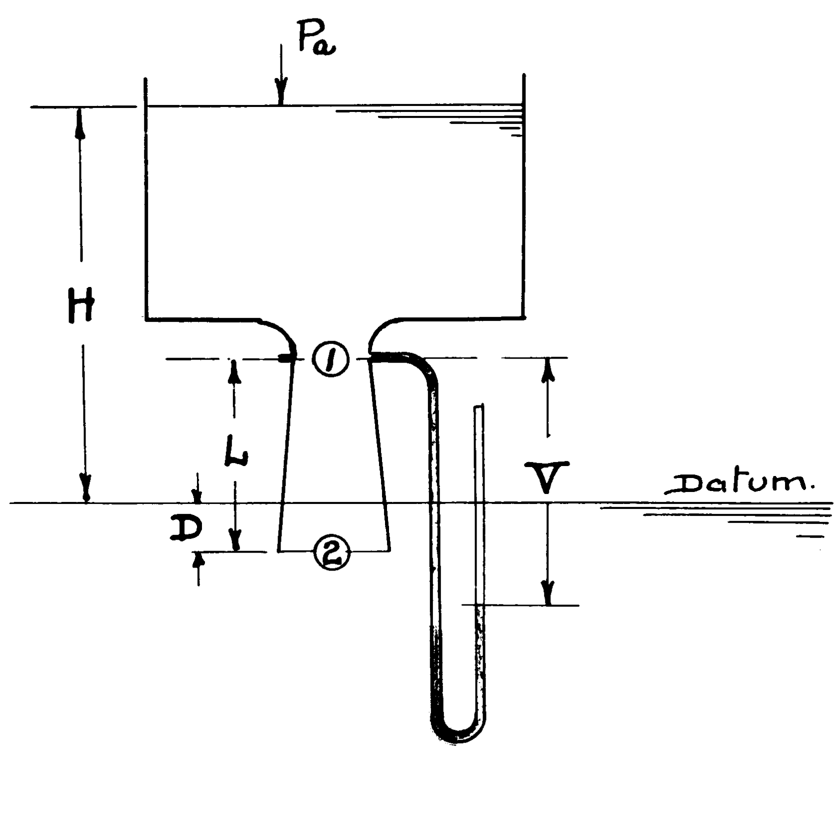

Fig. 4

Fig. 4 shows diagramatically a diverging tube of length L discharging under a head H , the end of the tube being submerged a distance D. Let V be the throat vacuum measured in ft. of water and let H, L and D be measured in ft. Denote the velocity, pressure and area, at the throat by suffix 1 and at the outlet by suffix 2.

The Efficiency.

If the throat of the diverging tube enlarged suddenly to the outlet diameter, the loss by shock per lb. of water would be $\frac{(v_1 - v_2)^2}{2g}$ ft. As, however, the enlargement is gradual, the loss will not be so great. If l ft. is the actual loss in the diverging tube, a measure of the efficiency will be

$$\frac{\frac{(v_1 - v_2)^2}{2g} - l}{\frac{(v_1 - v_2)^2}{2g}} \quad \text{or} \quad 1 - \frac{l}{\frac{(v_1 - v_2)^2}{2g}}$$

The actual loss in the tube is the difference between the total energy at the throat and the total energy at the outlet.

The total energy at the throat per lb. of water

$$= \frac{P_1}{w} + \frac{v_1^2}{2g} + L - D = \left(\frac{Pb}{w} - V\right) + \frac{v_1^2}{2g} + L - D \quad \text{as} \quad \frac{P_1}{w} = \frac{Pb}{w} - V$$

where Pb is atmospheric pressure in lbs. per sq. ft.

The total energy at the outlet per lb. of water

$$= \frac{P_2}{w} + \frac{v_2^2}{2g} - D = \left(\frac{Pb}{w} - D\right) + \frac{v_2^2}{2g} - D \quad \text{as} \quad \frac{P_2}{w} = \frac{Pb}{w} + D$$

Therefore

$$l = \left[\left(\frac{Pb}{w} - V\right) + \frac{v_1^2}{2g} + L - D\right] - \left[\left(\frac{Pb}{w} + D\right) + \frac{v_2^2}{2g} - D\right]$$

$$= \frac{v_1^2 - v_2^2}{2g} - (V + D - L)$$

which when substituted in the efficiency equation gives the result,

$$\text{Efficiency} = 1 - \frac{\frac{v_1^2 - v_2^2}{2g} - (V+D-L)}{\frac{(v_1 - v_2)^2}{2g}}$$ and as $Q = a_1 v_1 = a_2 v_2$

Therefore $\frac{v_1}{v_2} = \frac{a_2}{a_1} = R$, say.

$$= 1 - \frac{\frac{v_2^2}{2g}(R^2-1) - (V+D-L)}{\frac{v_2^2}{2g}(R-1)^2}$$

$$= 1 - \frac{R+1}{R-1} - \frac{2g(V+D-L)}{v_2^2(R-1)}$$

Efficiency of Kinetic Energy Regain.

When the flow through the diverging tube in Fig. 4 is steady, the total loss per lb. of water, including that of the nozzle entrance, is H. The kinetic energy per lb. of water in the throat section is $\frac{v_1^2}{2g}$.

Therefore, a measure of the kinetic energy efficiency will be

$$\frac{\frac{v_1^2}{2g} - H}{\frac{v_1^2}{2g}} \quad \text{or} \quad 1 - \frac{2gH}{v_1^2} \qquad \ldots\ldots\ldots\ldots\ldots(1)$$

Theoretically $v_2 = \sqrt{2gH}$, but, as there are losses in the tube due to friction and eddies, $v_2 = C_2\sqrt{2gH}$, where C_2 is an empirical constant. Substituting $v_1\frac{a_2}{a_1}$ for v_2 gives the result,

$$v_1 = \frac{a_1}{a_2}C_2\sqrt{2gH} = C_t\sqrt{2gH}$$

C_t is called the throat discharge coefficient and may be determined accurately by measuring the quantity of water discharged in a given time.

Substituting $C_t\sqrt{2gH}$ for v_1 in equation (1) gives the result,

Efficiency of Regain $= 1 - \dfrac{2gH}{C_t^2 2gH} = 1 - \dfrac{1}{C_t^2}$

Reference will be made in the following pages to the "volumetric efficiency". This quantity is the ratio $\dfrac{A\sqrt{2gH}}{Q}$ where A is the outlet area of the diverging tube measured in sq. ft.

Apparatus.

The object of the first series of experiments was to determine the discharge coefficients of the two similar diverging tubes. This necessitated the accurate measurement of the head on the tube and the quantity of water discharged in a given time. The second series of tests were carried out on the smaller tube modified by the addition of a bell-mouth and circular plate as shown in Plate 2. Discharge in this series took place between the flat surface of the bell and a plate held parallel to it. This apparatus will be referred to henceforth as the radial regainer. In the first experiment of this series a 6 in. diameter regainer was used, the variable factor being the distance between the parallel surfaces. In the second experiment the diameter of the regainer was varied, the distance between the parallel surfaces being kept constant. The object of the latter tests was to determine the effect of varying the two factors on the efficiency and discharge coefficient of the tube. Measurements were taken of the head and discharge as before and also the vacuum in the throat of the tube.

The general dimensions and details of the testing apparatus are shown in Plate 1. The photograph of the entire apparatus (Plate 3) gives a clear idea of the layout. The diverging tube being tested was set in the bottom of a water-tight wooden box supported on nine posts in a metal flume. The head causing the flow through the tube is the difference in levels of the water in the box and the flume.

The two similar tubes used in the first experiment were cast in bronze and then turned accurately to the required dimensions. Similar entrance nozzles were provided for each tube. The nozzles were turned from pine-wood which, in order to prevent warping, had been boiled in paraffin wax for several hours. The details of the tubes and nozzles are shown in the sketch (Plate 2) and in the photographs (Plates 4 and 5). The two bell-mouthed outlets, used to modify the smaller tube, were turned from wood which had been treated as described before. Three circular plates of diameters 6, 8 and 9.95 in. were used in conjunction with the outlets, the 8 in. and 9.95 in. plates both being used with the larger outlet. Each plate, which was carefully flattened so as to present a plane surface, was held at the outlet of the bell-mouth by three bolts equally spaced around the circumference. The distance between the bell-mouth and plate was maintained by spacers which were accurately machined to the required length. The assembly and details of the bell-mouths, plates and tube are shown in Plate 2 and in the photographs (Plates 4 and 5). Henceforth, throughout the thesis, the diverging tubes will be designated by their respective lengths, namely, 5 in. and 10 in.

The method of setting the diverging tubes in the bottom of the box is shown clearly in the sketch of the apparatus (Plate 1). The seat for the tubes was constructed so that the nozzle would fit flush with the inside bottom of the box, thus presenting an unobstructed intake to the tube. Tallow was used to fill the cracks between the nozzle and the box.

The water was supplied through a $2\frac{1}{2}$ in. pipe line from the city main, the quantity being regulated by a valve provided with a 1 in. by-pass for close adjustment. In order to avoid undue turbulence in the box a diffuser was placed at the end of the $2\frac{1}{2}$ in. line. A cylindrical sleeve which could be raised or lowered was used to cut off the discharge from the holes of the diffuser which were not submerged. The details of the diffuser are shown in the photograph (Plate 6) and in the sketch (Plate 1).

The level of the water in the flume was adjusted by raising or lowering the crest of a rectangular weir and by means of a regulating valve in a 2 in. pipe, the latter making possible a finer adjustment in maintaining a constant level. Both the weir and the pipe discharged into the same receptacle as is shown in Plate 1 and in the photograph (plate 3). The quantity of water discharged from the diverging tube passed out of the flume into a swivel receptacle or spout by means of which it was diverted as required to a measuring-tank, a weighing-tank or to waste. If a change of level occurred in the flume during a test, a correction was made to the quantity of water measured. A change in level of .01 in. necessitated a correction of 12 lbs. While testing the 5 in. tube a weir 5 in. wide was used. With the 10 in. tube it was found necessary to use a weir 20 in. wide in order to handle the greater discharge.

The discharge from the 10 in. tube was directed to a large measuring-tank fitted with a sensitive float gauge which was accurately calibrated to read the quantity directly to one cu.ft. and by estimating to one-half a cu. ft. As the average discharge during a test was approximately 20,000 lbs., the accuracy of the measurement was one-sixth of one percent. With the 5 in. tube the discharge was diverted to a weighing-tank where it was weighed accurately to 5 lbs. As the quantity in this case was approximately 4,000 lbs. the accuracy of the measurement was one-eighth of one percent.

A vertical gauge-glass was connected to a pipe extending into the box about 3 in. from the bottom. The level of the water in the glass was indicated on a fixed scale by a pointer which could be adjusted to the bottom of the meniscus. A sensitive float gauge indicated the level of the water in the flume. By means of a zero reading the two levels were connected, as will be explained in the next paragraph, giving the total head on the diverging tube accurately to 0.01 in.

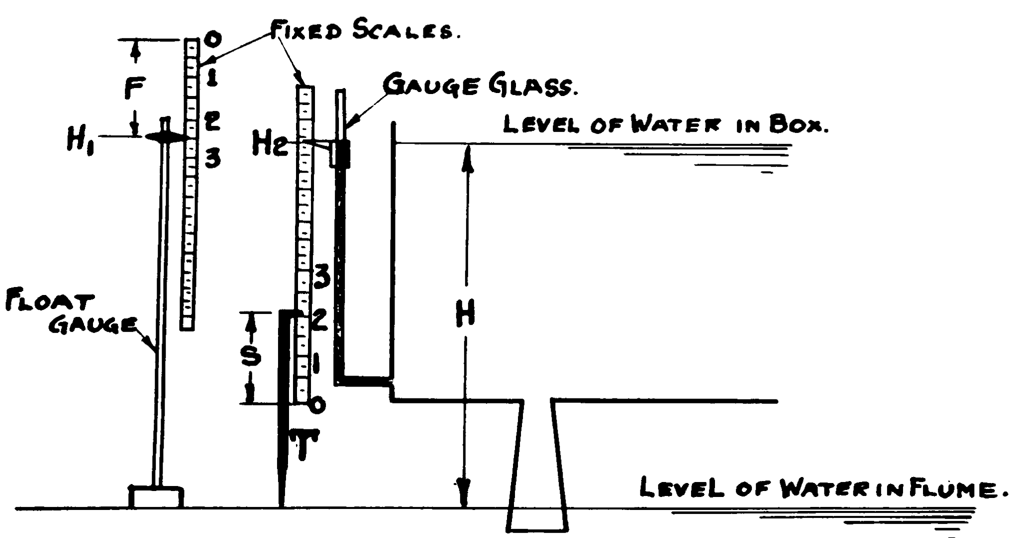

Fig. 5

Fig. 5 is a diagramatic sketch of the apparatus shown in Plate 1 and in the photograph (Plate 6) which was used to measure the total head. Note that the scale on the float gauge reads down and that the scale parallel to the gauge-glass reads up. Before taking a zero reading, the box was emptied and both the outlets of the flume were closed. When the water in the flume was quiet a pointed trammel T held perpendicularly beside the scale of the gauge-glass by a bracket was lowered until the end just touched the surface of the water in the flume. The position of the other end on the scale was then read. At the same time the float gauge reading was taken.

Let S and F be the simultaneous readings taken, respectively, with the trammel and float gauge and let T be the length of the trammel.

If H_1 and H_2 are the averages of the readings taken during a test of the float gauge and of the water level in the gauge-glass, respectively, and, if H represents the total head on the tube, it is seen from the diagram that

$$H = H_1 - F + H_2 - S + T$$
$$= H_1 + H_2 + (T - S - F)$$

The quantity $(T-S-F)$ is the zero reading.

In some of the tests a vortex tended to form above the diverging tube. This formation was effectively eliminated by floating a small block of wood on the surface of the water above the tube.

The apparatus used to measure the vacuum in the throat of the diverging tube is shown in Plate 1. It consisted essentially of a $\frac{5}{32}$ in. dia. pitot-tube $2\frac{1}{2}$ ft. long connected by a rubber tube to one end of a mercury U-tube. A three way stop-cock by means of which the pitot-tube could be opened to atmosphere was placed between the two. The glass tube of the pitot-tube ended in a brass tip of the

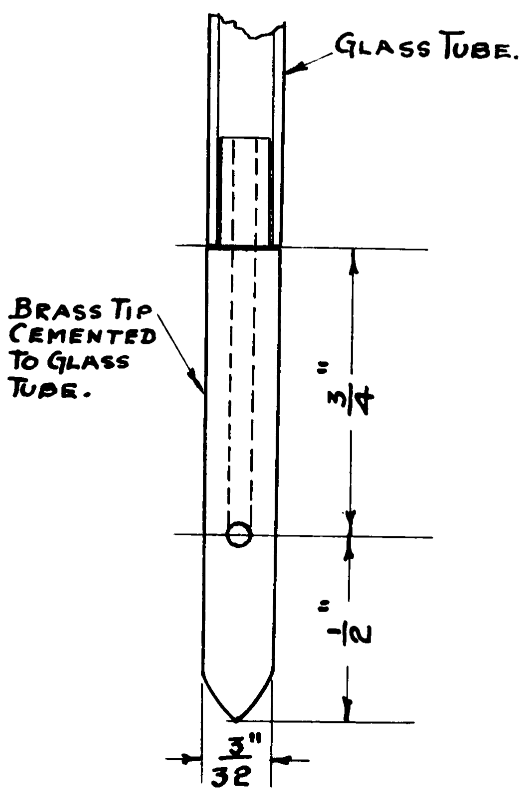

Fig. 6

same diameter, the details of which are shown in Fig. 6. When in use the pitot-tube was held vertically in a bracket, the hole in the tip being at the junction of the diverging tube and the wooden nozzle. A wire bracket was used to keep the end of the tube in the center of the throat. During the vacuum tests, as will be explained later, the level of the water in the box was falling. This caused water to rise in the pitot-tube causing a fictitious vacuum to be recorded by the U-tube. To obtain the correct vacuum the stop-cock was first turned so as to open the pitot-tube to the atmosphere, thus allowing any water in it to escape. The stop-cock was then turned back so as to connect the pitot-tube with the U-tube, the reading being taken just as the water appeared above the brass tip of the pitot-tube. Using the pitot-tube in this manner reduced the area of the throat slightly, but, as the method in itself is not very accurate, this difference was neglected. During the volumetric tests the pitot-tube was removed.

Procedure.

The following procedure comprised a volumetric test: After having placed a diverging tube in the test position, the box was filled with water to the required level. The 1 in. by-pass valve was adjusted until the level remained constant. The water level in the flume was then adjusted until the required submergence of the tube was obtained. Final adjustments of the two levels were made by means of the close adjustment valves until both levels remained constant. The swivel receptacle was then shifted from the waste position to direct the water to the weighing-tank or the measuring-tank, the time of shifting being noted. Simultaneously with this, and at minute intervals afterwards throughout the test, readings were taken of the float gauge and of the water level in the box. During the experiment both the levels were maintained as nearly constant as possible. The duration of the test was measured by the second hand of an ordinary watch and also by a calibrated stop-watch, the one checking the other. Each test was run until a quantity of water passed through the diverging tube sufficient to maintain the accuracy of the experiment to approximately one-eighth of one percent.

Vacuum tests were carried out as follows: The pitot-tube was inserted into the throat of the diverging tube and clamped vertically and centrally in position. After filling the box almost to capacity, the supply was reduced and the level in the box allowed to fall. At intervals corresponding to changes of level in the box of approximately 2 in., simultaneous readings were taken of the following (1) float gauge (2) water level in the box (3) throat vacuum.

Throughout the experiments zero readings of the gauges and readings of the barometric pressure and the temperature of the water were taken at frequent intervals.

Results and Discussion.

Plates 7 and 8 give the numerical results of the complete series of experiments. In the volumetric tests computations were made using the 5-place logarithms found in King's handbook of hydraulics. All efficiency calculations with the exception of the volumetric efficiency were made on the slide rule.

A comparison of Tests 1 - 33, Plate 7, carried out on the two similar tubes indicates very little variation over the entire range of heads used in the experiment of either the throat discharge coefficient or the volumetric efficiency. When plotted to a large scale on graph paper against the head the figures from the tests on both tubes show no direct variation, the points on the graph being scattered. The average results of the tests are given in Table 1. It will be noted that the average discharge coefficient of each of the tubes is 2.00. This coefficient checks with that obtained by Venturi. From the results of the experiments it was inferred that for all practical purposes when operated under similar conditions the performances of the tubes will be identical.

Tables of Average Results

Table 1 --- Similar Tube Tests

Test	$\frac{A_2}{A_1}$	Spacer Length	Vol. Effcy.	Th.Disch. Coef.	The Effcy.	Effcy. of Regain
Plain 5 in. Tube	3.52	-	56.9	2.00	69	75.0
Plain 10 in.Tube	3.52	-	56.7	2.00	-	75.0

Table 2 --- Modified 5 in. Tube Tests - Spacer length varied

Test	$\frac{A_2}{A_1}$	Spacer Length	Vol. Effcy.	Th.Disch. Coef.	The Effcy.	Effcy. of Regain
6 in. Regainer	4.55	.197	50.3	2.28	76	80.8
"	6.27	.272	38.6	2.42	76	82.9
"	7.13	.309	33.8	2.41	81	82.8
"	9.14	.396	25.4	2.33	77	81.6
"	15.18	.658	14.3	2.17	71	78.8

Tables of Average Results

Table 3 --- Modified 5 in. Tube Tests - Regainer diameter varied

Test	$\frac{A_2}{A_1}$	Spacer Length	Vol. Effcy.	Th.Disch. Coef.	The Effcy.	Effcy.of Regain
6 in. Regainer	7.13	.309	33.8	2.41	81	82.8
8 in. Regainer	9.59	.309	23.9	2.29	75	81.0
9.95 in. Regainer	12.02	.309	19.5	2.34	82	81.7

The variation of the throat vacuum with the head for each of the tests carried out on the modified 5 in. tube is shown graphically in the curves (Plates 9 and 10). It will be noted from these curves that for a given regainer diameter there is a definite spacer length giving maximum throat vacuum. In order to investigate this relation further the vacuum at one particular head, in this case 2 ft., was plotted on an $\frac{A_2}{A_1}$ ratio base. (Plate 11). This curve indicates that the ratio for maximum throat vacuum is 7.3. On the same sheet is shown the theoretical vacuum curve plotted from the relation $V = \frac{L - S + H\left[\left(\frac{A_2}{A_1}\right)^2 - 1\right]}{13.6}$. Actually V cannot be greater than atmospheric pressure, i.e. 30 in. of mercury. Substituting this value of V in the equation gives a ratio of $\frac{A_2}{A_1}$ equal to 4.2. It is interesting to note that at this ratio the slope of the actual vacuum curve changes abruptly, reaching a maximum shortly afterwards. Any increase in the spacer length after the best ratio is reached reduces the vacuum gradually until a further increase has no more effect on the discharge.

When this condition is reached the curve will coincide with the asymptote shown in the graph. The reason for the shape of the vacuum curve is due to the action of the water impinging on the bottom plate of the regainer. On impingement the water turns at right angles and flows radially from the regainer. With a spacer length smaller than that required for maximum flow the regainer reduces the discharge from the tube and consequently the throat vacuum. With spacer length greater than that required for maximum discharge the tendency of the water is probably to leave the upper surface of the regainer with a consequent formation of turbulence and eddies. This tends to reduce the discharge from the tube and with it the throat vacuum.

The curves (Plate 12) plotted from Table 2 on an $\frac{A_2}{A_1}$ ratio base show graphically the variation of the efficiencies and throat discharge coefficients obtained with the modified 5 in. tube. In general the shape of each curve corresponds to that of the throat vacuum, the maximum point being reached at a value of $\frac{A_2}{A_1}$ approximately equal to 7. As the efficiencies and discharge coefficients of the tube depend directly on the throat vacuum this is what is to be expected.

Similar experiments to those conducted by the author on the radial regainer were carried out by both Mr. A. H. Gibson and Mr. W. M. White. In Mr. Gibson's tests water was introduced directly through a 3 in. pipe to a 13.52 in. dia. radial regainer. Maximum efficiency was obtained with a distance between the parallel surfaces of approximately 0.5 in. which corresponds to a value of $\frac{A2}{A1}$ equal to $\frac{\pi \times 13.52 \times .5}{\frac{\pi}{4}(3)^2} = 3$. In Mr. White's experiments two regainers were used of outlet diameters 33 in. and 18 in. respectively. In each test water was delivered directly to the regainer through a straight 4 in. dia. pipe. With the 33 in. regainer maximum efficiency was obtained with a distance between the surfaces of 0.58 in. and with the 18 in. regainer a distance of 0.66 in. This corresponds to values of $\frac{A2}{A1}$ equal to $\frac{\pi \times 33 \times .58}{\frac{\pi}{4}(4)^2} = 4.8$ and $\frac{\pi \times 18 \times .66}{\frac{\pi}{4}(4)^2} = 3$ respectively. Direct comparison of the results of the above tests with those of the author cannot be made as in the author's tests water was not conveyed directly to the regainer through a straight pipe but passed first through a diverging tube. However, if the diameter of an equivalent pipe handling the same discharge as that of the tube be used, a fair comparison may be made. In calculating this pipe diameter the contraction coefficient at the entrance was assumed to be 0.95.

Discharge per sec. through the diverging tube $= 2\frac{\pi}{4} D_t^2 \sqrt{2gH}$

Discharge per sec. through the equivalent pipe $= 2\frac{\pi}{4} D_p^2 \sqrt{2gH}$

where D_t and D_p are the inlet diameters of the tube and pipe respectively.

Therefore, $D_p^2 = \dfrac{2 \cdot \frac{\pi}{4} D_t^2 \sqrt{2gH}}{.95 \frac{\pi}{4} \sqrt{2gH}} = \dfrac{2(1)^2}{.95} = 2.10$

whence $D_p = 1.45$

Using 1.45 as the inlet diameter of the regainer used by the author gives a value for maximum efficiency of $\frac{A_2}{A_1}$ equal to $\frac{\pi \times 6 \times .309}{\frac{\pi}{4}(1.45)^2} = 3.5$ which compares favourably with that obtained by Mr. Gibson and that obtained by Mr. White with the smaller regainer.

Table 3 gives the average results of the tests carried out with regainers of different diameters. The figures indicate that an increase in the diameter of the regainer increases the efficiency, the efficiency of regain and the discharge coefficient of the diverging tube.

Conclusion.

The following is a resumé of the general conclusions drawn from the entire investigation:

(1) For all practical purposes the performances of similar diverging tubes are identical.

(2) For a diverging tube modified by the addition of a regainer there is a definite ratio of outlet area to throat area giving maximum efficiency, discharge and efficiency of regain.

(3) For any width of passage of the radial regainer the efficiency, discharge and the efficiency of regain of the tube increases as the diameter of the regainer is increased.

It would be of interest in the future to conduct tests on diverging tubes under conditions whereby water entering the tube could be given a definite angle of whirl. This possibly could be accomplished by placing several guide vanes under a circular plate around the entrance to the tube in much the same manner as the movable vanes of a hydraulic turbine.

PART TWO

Cavitation Phenomena.

The phenomenon known as cavitation appears to have been first identified in the trials in 1894 of the torpedo boat destroyer "Daring" which had reciprocating engines. When driven at full power with the original screws, this vessel showed very serious vibration due to the formation, in the water about the propellers, of cavities or voids filled with water-vapour. This difficulty was overcome for the time being by modifying the design and broadening the blades of the propellers.

During the World War and after, however, with the advent of quick-running turbine driven propellers, cavitation difficulties were re-encountered in greater severity. Not only was the vibration very severe but in many cases, after a comparatively short time, the propeller blades were found to be so badly eroded and pitted that it was necessary to renew them. On examining many of the blades it was also found that in some cases the tips were severely bent, the eroded area being on the side convex to the bend.

The problem was taken up by the Hon. Sir Charles A. Parsons and Mr. Stanley S. Cook[1] who investigated the phenomena from the points of view of the following five possible causes:

(1) The nature of the surface of the metal and the state of the initial stress on this surface.

(2) The stresses in the blade under working conditions.

(3) Impingement of water at high velocities on the surface of the blade.

(4) Cavitation.

(5) Water hammer produced by the collapse of vortex cavities.

Following out the procedure suggested by these topics, the structure of the blades was first of all examined under the microscope. This examination failed to show any connection between the structure of the metal and the distribution of erosion.

Bars of brass alloys, similar to those used in the manufacture of propellers, were then stressed in various ways and placed under sea-water. After a few weeks had elapsed these were removed and, on inspection, presented a uniform appearance to the eye. Thus again valuable negative results were obtained.

Experiments were then made with jets of sea-water

[1] See Reference 7, Bibliography.

impinging on flat plates of brass alloys under a pressure as high as 1500 lbs. per sq.in. These tests produced an etching effect, as if the plates had been attacked slightly by an acid.

Rods of brass alloys were next placed in the throats of diverging tubes through which sea-water was pumped, the tubes being designed so that the throat pressures would be low enough to allow the formation of voids or cavities, thus exposing the rods to the action of water-vapour. This action produced an etching effect similar in character to that obtained in the previous test.

An endeavour was then made to determine the effect of the collapse of vortex cavities formed in water. Previous to this, Mr. Stanley S. Cook, whose results were later corroborated by Lord Rayleigh, had calculated that remarkably high pressures were produced instantaneously by the sudden collapse of vapour bubbles or voids at the point at which the collapse takes place, and furthermore, that the magnitude of the resulting pressure was independent of the original diameter of the bubble but depended only on the ratio of the initial to the final diameter of the bubble. With a ratio $\frac{d_1}{d_0}$ equal to 0.1, where d_1 is the final diameter and d_0 the initial diameter of the bubble, Mr. Cook calculated that a pressure of 24.2 tons per sq.in. would be obtained and with a ratio of 0.01 a pressure of 765.0 tons per sq.in. Thus, if the void collapsed on a metal surface, it would not seem unreasonable to suppose that the metal would be pitted or marked in some way, and that, after the continued action of many such collapses, it would become badly eroded. The hammering action on one side of the metal would also tend to stretch that side, explaining the bent ship-propeller blades.

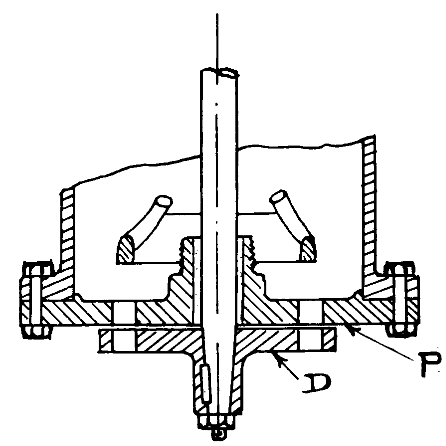

Fig. 7

In some experiments made on water-sirens, erosion, similar in character to that occurring in propeller blades, has been observed. The water-siren, shown in Fig. 7, consisted of a fixed plate **P** with ten circular holes perpendicular to its plane equally spaced on a 2 in. diameter. Beneath this plate was a revolving circular disc **D** with corresponding holes, which could be turned at as high a speed as 1500 r.p.m. Water was maintained under pressure behind the fixed plate and discharge took place through the apertures whenever they coincided. Thus there resulted a rapid succession of interrupted jets. On examining the apparatus after a short run of a few minutes, the lower edges of the holes in the fixed disc were found to be badly eroded and laid over as if by a hammering action. Tests were then run with fixed plates of very hard steel and from the erosion that resulted it was estimated that instantaneous pressures as high as 140 tons per sq.in. were present. To explain this action, imagine one of the jets quickly interrupted. The momentum of this jet will momentarily form a small void which will be immediately annihilated by the inrush of the surrounding water which, on destroying its momentum, will recoil back on itself. This return action by concentration will finally produce high velocities and pressures in a very small volume of the water towards the interior of the cavity, giving rise when the collapse takes place on the metal surface to severe water-hammering over a small area.

To test the principle more directly a means was sought by Sir Charles Parsons and Mr. Cook under conditions which could be better controlled. A brass conical shell 18 in. long, shown in Fig. 8, was made and fitted with a brass endpiece E , the interior of which was a continuation of the interior of the cone. The endpiece terminated with an internal diameter of 0.015 in. This cone was placed under water, in a tank, and allowed to fill completely. It was then thrust quickly downwards until its mouth struck against a rubber block at the bottom of the tank. This sudden arrestment of the cone gave a high relative acceleration of its contents, momentarily producing a cavity at the apex, which, however, immediately filled giving an audible metallic clicking sound.

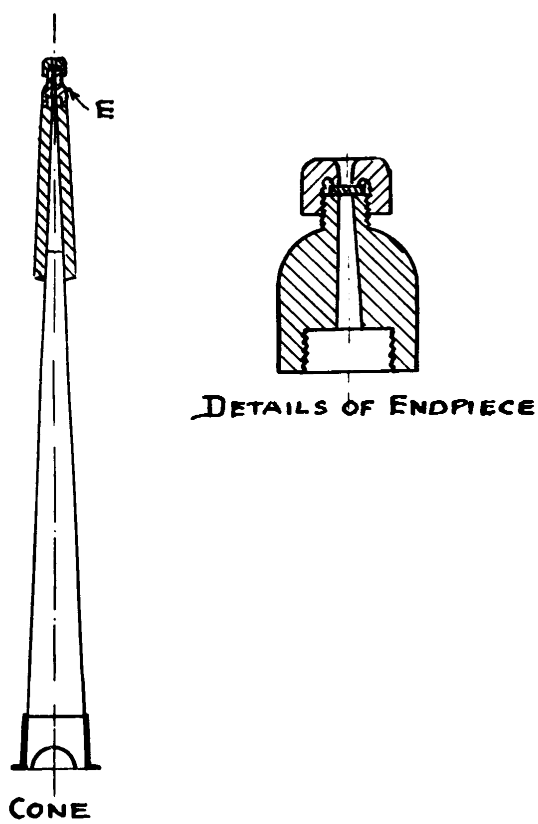

Fig. 8

Efforts were then made to determine the pressure produced at the apex of the cone. A new endpiece was constructed similar to the first one but providing a means of closing the end with a sheet of metal of definite thickness. On repeating the tests with the endpiece sealed with a disc of ordinary commercial brass, it was found that thicknesses up to 0.0035 in. were perforated with one blow and thicker plates severely dented. This showed that pressures at least as high as 15 tons per sq.in. were reached. In one test the cap of the endpiece was fitted loosely, allowing the plate to move laterally, and after about 200 blows the plate was covered with small indentations which corresponded very closely to those found on the eroded propeller blades. On lengthening the cone and replacing the brass endpiece with one of hard steel, it was found possible to puncture thicker plates, showing definitely that pressures as high as 140 tons per sq. in. were present at the apex of the cone.

It was inferred from these experiments that the corrosion of propeller blades is very slight, the pitting and erosion being caused mainly by the severe water hammer caused by the collapse on the surface of the blades of vortex cavities.

In the early type of inefficient water turbine cavitation problems were not encountered. With the development of high specific speed wheels necessitating the efficient draft tubes now in use, erosion and pitting, caused by cavitation, became a factor which must be guarded against.

One of the advantages in the use of the draft tube is the possibility, by its use, of setting the runner at such an elevation above the tail race that the wheel and its parts can be properly inspected by draining the water from the wheel pit and still not lose any of the available head of water. The draft tube, however, not only enables the turbine to utilize by suction that part of the fall from the exit tips of the runner blades to the level of the tail race, but it also gradually slows down the exit velocity of the water discharged from the runner with as little shock as possible, thus enabling the unit to utilize as much as possible of the velocity head of the water leaving the runner. In modern low head power plants using propeller type runners the velocity head to be recovered in the draft tube sometimes amounts to almost one-third of the total available head, necessitating draft tubes of very high efficiency. When in operation, as will be explained in the next paragraph, the pressure in the throat of a draft tube beneath the runner is reduced considerably below atmospheric pressure. When the absolute pressure in the draft tube throat is reduced below a certain limit it is thought that cavities are formed by the release of gases dissolved in the water.

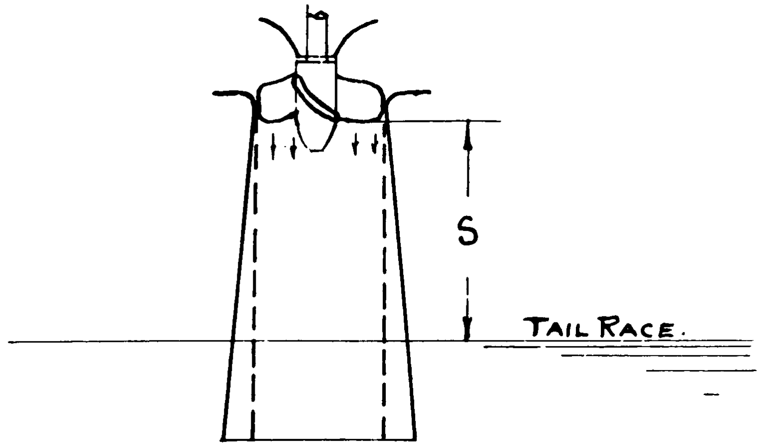

Fig. 9

If, in a hydraulic turbine unit, H ft. is the effective head on the turbine and S ft. is the height of the discharge tips of the runner above the tail race, the static absolute pressure measured in feet of water at the throat of a draft tube of uniform section, shown by dotted lines in Fig. 9, will be $P_b - S$, where P_b is the barometric pressure corrected for the vapour pressure of the water at river temperature. If this uniform section draft tube is replaced by a diverging tube, shown in the sketch by full lines, the absolute throat pressure, as is seen by applying Bernoulli's theorem, will be reduced still lower by an amount corresponding to the regained velocity head in the tube. Thus, if P_2 feet of water is the absolute pressure in the throat,

$$P_2 = P_b - S - E_d \frac{v_2^2}{2g}$$

v_2 being the absolute velocity of the water leaving the runner, E_d the draft tube efficiency, and g the gravity constant. Actually in practice an additional term $K_c H$ is introduced to allow for the existence of local eddies or zones of low pressure. From experience it has been found that K_c , called the cavitation coefficient, is constant for various heads but varies with the runner design. Therefore, K_c varies with N_s, the specific speed of the runner, which is defined as the r.p.m. speed for a similar runner developing 1 h.p. at 1 ft. head. So allowing for the local eddies we may write

$$P_2 = P_b - S - E_d \frac{v_2^2}{2g} - K_c H$$

Therefore,
$$HK_c = (P_b - P_2) - s - E_d \frac{v_2^2}{2g}$$

Putting $(P_b - P_2) = P$ and rearranging the terms, we have,
$$HK_c + E_d \frac{v_2^2}{2g} = P - S$$

As the efficiencies of well designed draft tubes do not vary greatly and, as stated before, are constant for various heads we may write
$$HK_c + E_d \frac{v_2^2}{2g} = \sigma H$$

Therefore $\sigma = \frac{P-S}{H}$ and is called the cavitation factor. For large values of this factor the turbine will probably be free from cavitation, but for small values, cavitation with the ensuing eroding and pitting will take place. It has been shown by investigations of the National Electric Light Association and by Mr. L. F. Moody that the minimum safe value of the cavitation factor depends on the specific speed of the runner, 0.04 being the order of the minimum value for specific speeds of 12 and 0.60 for specific speeds of 90.

By actual analysis it has been found that gases held in solution by river water are richer in oxygen than the air from which they were derived. The theory has been advanced that such gases, on being released in low pressure regions in a turbine, oxidise the metal surfaces with which they come in contact. However, the general opinion is that this corrosive action is slight and that the main cause of the erosion and pitting in turbine units is due to the intense pressures ensuing from the sudden collapse of vapour voids in close proximity to metal surfaces.

Any irregularities in the passages of a hydraulic unit causing eddies or swirls tend to increase cavitation due to the lower pressure in the eddy itself. The interference causing the swirl may lie a long way in front of the inlet to the turbine but the void or cavity is not usually formed until the eddy actually comes into the low pressure region.

A peculiar erosion problem has arisen in some propeller type units with a high cavitation factor. The runner vanes which were of cast steel were found to be pitted slightly on the upper face of the leading tips and on the lower edge of the trailing tips. Heavy pitting also occurred in the draft tube itself in areas each immediately below the guide vanes, which, at normal gate openings, projected well beyond the curved edge of the throat. Many tests have been made in order to determine just what caused the pitting of the runner blades, but, although several theories have been advanced, no definite explanation has yet been made. It is probable that, as the guide vanes project over the curved edge of the draft tube, eddy formations will take place beneath each of the projections. Bubbles formed in these eddies, on being carried down the draft tube into areas of greater pressure, will probably collapse and erode the walls in individual areas each below one of the projecting vanes.

In general the water, coursing through the turbine, passes very quickly through the low pressure areas. The question has been asked, whether, in this short space of time, the vaporization of water is at all possible, the argument being made that evaporation requires conduction of heat from the water to the bubble which is achieved by the comparatively slow process of heat transfer. As the available knowledge concerning the origin of these minute bubbles is vague, no mathematical proof has yet been made that there is enough time. However, by means of a cinematographic arrangement called a "time expander" which is capable of making 5,000 exposures per second,photographs have actually been taken showing the formation and collapse of bubbles in a test plant.

It is known that the velocity of the water in close proximity to the walls or metal faces is much less than that out in open passages, in fact, theoretically, the layer next to the surface is at rest. Water also tends to stagnate behind the slightest protrusion or obstacle, so it seems probable that if due precaution is not taken in the modern turbine to maintain the cavitation factor high, ample opportunities for a sufficient transfer of heat to take place will occur to cause the formation of cavities with the resulting pitting and erosion of the turbine unit.

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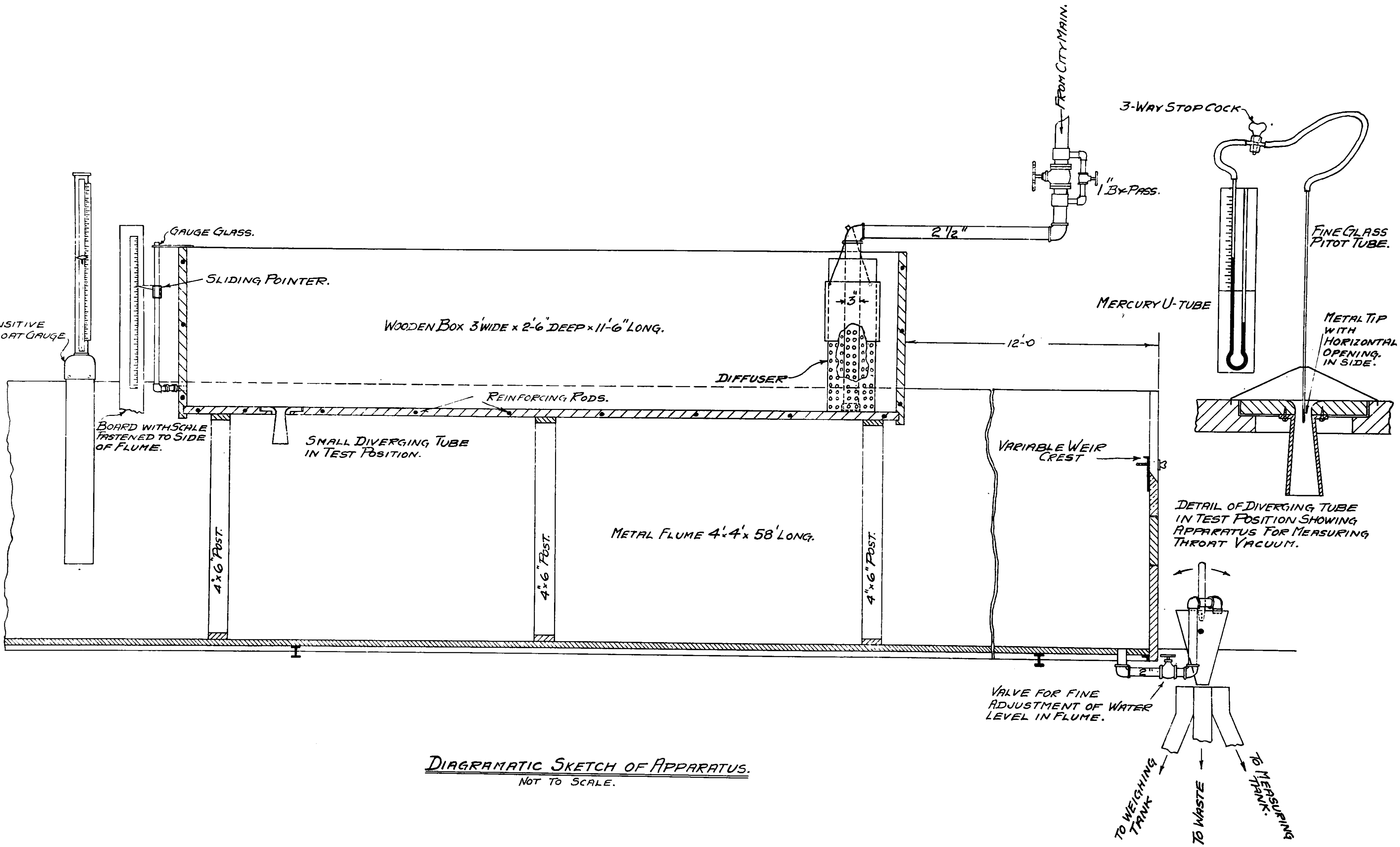
PLATE I.



DIAGRAMATIC SKETCH OF APPARATUS.
NOT TO SCALE.

View Showing Layout of Testing Apparatus

5 Inch Diverging Tube with
9.95 Inch Bell Mouth

5 Inch and 10 Inch Diverging
Tubes

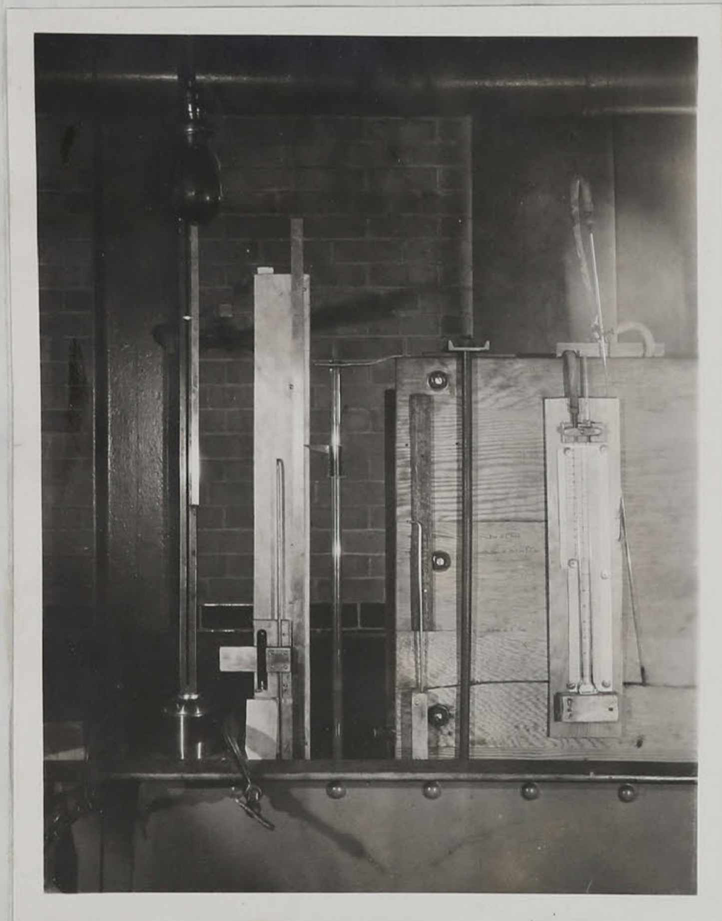
Plate 4

5 Inch Diverging Tube
with 6 Inch Bell Mouth

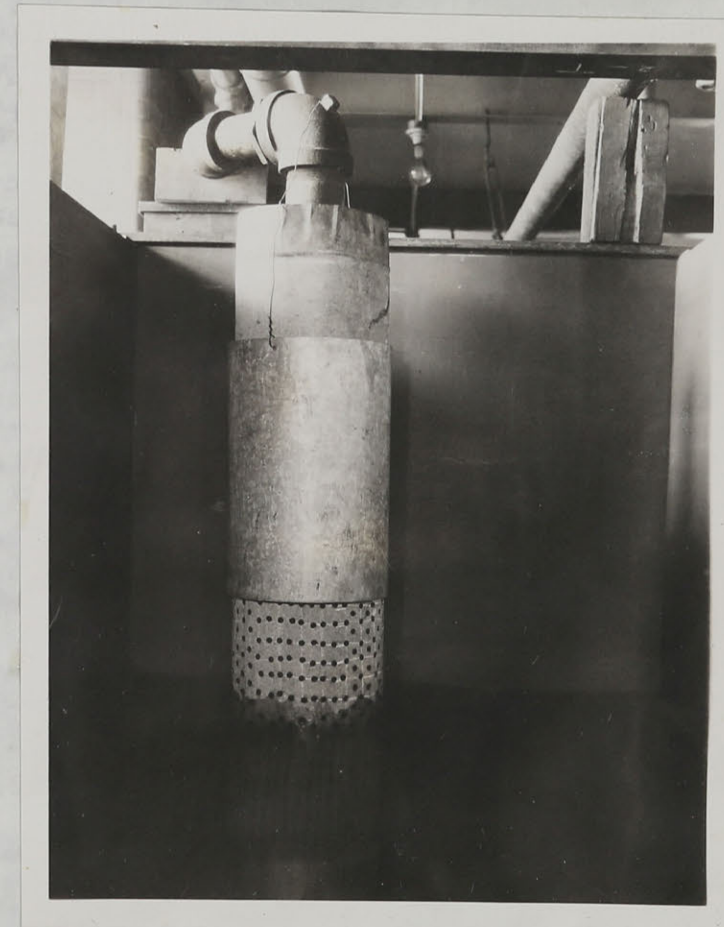
Diverging Tube in Test
Position

Plate 5

Plate 6



Apparatus Used in Measuring
Head and Throat Vacuum



Diffuser

Test	Tube	Areas. (Sq. In.) Throat	Outlet	Head (In.)	Sub. (In.)	Weight (Lbs.)	Time (Sec)	Velocity (Ft./Sec.) Throat	Outlet	Throat Vacuum (In Hg)	Throat Disch. Coef.	Vol. Effcy %	The Effcy. %	Remarks.
1	5"	·785	2·761											Insufficient number of Readings.
2	"	"	"											Vortex Spoiled test.
3	"			19.50	1/2	4213	600	20·64	5·87	4.73	2.02	57.3	66	
4	"	"		14.10	1/2	3962	660	17·64	5·02	3.47	2.03	57.6	63	
5	"	"		31.62	1/2	4206	480	25·75	7·33	7.51	1·97	56.1	73	
6	"	"		30.80	1/2	3714	420	26·00	7.40	7.32	2.02	57.4	66	
7	"	"	"	27.22	1/2	3974	480	24·30	6.92	6.50	2.01	57.2	68	
8	"	"	"	23.23	1/2	4154	540	22·65	6.44	5.60	2.02	57.5	65	Vortex stopped with Floating
9	"	"	"	18.61	1/2	4081	600	20·07	5·71	4.51	2.00	56.8	66	block of Wood. (Do.)
10	"	"	"	14.67	1/2	3982	660	17·76	5.05	3.60	2.00	56.8	68	Do.
11	"	"	"	12.31	1/2	4237	780	16·00	4.55	3.06	1·96	55.8	70	Do.
12	"	"	"	11.37	1/2	4023	780	15·18	4 32	2.84	1·94	55.1	73	Do.
13	"	"	"	13.16	1/2	4026	720	16·47	4.68	3.25	1.95	55.6	71	
14	"	"	"	15.75	1/2	4098	660	18·30	5 20	3.85	1 98	56.4	68	
15	"	"	"	29.07	1 1/2	4670	540	25.43	7.24	6.93	2.04	57.8	67	
16	"	"	"	23.33	1 1/2	4190	540	22·83	6.49	5.61	2.04	57.9	66	
17	"	"	"	19.02	1 1/2	4147	600	20·32	5·78	4.61	2.01	57.1	70	Vortex stopped.
18	"			16.59	1 1/2	4231	660	18·90	5 37	4.05	2.00	56·7	71	Do.
19	"	"	"	32.97	1/2	4364	480	26·70	7.60	7.80	2.01	57·1	71	Check Test.
20	"	"	"	13.82	1 1/2	4164	720	17·03	4.84	3.40	1.97	56·1	74	Vortex Stopped.
21	"	"	"	11.45	1 1/2	3900	720	15·95	4 53	2.86	2.03	57·7	67	Do.
22	"	"	"	18.77	8 1/2	4160	600	20·40	5·80	No Vacuum Readings Taken.	2.03	57 7		
23	"	"	"	3.85	8 1/2	2368	900	7 74	2.20		—	—		Vortex stopped.
24	"	"		17.65	8 1/2	3602	540	19.60	5.58		2.01	57.2		
25	"	"	"	15.22	8 1/2	4463	720	18·24	5 18		2.02	57.3		
26		"	"	8.47	8 1/2	4207	960	12.90	3.67		—	—		
27	10"	3·142	11.043	35.69	1									Water lost to Waste.
28	"	"	"	36.95	1									Do.
29	"	"	"	38.69	1	35125	900	28·64	8.15	No Vacuum Readings Taken.	1·99	56.5		
30	"	"	"	39.09	1	23408	600	28·63	8 15		1.98	56.2		
31	"	"	"	32.47	1	21304	600	26·09	7 42		1.98	56.2		
32	"	"	"	26.29	1	19511	600	23·87	6.79		2.01	57.2		
33	"	"	"	23.72	1	18595	600	22·79	6 48		2.02	57.4		Vortex stopped.

Water Temperature 34°F.

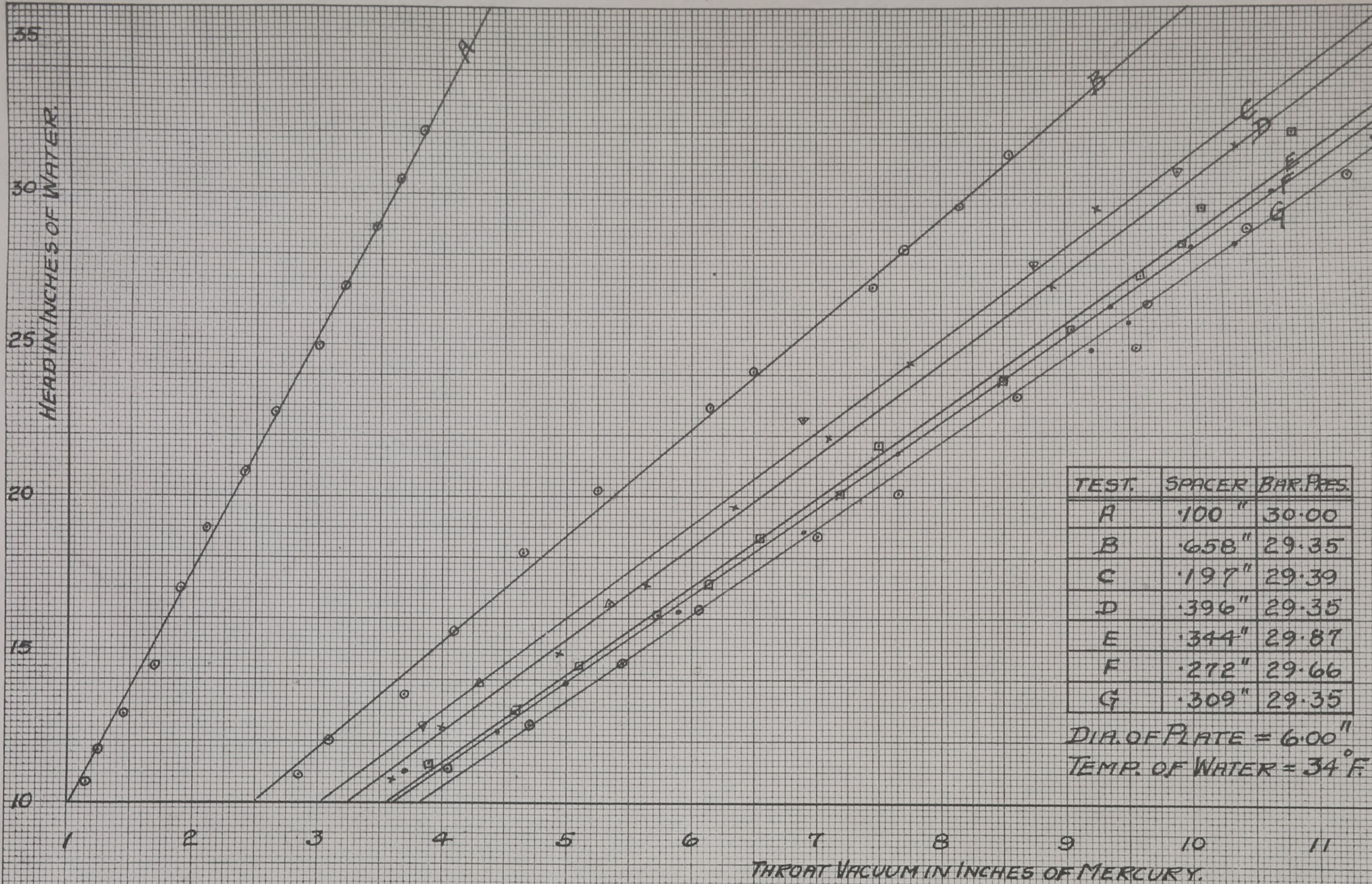
Plate 7.

TEST	TUBE	AREAS (SQ IN.)		HEAD	SUB.	WEIGHT	TIME	VELOCITY (FT/SEC)		THROAT VACUUM	THROAT DISCH. COEF.	VOL. EFFCY. %	THE EFFCY. %	REMARKS.
		THROAT	OUTLET	(IN.)	(IN.)	(LBS.)	(SEC.)	THROAT	OUTLET	(IN. Hg)				
34	5" TUBE WITH 6" DIA. PLATE	·785	3.566	31·42	2	4327	420	30·30	6·69	8·77	2·33	51·3	78	Spacer = ·197" $\frac{A_2}{A_1}$ = 4.55
35		"	"	31·33	2	4320	420	30·25	6·66	10 04	2·33	51·3	72	Do.
36		"	"	21 73	2	4043	480	24·75	5·45	6 90	2.29	50·4	74	Do.
37		"	"	14.75	2	3988	600	19·58	4 30	4·60	2.19	48·3	80	Do.
38		"	11.910	32·92	2									Spacer = ·658" $\frac{A_2}{A_1}$ = 15·18
39		"	"	22·98	2									At end of test 44 angle
40		"	"	14.82	2									holding bell to tube was
41		"	5.593	29.48	2									found to be loose.
42		"	"	24. 23	2									
43		"	"	19·62	2									
44		"	"	14.64	2									
45		"	"	29·79	2	4972	480	30·47	4 28	10·87	2.41	33·8	80	Spacer = ·309" $\frac{A_2}{A_1}$ = 7.13
46		"	"	24.30	2	4509	480	27·63	3·88	8·94	2 42	33·9	80	Do. Vortex Stopped.
47		"	"	18·72	2	3945	480	24·17	3 39	6·95	2.41	33·8	81	Do. Do.
48		"	"	13·45	2	4183	600	20 53	2 89	5·07	2.41	33·8	80	Do. Do.
49		"	7.168	32·90	2	4399	420	30·83	3·38	10·85	2·34	25·4	78	Spacer = ·396" $\frac{A_2}{A_1}$ = 9.14
50		"	"	26·27	2	4394	480	26·94	2·95	8·62	2.27	24·8	82	Do.
51		"	"	18·99	2	3790	480	23.25	2 54	6·24	2·30	25·?	78	Do. Vortex Stopped.
52		"	"	12·55	2	4013	600	19·67	2·15	4·10	2.40	26·2	68	Do. Do.
53		"	11.910	33·06	2	4098	420	28·72	1 89	9·13	2.15	14·2	76	Spacer = ·658" $\frac{A_2}{A_1}$ = 15·18
54		"	"	25·81	2	4155	480	25·44	1 68	7·05	2.15	14·2	74	Do. Vortex Stopped.
55		"	"	17·86	2	3861	540	21·03	1 39	4·78	2·14	14·1	72	Do. Do.
56		"	"	13·20	2	3839	600	18·83	1 24	3·42	2 23	14·7	61	Do. Do.
57		"	4.923	32·09	2	4582	420	32·20	5·13	11·42	2·44	38·9	73	Spacer = ·272" $\frac{A_2}{A_1}$ = 6.27
58		"	"	27·98	2	4276	420	30·00	4·78	9·96	2·44	38·9	74	Do.
59		"	"	20·89	2	4221	480	25·90	4 13	7·50	2·43	38·8	74	Do. Vortex Stopped
60		"	"	14·59	2	3846	540	20·95	3·34	5·28	2 36	37·7	81	Do. Do.
61	5" TUBE WITH 9.95" PL.	"	9·428	33·10	2	4376	420	30·70	2 55	11·20	2 30	19·1	83	Spacer = 309" $\frac{A_2}{A_1}$ = 12.02
62		"	"	23·16	2	4310	480	26·42	2 20	8·02	2 37	19·7	79	Do
63		"	"	14·48	2	4244	600	20·87	1.73	5·26	2·36	19·6	83	Do. Vortex stopped.
64	5" TUBE WITH 8" PL.	"	7·533	32·39	2	4288	420	30·12	3.14	10 70	2·27	23·7	66	Spacer = ·309" $\frac{A_2}{A_1}$ = 9.59
65		"	"	23·65	2	4297	480	26·38	2.75	7·97	2 33	24·3	78	Do.
66		"	"	16·14	2	3965	540	21·62	2 25	5·64	2 26	23·6	82	Do. Vortex stopped.

WATER TEMPERATURE 34°F.

PLATE 8.

PLATE 9.



TEST.	SPACER	BAR. PRES.
A	·100"	30·00
B	·658"	29·35
C	·197"	29·39
D	·396"	29·35
E	·344"	29·87
F	·272"	29·66
G	·309"	29·35

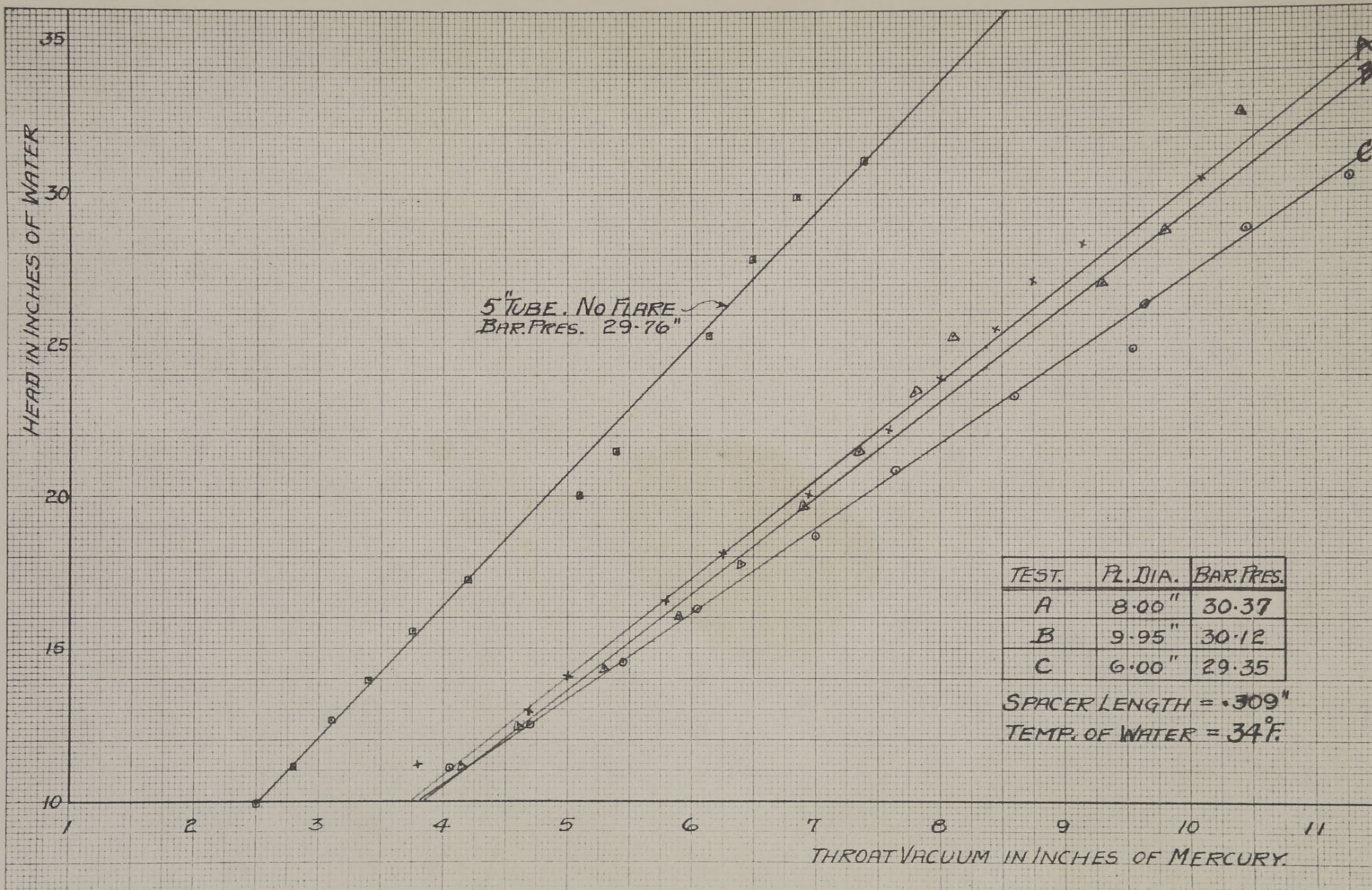
DIA. OF PLATE = 6·00"
TEMP. OF WATER = 34° F.

CURVES SHOWING RELATION BETWEEN THROAT VACUUM AND HEAD FOR
5" TUBE WITH FLARED OUTLET AND PLATE.

THIS MARGIN RESERVED FOR BINDING.

IF SHEET IS READ THIS WAY (HORIZONTALLY), THIS MUST BE TOP.

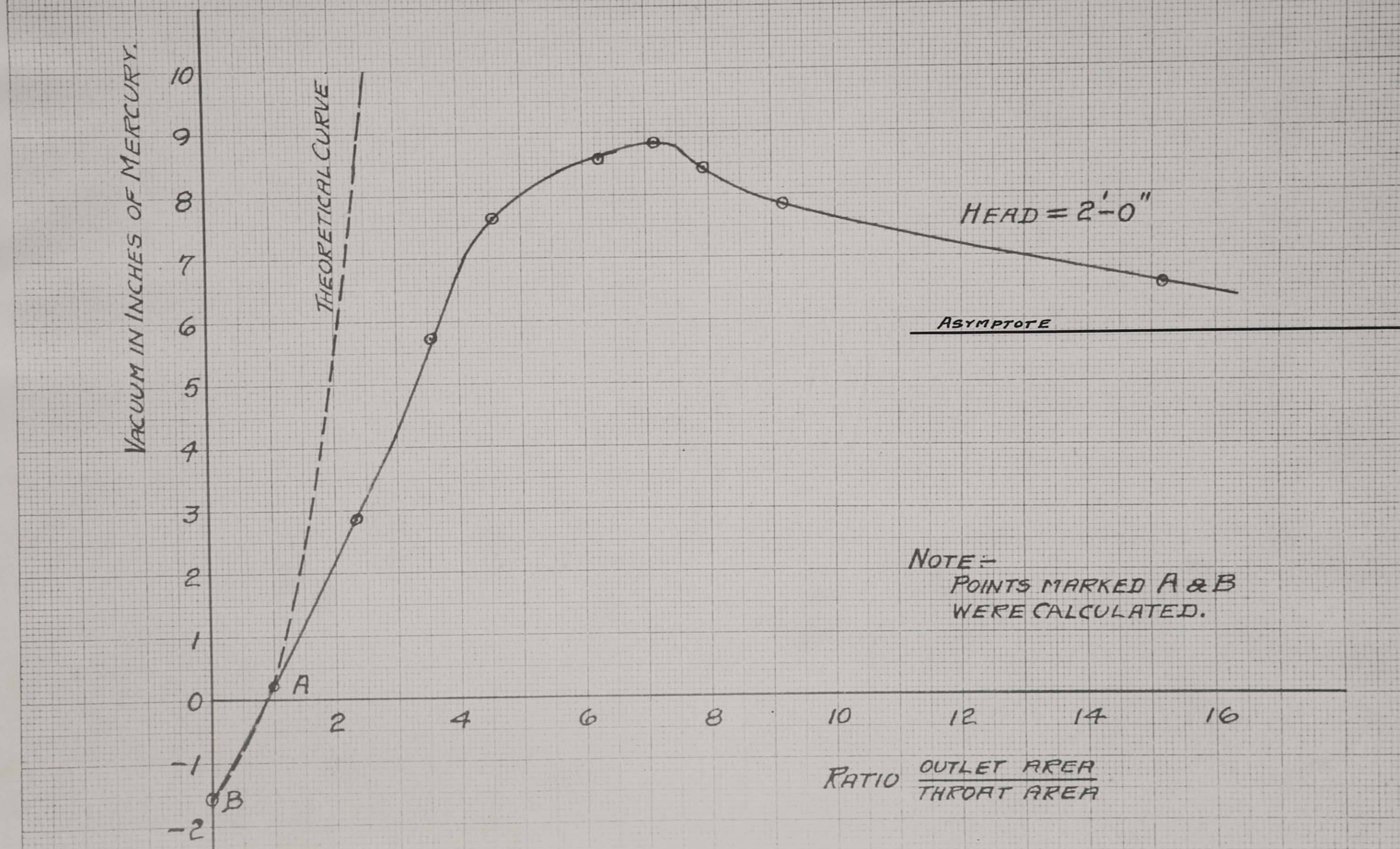
IF SHEET IS READ THE OTHER WAY (VERTICALLY), THIS MUST BE LEFT-HAND SIDE.



CURVES SHOWING RELATION BETWEEN THROAT VACUUM AND HEAD FOR 5" TUBE WITH AND WITHOUT FLARED OUTLET AND PLATE.

MANUFACTURED BY RENOUF PUBLISHING CO., MONTREAL FORM 100

PLATE 10.



CURVE SHOWING THE RELATION BETWEEN THROAT VACUUM AND THE RATIO OF OUTLET TO THROAT AREA.

PLATE II.

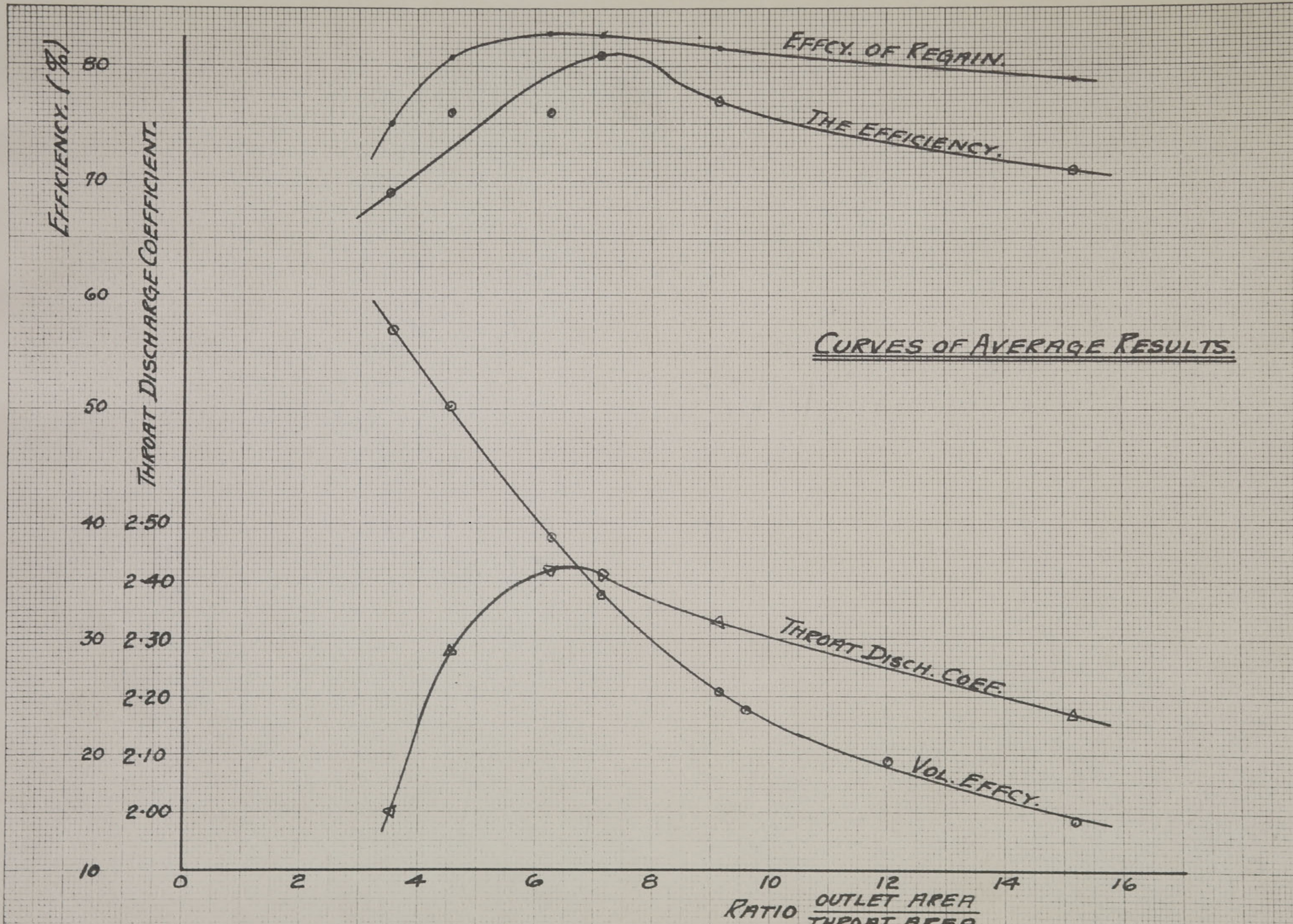

EFFICIENCY. (%)
THROAT DISCHARGE COEFFICIENT.
80
70
60
50
40
30
20
10
2·50
2·40
2·30
2·20
2·10
2·00
EFFCY. OF REGAIN.
THE EFFICIENCY.
CURVES OF AVERAGE RESULTS.
THROAT DISCH. COEF.
VOL. EFFCY.
0
2
4
6
8
10
12
14
16
RATIO OUTLET AREA / THROAT AREA

PLATE 12.