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Jack Carter Jones

INFLUENCE OF SEVERAL GEOMETRIC VARIABLES
ON THE EFFICIENCY OF
TWO MODERN TURBINE DRAFT TUBES

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By

Jack Carter Jones

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INFLUENCE OF SEVERAL GEOMETRIC VARIABLES

ON THE EFFICIENCY OF TWO MODERN TURBINE DRAFT TUBES

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ABSTRACT

As an example of a common hydraulic device, a draft tube is a diffusor, the primary duty of which is to convert high velocities to low velocities with a minimum loss of flow energy. It was the purpose of this investigation to determine the influence of several critical geometric variables on the hydraulic efficiency of two typical modern draft tubes. In order to accomplish this purpose, scale models of the draft tubes were installed in an apparatus which included a vertical water-supply pipe line; an adjustable, deflection-vane device; and a tailwater flume.

Comparative hydraulic efficiency tests were made for the following conditions: original designs, determination of the effect of the Reynolds number and the tailwater level; modified designs, investigation of the relative influence of roof slope and discharge opening, number of piers, lateral position of a single pier, angular position of a single pier, length of a single pier, and length of the horizontal leg. The efficiency was computed as a dimensionless piezometric-head ratio, involving the difference in piezometric heads at the throat and in the tailrace and the average axial velocity at the throat.

The test results are shown in graphical form. It was clearly demonstrated that, in most respects, the geometric details of the original designs were of nearly optimum form. It was also demonstrated, however, that some saving in cost without a serious loss of efficiency might be achieved by lowering the roof and shortening the horizontal legs of the draft tubes. The thesis includes a bibliography of 66 items.

CHAPTER I

INTRODUCTION

Definition of the Problem.--A draft tube is a divergent, enclosed conduit which conducts the discharge from a reaction-type hydraulic turbine to the tailrace. It has two principal functions: first, it provides that the turbine can be located above normal tailwater level without a corresponding loss of potential energy; second, and most important, it makes possible the use of a relatively small, high-speed turbine without an excessive loss of kinetic energy. As an example of a common hydraulic device, a draft tube is a diffusor, the primary duty of which is to convert high velocities to low velocities with a minimum loss of flow energy. It follows that, in general, the best draft tube is the most efficient diffusor. A definitive sketch of a draft tube is shown in figure 1.

The purpose of this investigation was to determine the influence of several critical geometric variables on the hydraulic efficiency of two typical modern draft tubes. Relative performance evaluations were based on hydraulic model tests in which the selected shape characteristics, all involving the downstream leg of the tube, were systematically varied. The variables included in this study were: slope of the roof and area of the discharge opening, number of piers, lateral position of a single pier, angular position of a single pier, length of a single pier, and length of the downstream leg of the tube.

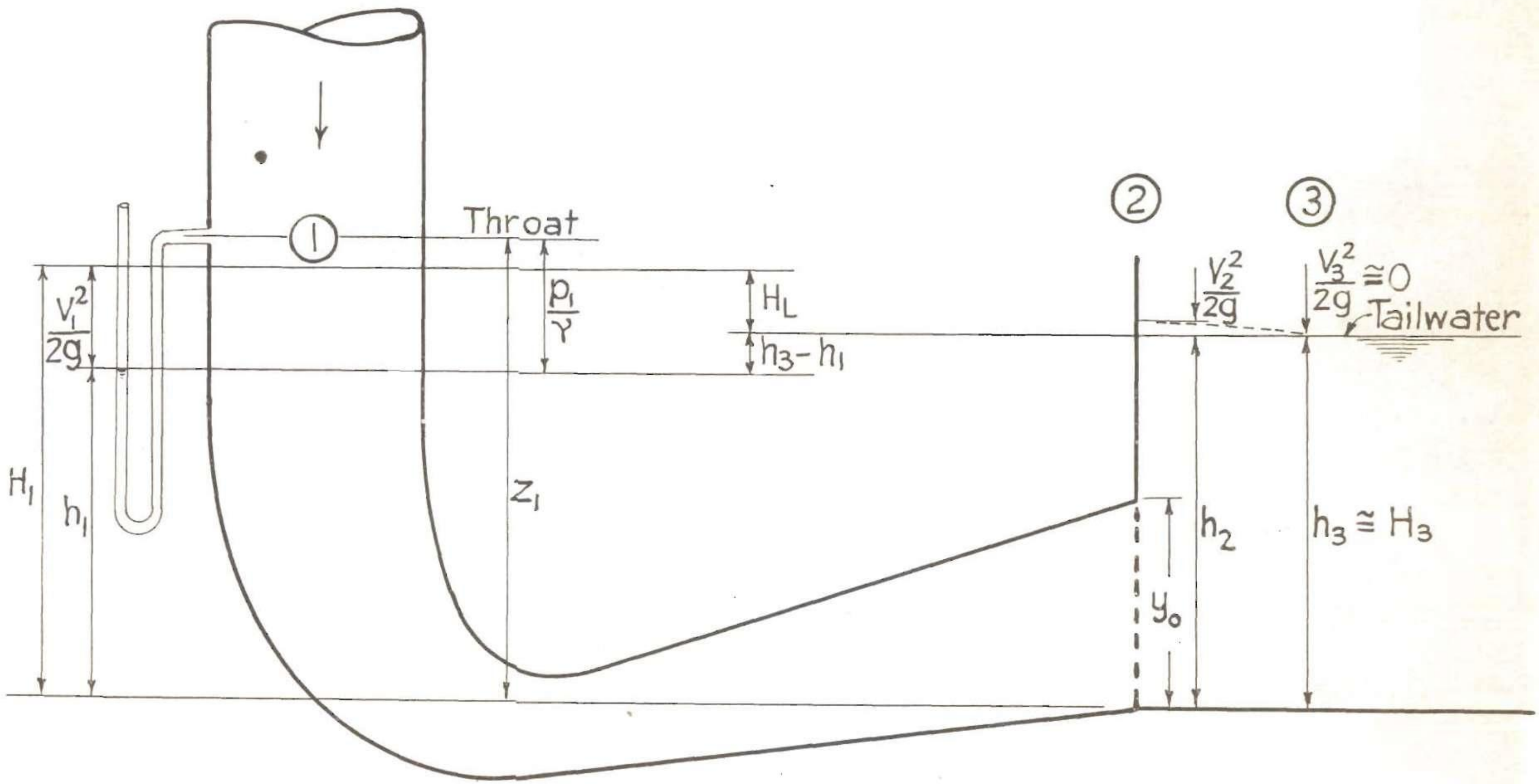


Fig. 1. Definition Sketch of Flow in a Draft Tube

History of Draft-Tube Research.--The important hydraulic research dealing with draft-tube design dates from the development of the modern high-speed, reaction turbine. As large Francis and propeller-type turbines were adapted to lower and lower heads, the efficiency of the draft tube became an increasingly important consideration in the design of the unit.

American research on turbine draft tubes and experience with them have been concerned with two general types. In the early part of the present century several very efficient forms of the conical or trumpet-shaped tube were developed and widely used. Recently the more economical elbow or quarter-turn type has largely replaced the conical tubes. The elbow tube has long been preferred in European practice.

Prasil (1) was one of the first to attempt a mathematical analysis of flow in draft tubes. Like many others, including Hillberg (2, 1915), Grimm (3, 1927), and Kaplan (4, 1929), Prasil ignored the effect of residual whirl in the discharge from the turbine. Contributions of considerable importance were made by workers in related fields. Gibson (5, 1912), for example, published the results of important research on the energy losses in straight conical expansions in circular pipes. Zur Nedden (6, 1916), Yarnell and Nagler (7, 1935), and others contributed valuable information on the flow of fluids in bends.

One of the earliest hydraulic model studies to determine draft-tube characteristics was reported by White (8, 1921). A product of his investigation was the Hydraucone Regainer, a vertical trumpet-shaped tube which utilizes a central cone of whirling water to facilitate the diffusion of the high-velocity discharge from the turbine. At about the same time Moody (9, 1921) described a similar conical-type draft tube which features a fixed axial cone extending upward from the floor of the draft tube. Various modifications of the White Hydraucone Regainer and the Moody Spreading Draft Tube have been widely used in the United States. In 1924 Allen and Winter (10) described the results of a comprehensive model investigation in which all of the common types of draft tubes were tested under similar conditions. An exhaustive review of draft-tube research and experience was contributed by the many outstanding engineers who engaged in the published discussion of the Allen-Winter paper. Six years later, in 1930, the National Electric Light Association published a report (11) which summarized the results of all of the experimental data on draft tubes available at that time.

In 1935, at the request of the Tennessee Valley Authority, the U. S. Bureau of Standards prepared an annotated bibliographic report (12) on the English and foreign language literature on draft tubes. The current (1935) trend in draft-tube design, according to that report, "shows a persistent tendency to depart from the use of spreading and symmetrical types." A detailed laboratory investigation of the flow characteristics in elbow draft tubes was published by Mockmore in 1938 (13). Verification of some of the reasons which accounted for the trend to elbow tubes was provided by Leutelt (14, 1940), who conducted a systematic laboratory comparison of the performance of conical and elbow tubes.

Most laboratory investigations of draft-tube performance have been conducted with models of a complete turbine unit. This is probably the best procedure, but it is not ideal. Draft-tube characteristics and runner characteristics are inextricably related. Thus, comparative tests on different forms of draft tube should be correlated with the characteristics of the test runner. A comprehensive investigation, therefore, would require the use of many different runners. The cost of this procedure is usually prohibitive.

Winter, in some unpublished studies conducted for the Alabama Power Company in 1928, was one of the first to base the relative performance of the draft tube on its hydraulic efficiency as a diffusor. In Winter's tests, the whirl component of runner discharge was simulated by means of fixed vanes located in the position of the turbine speed ring. Similar tests, in which draft-tube efficiency was related to an orifice coefficient, were performed by the U. S. Bureau of Reclamation for the Wheeler Dam project (15, 1934). The technique of hydraulic testing was considerably simplified and improved by Kindsvater and Randolph (16, 1954). In their investigation, conducted in the Georgia Tech Hydraulics Laboratory, the draft tube was connected to a uniform, vertical pipe in which adjustable deflection vanes provided a means of simulating runner discharge whirl. This apparatus, first developed for the Alabama Power Company's Martin Dam project, was subsequently adapted to a limited investigation of elbow draft tubes for the S. Morgan Smith Company. In substantially the same form, it was used for the tests conducted for this thesis.

Present Trend in Draft-Tube Research.--As in the past, most of the recent research on draft tubes has been conducted or sponsored by the major turbine manufacturers of the world. Their search for the best draft tube has not been concerned with efficiency alone. The pressure of competition and the demands of the consumers have caused equal emphasis to be placed on economy. Nevertheless, the continued effort to lower the cost of hydroelectric plants in comparison with fuel power plants has resulted in renewed attempts to find a draft tube which will be cheaper to build without being less efficient. Confirming this objective, a committee of the Edison Electric Institute recently stated (17):

> Present day high specific speed runners require low settings with consequent deep excavation or long horizontal length to conform to turbine manufacturers' efforts to obtain high efficiency and output, since guaranteed turbine efficiency includes regain of suction head and is penalized by exit velocity. Plant designers and operating companies, however, are concerned with the overall economics involved, including cost of construction, operation and maintenance. Some sacrifice in hydraulic efficiency may be justified by savings obtained by more simplified design.

Considerations of efficiency alone invariably lead to the long, vertical, conical or trumpet-shaped draft tube. Considerations of economy lead to the short, elbow draft tube. Research has naturally been concerned with compromising these two considerations. The result is that the principal turbine manufacturers have developed tubes of very similar design. Future improvements will depend on a systematic study of each of the separate geometric variables which influence the performance of a draft tube. This thesis is intended to be a partial accomplishment of such a study.

Review of Research on the Elbow Draft Tube.—The literature of research on the elbow draft tube is voluminous, but not exhaustive. It would be more appropriate to observe that the results of the research reported to date have been poorly integrated.

Development of the trumpet-shaped tube from the straight conical tube eventually led to the elbow tube. Thus, Williams (10) reasoned that an elbow tube with a trumpet-shaped vertical leg could be patterned after the optimum dimensions of a true trumpet-shaped tube with bottom deflection plate. As a result, his draft tube was one of the first elbow tubes to yield an efficiency comparable with that of the better conical tubes.

Tests to determine the optimum radius of curvature for the inside and outside bends of elbow tubes have not been conclusive. Similarly, the ideal relative length of the downstream leg has not been established, although several investigators (10, 11, 15) have concluded that the efficiency is directly proportional to the length of the tube. Englesson (18) concluded that the optimum length and depth of the elbow tube relative to its diameter at entrance is a function of the specific speed of the runner. Voorduin (19), in an empirical analysis of the best modern practice, confirms this conclusion.

Tests by the U. S. Bureau of Reclamation (15) on the Wheeler Dam draft tubes indicated that the efficiency of the tube would not be decreased if the floor of the horizontal leg were changed from horizontal to a 1:4 positive slope. The limitations on this variable have not been established.

Considerable contradiction exists in the literature regarding the influence of horizontal splitters and vertical piers. On large units, piers are a structural necessity. Nevertheless, like splitters, piers have been utilized as straightening vanes, and their efficacy for this purpose is not clearly determined. Several investigators have found that splitters have a beneficial effect on the efficiency of elbow draft tube. Others have found, however, that the benefit is a function of the length of the splitter and the whirl component in the discharge from the turbine. Caflisch (20), for example, found that a splitter extending to the top of the tube decreased the efficiency. He also demonstrated that the detrimental effect of the splitter increases with the degree of whirl. It is known that the magnitude of the whirl component varies with the type and specific speed of turbine. Authorities, however, do not agree on this relationship.

It is apparent that the modern elbow draft tube is a product of experiment and conjecture. Nevertheless, as numerous tests have shown, the best modern tubes have achieved a remarkable degree of perfection. Additional refinements will depend on systematic investigations of critical geometric variables with due consideration of the influence of turbine characteristics.

CHAPTER II

LABORATORY SET-UP

General.--The laboratory tests for this thesis were made in the Hydraulics Laboratory of the Georgia Institute of Technology, School of Civil Engineering. Most of the special equipment required for the tests, as described below, had been developed for two previous draft-tube investigations sponsored by private agencies. The general arrangement of the laboratory set-up is shown on figure 2.

Draft-Tube Models.--The designs for the draft-tube models used in this investigation were taken from plans furnished by the S. Morgan Smith Company. The two models tested, as shown on figures 3 and 4, represent typical modern draft tubes of the quarter-turn or elbow type. The scale selected for both models was such that the throats of the model draft tubes were exactly six inches in inside diameter. The models were constructed of plastic-impregnated, glass-fiber cloth and were molded over accurately built plaster forms. Both tubes were built by a professional model builder with previous experience in building sheet-plastic draft-tube models.

Deflection-Vane Section.--At their inlet ends, the draft-tube models were connected to a vertical, six-inch pipe line. In order to provide a means of simulating the effect of runner-discharge whirl, manually adjustable deflection vanes were located in a short section of plastic

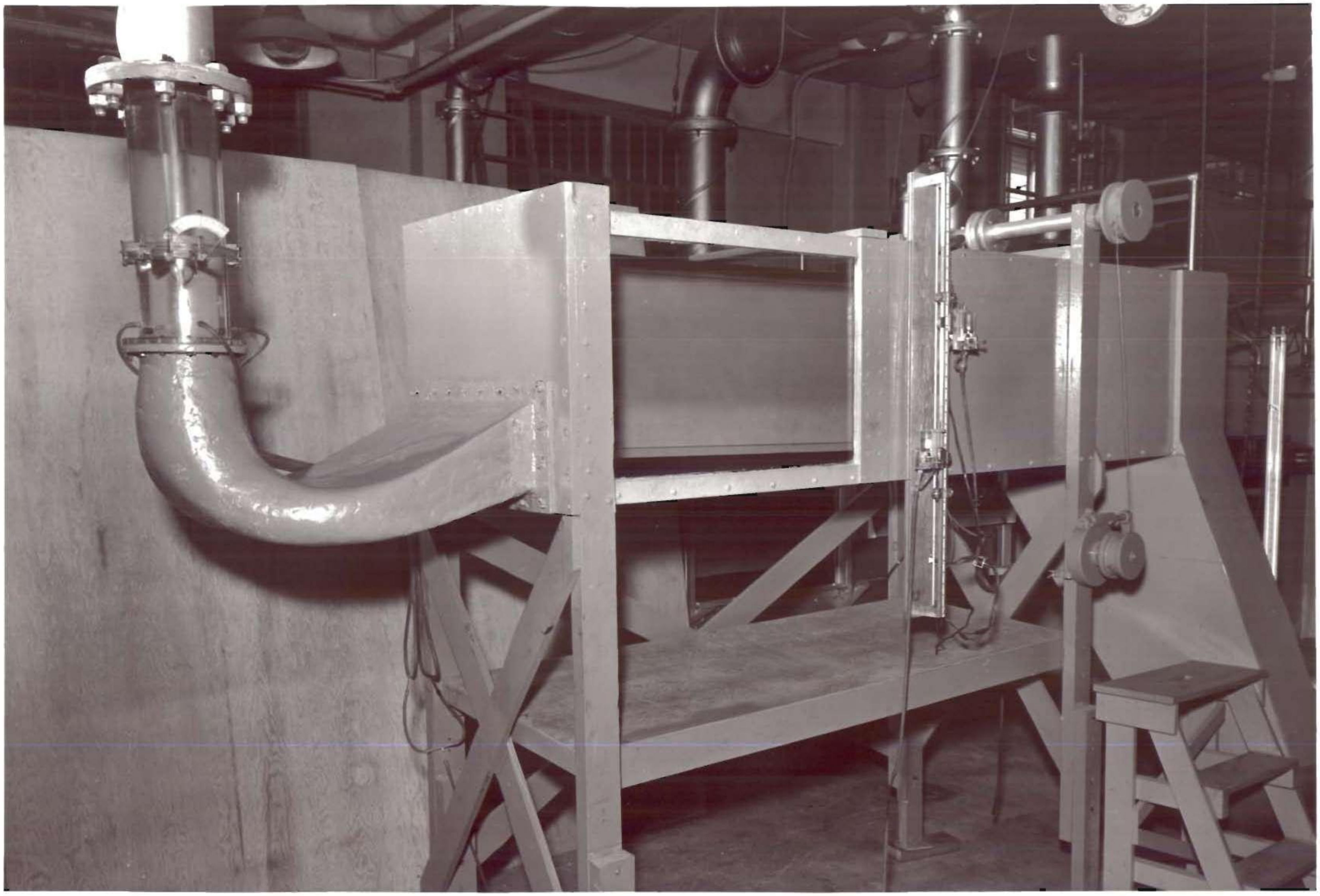
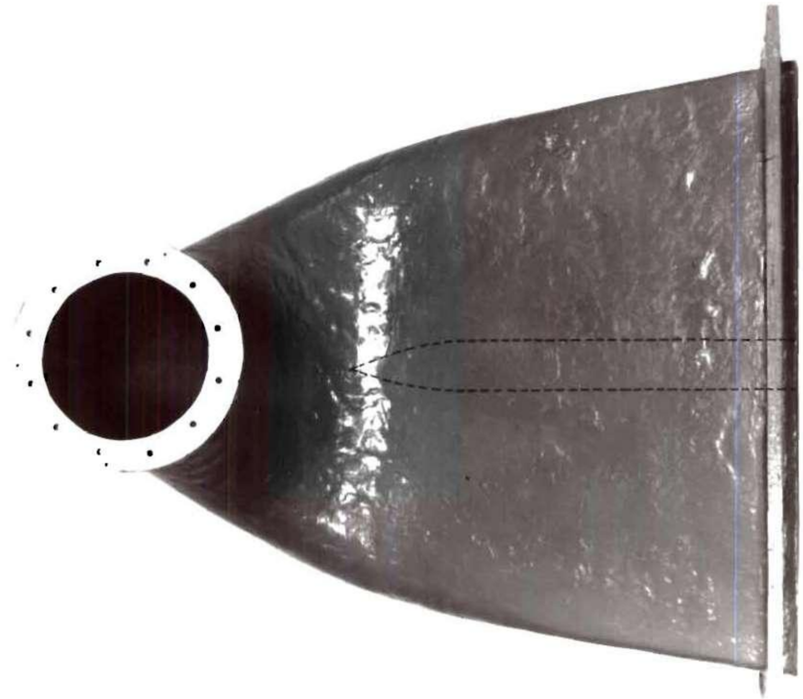


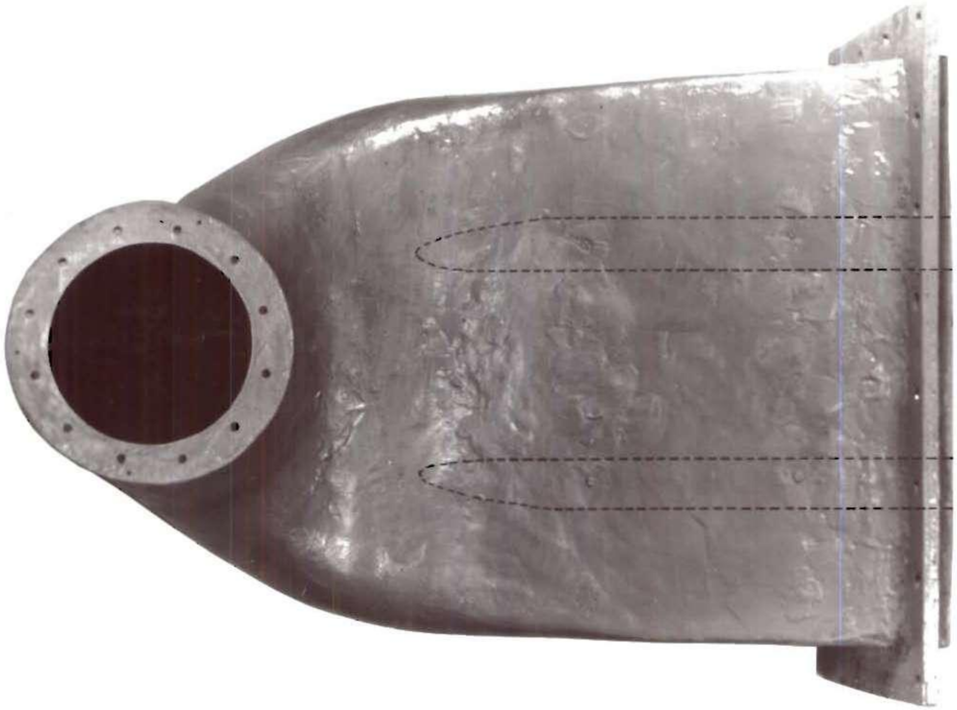
Figure 2. Laboratory Set-up for Efficiency Tests.



(a) Plan

(b) Side Elevation

Figure 3. Draft Tube Model No. 1, Original Design.



(a) Plan

(b) Elevation

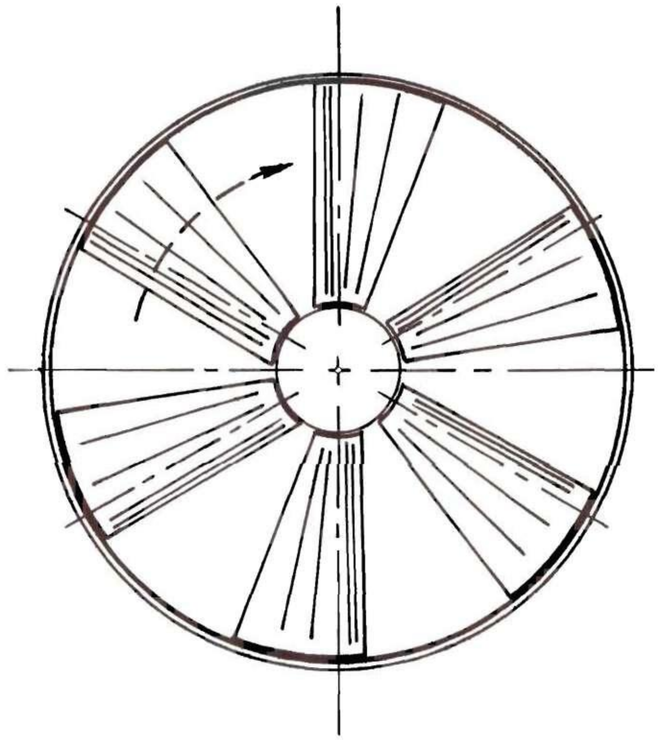
Figure 4. Draft Tube Model No. 2, Original Design.

pipe immediately above the draft tube. An external shift-ring connected to the radial vanes provided for their adjustment to any angle from 0 degrees to 35 degrees (measured with respect to the axis of the tube). The deflection-vane section is shown on figure 5.

Flume.--The draft tubes at their discharge ends were connected to an open flume having a width equivalent to a typical unit spacing. Tail-water control in the flume was provided by means of a hinged end-gate.

Piezometric-Head Measurement.--Four wall piezometers located about four inches below the deflection vanes were connected to a zero-displacement manometer for the measurement of the piezometric head at the throat of the draft tube. Two piezometers located in the floor of the flume, about five feet downstream from the end of the draft tube, were connected to a similar manometer for measuring the tailwater level. The manometers were so arranged that the difference in piezometric head between the throat and the tailwater could be measured directly. The manometers are shown on figure 2.

Water Supply and Discharge Measurement.--Water was supplied to the models from the laboratory's constant-head, recirculating system. A valve in the supply pipe was used to control the quantity of flow through the draft tubes. A maximum discharge of 2.2 cubic feet per second was used. The discharge was measured by means of a calibrated bend-meter in the supply pipe. The piezometric-head difference at the bend-meter was measured by means of an inverted, air-water, differential manometer.



Plan of Deflection Vanes, Looking Down, Showing Direction of Whirl for all Tests.

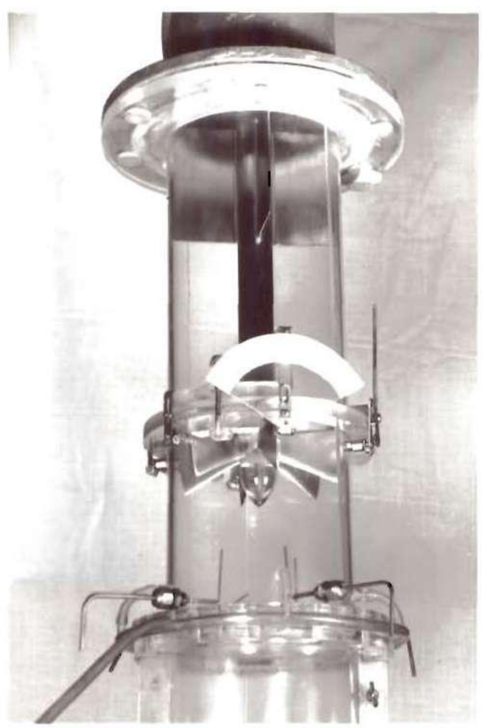


Figure 5. Deflection-Vane Section.

CHAPTER III

ANALYSIS AND PROCEDURE

Draft-Tube Efficiency.--Figure 1 is definitive for an analysis of flow through draft tubes. Assuming, first, that the flow past sections (1) and (3), at the throat and in the tailrace respectively, is axial and uniform, the average piezometric head, h, at both sections is indicated by the piezometric head at the boundaries. Futhermore, the average kinetic energy at section (1) is closely approximated by the velocity head computed on the basis of the average, axial velocity in the throat. Thus, from the one-dimensional (Bernoulli) energy and continuity equations,

$$h_3 - h_1 + H_L = \left[1 - \left(\frac{A_1}{A_3}\right)^2\right] \frac{V_1^2}{2g}, \tag{1}$$

where $(h_3 - h_1)$ is the difference in average piezometric heads between sections (1) and (3), (H_L) is the total head loss between the two sections (A_1) and (A_3) are the corresponding total cross-sectional areas, and $(V_1^2/2g)$ is the average kinetic energy at section (1).

The total head loss, H_L, between sections (1) and (3) consists of three parts--the total boundary friction loss, the expansion loss in the tube, and the expansion loss between the end of the tube and the tailrace. Each of these components is a function of $(V_1^2/2g)$. Thus, the total loss can be expressed in terms of a gross head-loss coefficient, C_L, as,

$$H_L = C_L \frac{V_1^2}{2g} \quad . \tag{2}$$

Substituting equation 2 in equation 1, solving for $(h_3 - h_1)$, and dividing by $(V_1^2/2g)$ yields

$$\frac{h_3 - h_1}{\frac{V_1^2}{2g}} = \left[1 - \left(\frac{A_1}{A_3}\right)^2 - C_L\right] , \tag{3}$$

where the left-hand member is a dimensionless piezometric-head ratio, henceforth given the symbol (η_e).

From dimensional analysis, (C_L) is a function of the Reynolds number, R, and the geometry of the fixed boundaries. It follows that the piezometric-head ratio, η_e, is also a function of (R) and geometry, or,

$$\frac{h_3 - h_1}{\frac{V_1^2}{2g}} = \eta_e = \phi \text{ (R, geometry)}. \tag{4}$$

For the assumed conditions (uniform, axial flow), (η_e) is a measure of the energy efficiency of the draft tube as a diffuser. This fact is made more apparent by substituting for $(h_3 - h_1)$ the equivalent expression from the Bernoulli equation, whence,

$$\eta_e = \frac{\frac{V_1^2}{2g} - \frac{V_3^2}{2g} - H_L}{\frac{V_1^2}{2g}} . \tag{5}$$

From equation 5 it is apparent that (η_e) is the ratio of the flow energy recovered by the draft tube to the energy available at the throat. The term $(V_1^2/2g)$ in equation 5 approaches zero for small values of the area ratio, as is true for most draft tubes.

The Influence of Whirl.--If it is assumed that the flow at section (1) contains a whirl component, equation 4 continues to be an adequate measure of draft-tube efficiency--subject to the following conditions. In the derivation of equation 4, it was assumed that the flow was axial and uniform. Thus, it was observed that the average piezometric heads at both sections (1) and (3) would be represented by the piezometric head at the boundaries. This fact is important, because only at the boundaries are the piezometric heads readily measured. With the added complication of whirl, the piezometric head at section (1) varies from a minimum at the center to a maximum at the wall. The magnitude of this variation depends on the magnitude and distribution of the tangential components of velocity in the cross-section. Unfortunately, neither the velocity pattern nor the piezometric-head distribution can be measured by ordinary means. At section (3), which is a vertical section in the tailrace, the residual whirl for all normal conditions is so small that it has negligible influence on the normal, hydrostatic pressure variation in the section.

A second complication resulting from whirl at section (1) concerns the denominator of the efficiency ratio, equation 4. Thus, when the flow contains a whirl component, (V_1) is the vector sum of the axial velocity and the tangential velocity. For a given discharge, therefore, the average kinetic energy at section (1) increases with increasing degrees of whirl.

Analysis of Efficiency Tests.--The only practicable method of measuring the piezometric head at section (1) in the laboratory is by means of wall piezometers. From the preceding discussion it is apparent that the piezometric head at the wall, h_1', will be in excess of the average piezometric head, h_1, except when the flow is axial (zero whirl). The effect of using (h_1') in equation 4 is to cause the numerator of the efficiency ratio to decrease with increasing degrees of whirl. On the other hand, the effect of using the average axial velocity, V_{1a}, instead of the total velocity, V_1, is to cause an increasing underestimation of the kinetic energy at the throat with increasing degrees of whirl. To an unknown degree, therefore, the use of (h_1') and (V_{1a}) instead of (h_1) and (V_1) in equation 4 would be partially compensating. More important, however, a coefficient which is proportional to (η_e), involving the easily determined quantities (h_1') and (V_{1a}), is quite adequate as a measure of the relative performance of different draft tubes at the same degree of whirl. It follows that such a simplified definition of efficiency can be used to demonstrate the influence of minor variations in form of a basic draft tube design. Thus, for the purposes of this investigation, the efficiency is conveniently defined as,

$$\eta = \frac{h_3 - h_1'}{\frac{V_{1a}^2}{2g}} = \phi\ (\underline{R},\ \text{geometry}). \tag{6}$$

Similitude Requirements.—It is apparent from equation 6 that two conditions must be satisfied in order that a draft tube model indicate correctly the efficiency of its prototype. These are, first, that the models be geometrically similar to the prototype and, second, that the Reynolds number, $\underline{R}$, be identical in model and prototype. In the present instance it is sufficient to know that the model results can be related to prototype performance. Actually, it is the purpose of these tests to compare the efficiency of model draft tubes of the same size but of variable form. The most significant geometric characteristic common to all models is the angle of the deflection vanes in the throat. Thus, in order to compare the results of tests made on tubes of different form, it is required that their deflection-vane angles be the same. Similarly, for comparison of model results it is necessary that tests be made at equal values of the Reynolds number. It is anticipated, however, that the efficiency ratio, like the coefficient of discharge for orifices and venturi tubes, will be independent of the Reynolds number under conditions of operation obtainable in the laboratory. Kindsvater and Randolph (16), seeking to simulate prototype results on a 1:25-scale model draft tube, demonstrated that the influence of the Reynolds number could be ignored at values of ($\underline{R}_1$) greater than $3x10^5$. Because models used for this investigation are equal in absolute size to those tested by Kindsvater and Randolph, it is reasonable to anticipate that the influence of the Reynolds number will also be similar. The proof in any case depends on the evidence obtained in the laboratory for the maximum possible range of model discharges.

<u>Experimental Procedure</u>.--The purpose of the deflection vanes shown in figure 5 was to simulate the whirl in the turbine discharge. As indicated in the photograph, this apparatus was equipped with a protractor to measure the angle between the deflection vanes and the axis of the tube. For uniformity of test conditions, the deflection vanes were always set so as

to produce a clockwise whirl, looking in the direction of flow.

For comparative purposes, all efficiency tests were made with the tailwater level (and therefore the piezometric head, h_3) equal. This standard value corresponded to a depth of tailwater equal to twice the height of the discharge opening of the draft tubes (original designs). The relative effect of varying tailwater levels was determined by a special series of tests made for each of the two basic models. The results of these tests are described subsequently.

Comparative efficiency tests were made at a constant rate of discharge. This rate, subsequently shown to be in the range where the influence of the Reynolds number is negligible, was 2.0 cubic feet per second. The corresponding velocity (V_{1a}) and Reynolds number ($\underline{R}_1$) at the throat section is approximately 10 feet per second and 5×10^5, respectively. The discharge was adjusted during each series of tests to maintain this constant flow for all settings of the deflection vanes.

A typical efficiency test series on a given model draft tube involved only one independent variable--the deflection-vane angle. Thus, with the discharge and tailwater level fixed at their standard values, the piezometric-head difference ($h_3 - h_1'$) was determined for several deflection-vane positions. The number of positions tested in each instance was determined by the requirements for drawing a smooth efficiency curve through the plotted test points.

All model changes involved the form of the downstream leg of the two basic draft-tube shapes. The manner in which these model modifications were made is described in the discussion of test results. In general, the efficiency characteristics of the original design of each of the two basic shapes was used as a basis for evaluating the influence of the various geometric modifications.

CHAPTER IV

DISCUSSION OF TEST RESULTS

Influence of Reynolds Number.--The influence of the Reynolds number, R, on the efficiency ratio, η , is shown on figures 6 and 7 for Models No. 1 and 2, respectively. A scale of values of the Reynolds number, based on an approximate value of kinematic viscosity equal to 1×10^{5} square feet per second, is shown below the scale of velocity. The Reynolds number (R_1), was varied from 1.0×10^{5} to 5.5×10^{5}, values which correspond to throat velocities of approximately 2 and 11 feet per second, respectively. The different curves shown on each figure correspond to various deflection-vane positions. All of the curves show a tendency for (η) to increase with increasing values of (R_1) up to about 3×10^{5}, then to remain constant for higher values of (R_1). As anticipated, the curves are similar to those which represent the effect of (R) on venturi-tube and orifice coefficients.

The results indicate that the relative effect of experimental changes in certain minor geometric features can be determined on the basis of model tests without attaining identical values of the Reynolds number, provided only that values of (R_1) in the model are equal to or greater than 3×10^{5}. The standard value of (R_1) selected for all subsequent tests was 5×10^{5}, corresponding to a velocity of 10 feet per second at the throat.

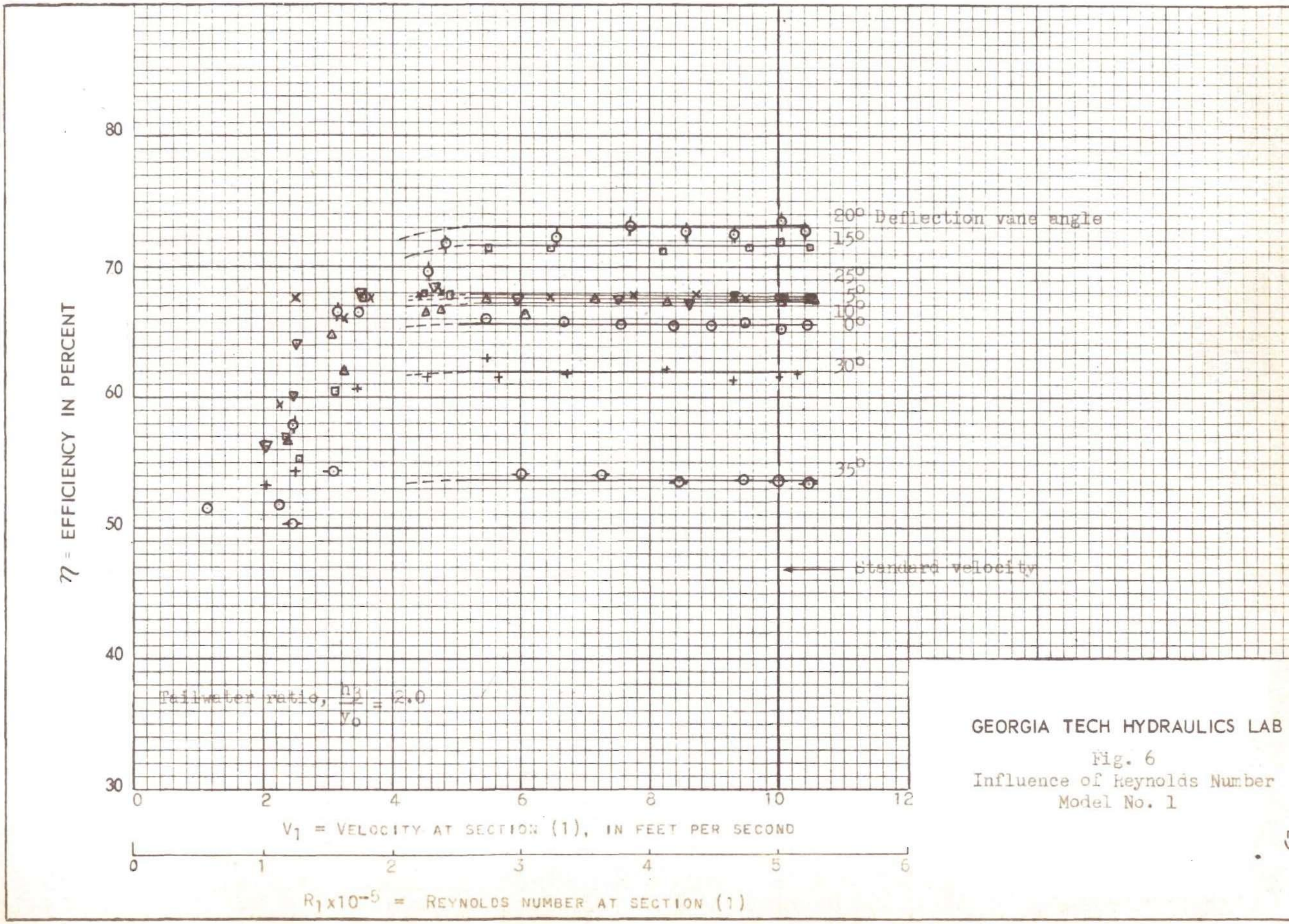


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Fig. 6
Influence of Reynolds Number
Model No. 1

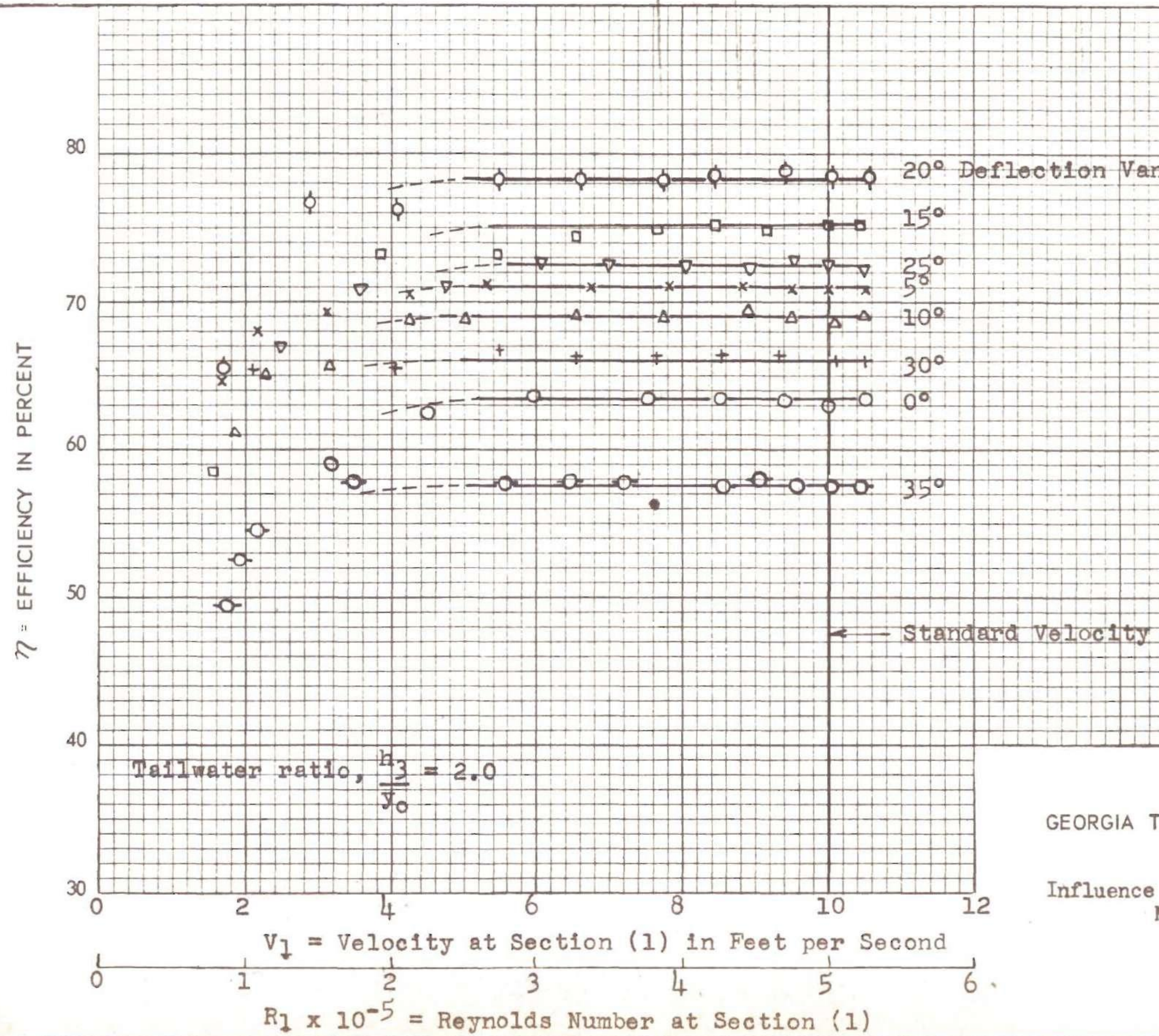

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Fig. 7
Influence of Reynolds Number
Model No. 2

Influence of Tailwater Level.--The relative influence of tailwater level was determined by a series of tests on the original design of each model. The results are shown on figures 8 and 9, where the measured efficiency is plotted as a function of the tailwater ratio, h_3/y_o. As shown on figure 1, (h_3) is the tailwater depth and (y_o) is the original height of the draft-tube opening at the outlet. Tests were made for a range of tailwater ratios from 1.0 to 2.5. For both models, the test results indicate a slight decrease in efficiency for increasing tailwater depths. Incidental to tests made for another purpose, it was demonstrated that a similar effect would result when the tailrace floor level was lowered below the bottom of the draft tube at the outlet. It is suggested that the decrease in efficiency in both instances is associated with the occurrence of horizontal-axis eddies surrounding the live stream at the outlet of the draft tube.

The standard value of the tailwater ratio selected for the subsequent tests was $h_3/y_o = 2.0$.

Efficiency of the Original Designs.--As a basis for comparison with tests made on modifications of the original models, the efficiency characteristics of Models No. 1 and No. 2 are shown on figures 10 and 11. On these figures the efficiency ratio, η , is plotted as a function of deflection-vane angle, with ($\underline{R}_1$) and the tailwater ratio constant and equal to their standard values.

The general form of the efficiency curves for the two models is similar. Both designs showed a marked deficiency at a deflection-vane angle of zero degrees, and both showed two points of peak performance

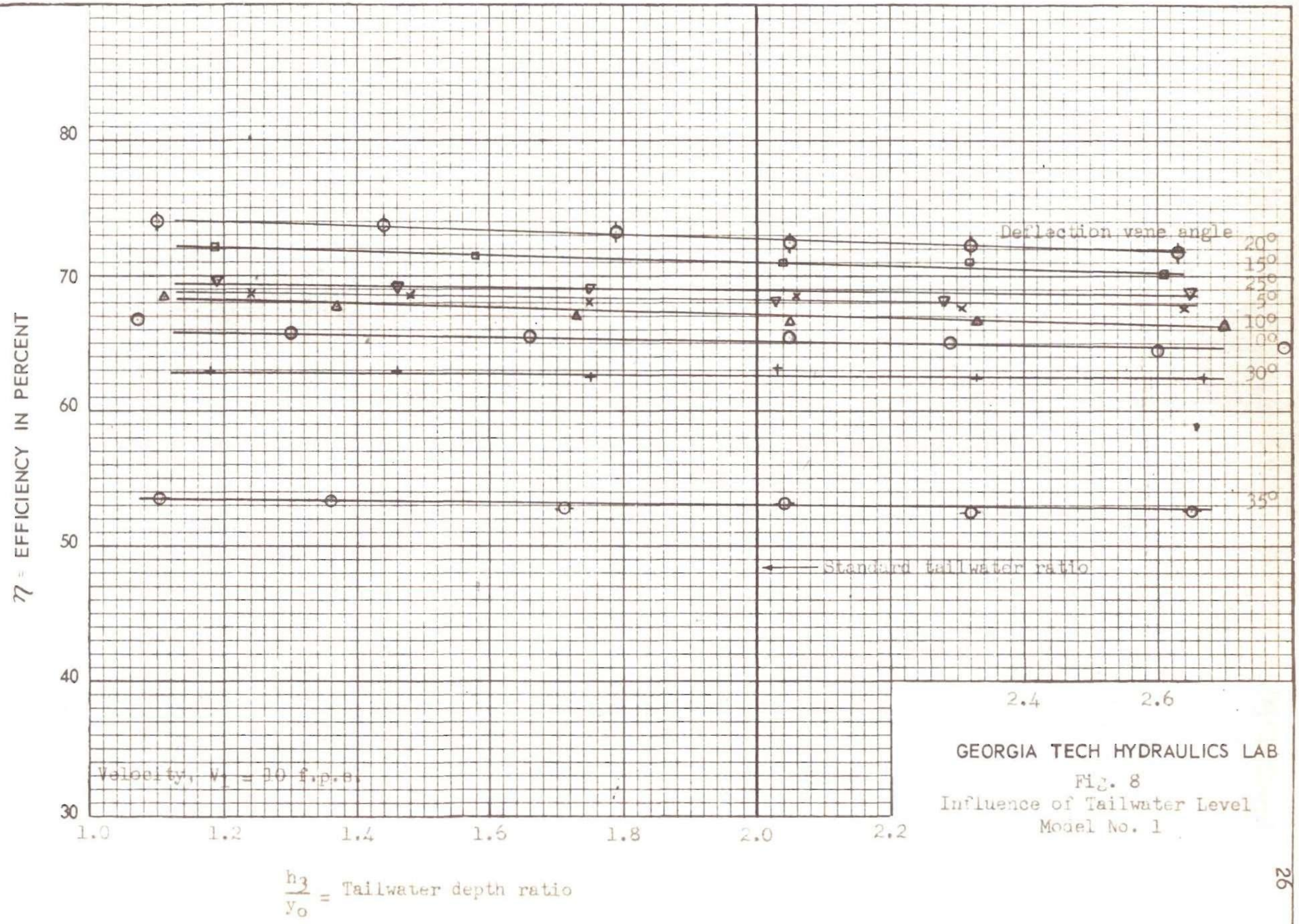

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Fig. 8
Influence of Tailwater Level
Model No. 1

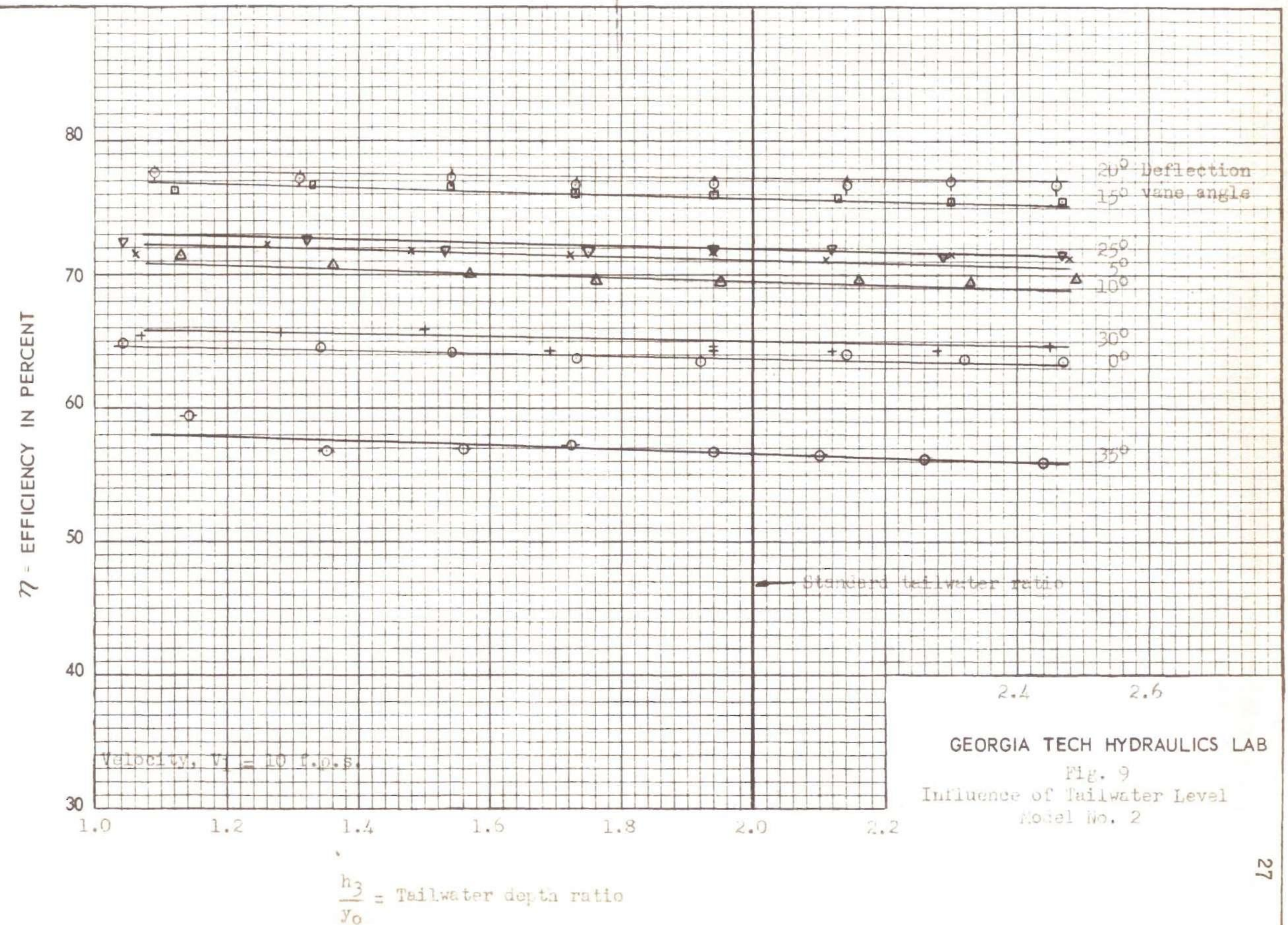

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Fig. 9
Influence of Tailwater Level
Model No. 2

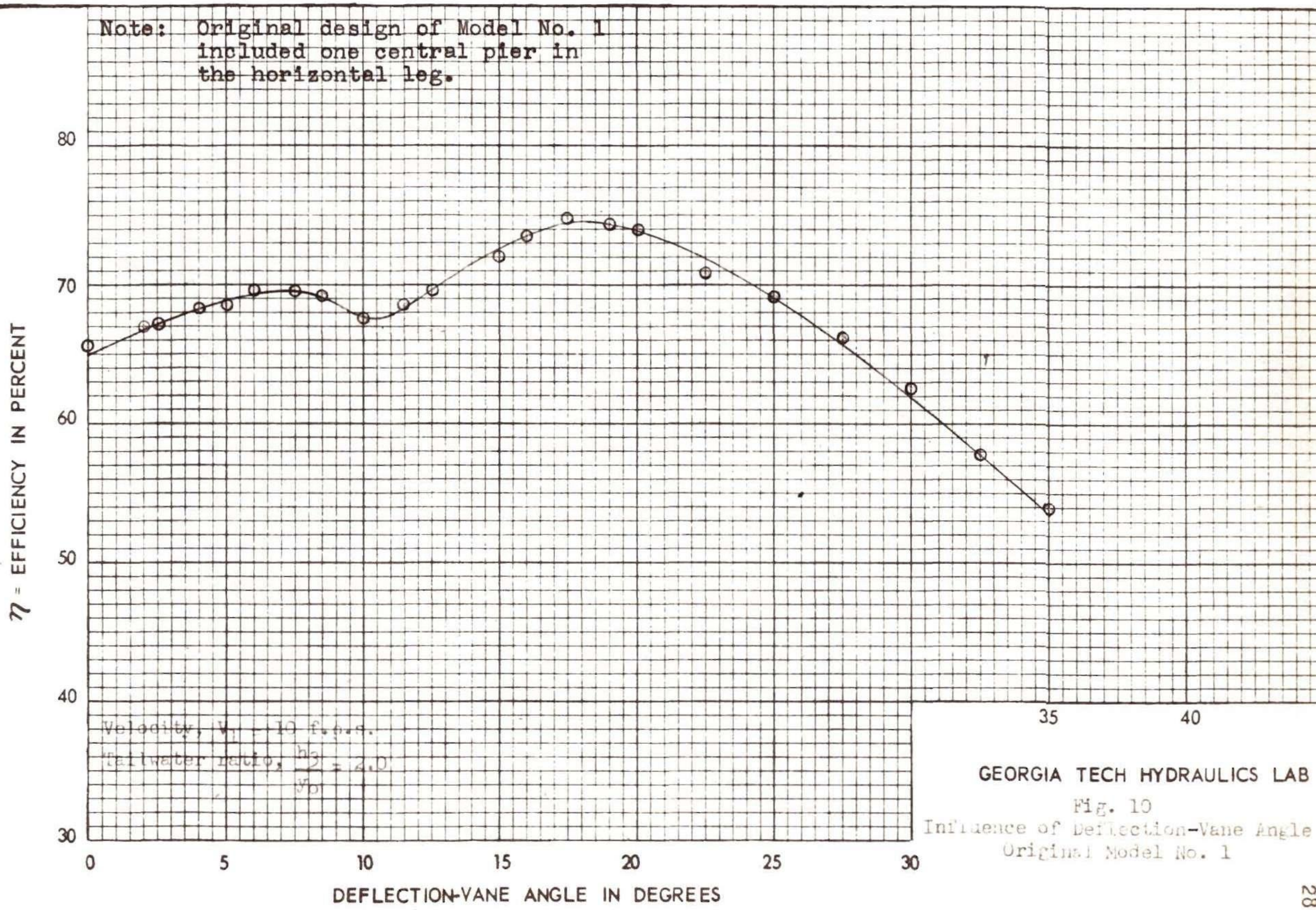

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Fig. 10

Influence of Deflection-Vane Angle

Original Model No. 1

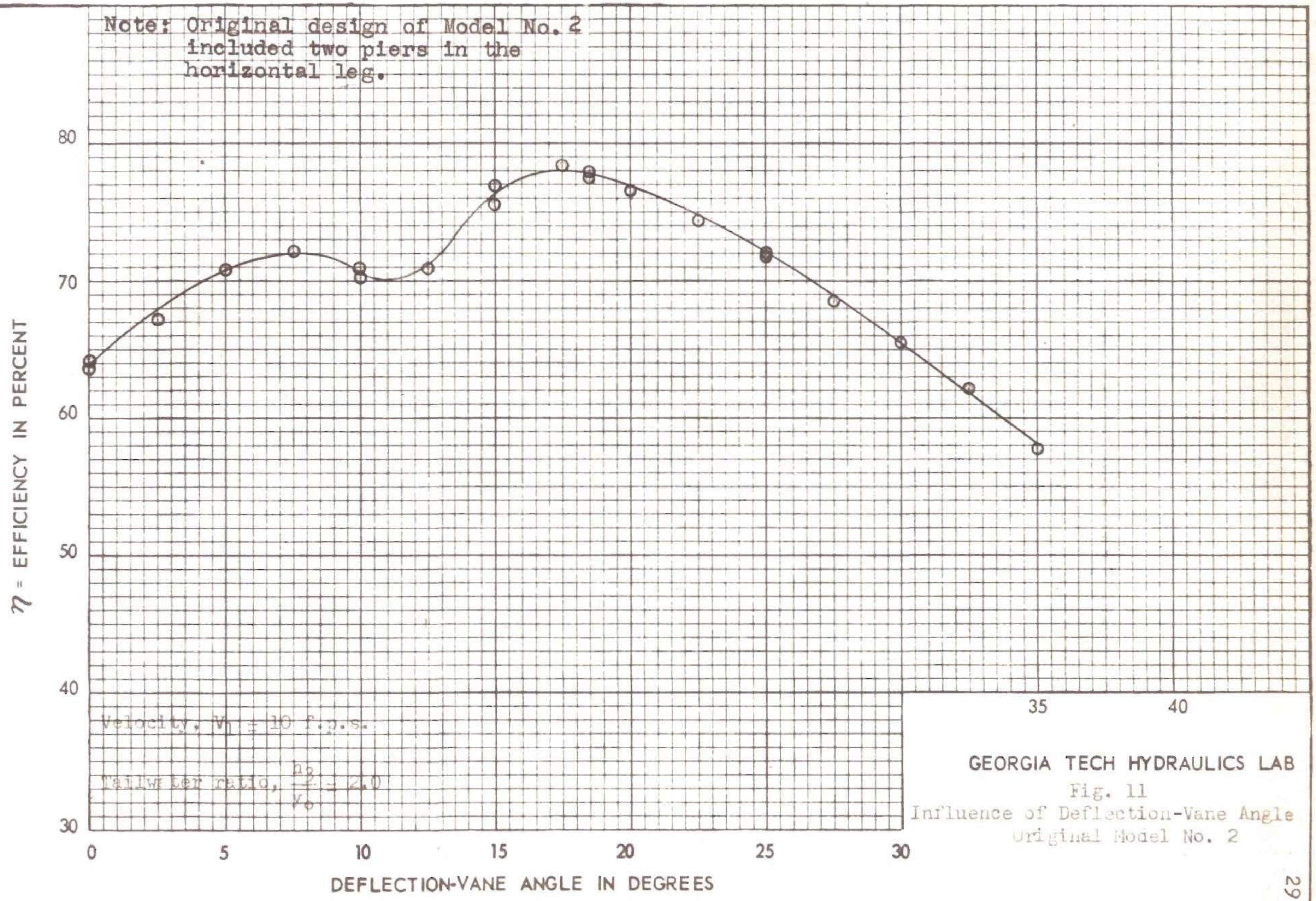

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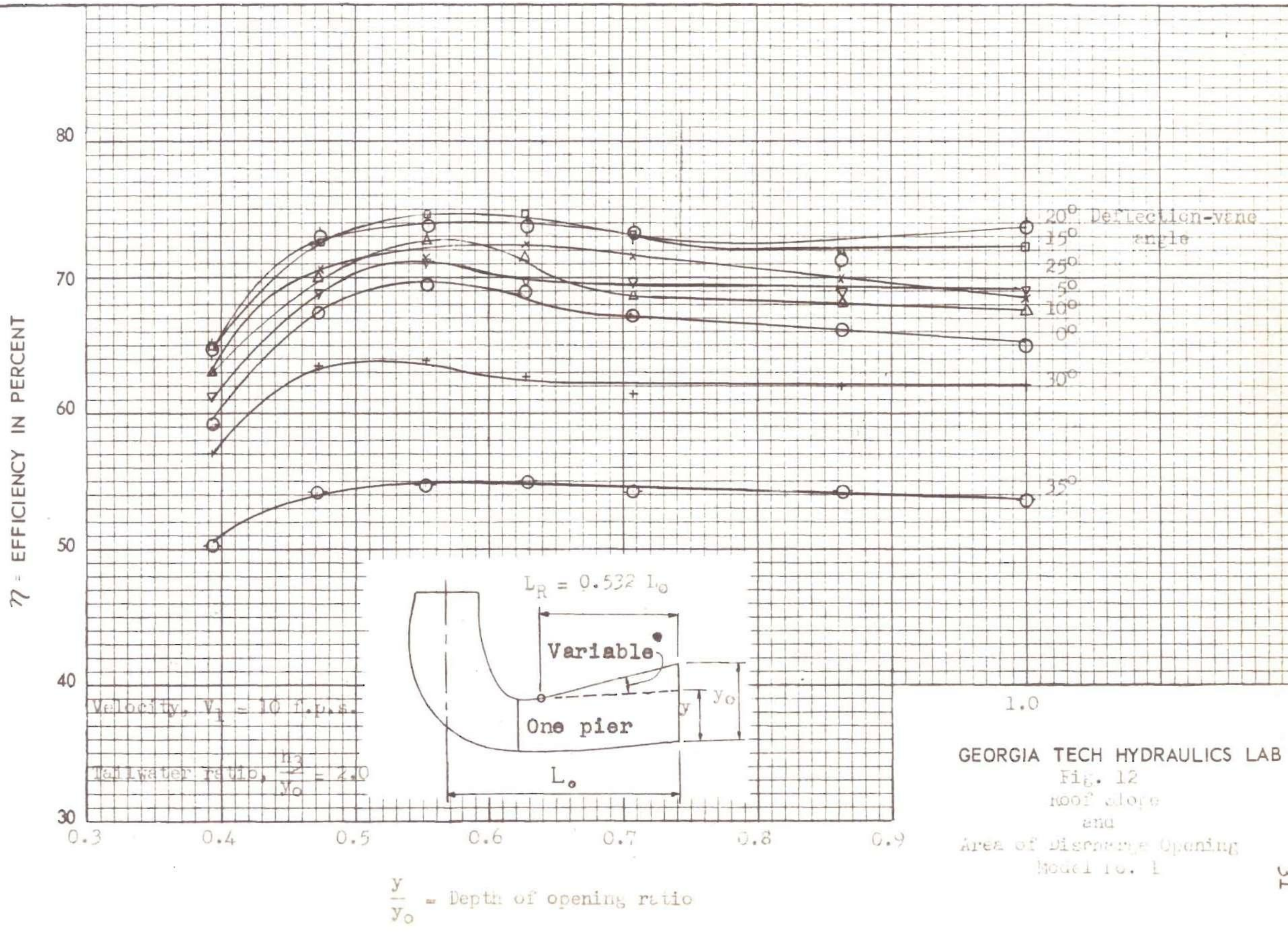
Fig. 11
Influence of Deflection-Vane Angle
Original Model No. 2

separated by a trough in the efficiency curve at about 10 degrees. The maximum efficiency of Model No. 1 was 74.5 per cent at 18 degrees. The maximum efficiency of Model No. 2 was 78 per cent at 17.5 degrees. The increase in efficiency with increasing whirl up to an angle of about 18 degrees is attributed to the influence of the normal acceleration in counter-acting tendencies toward separation in the vertical leg of the tube.

It should be observed, and kept in mind during the subsequent discussion, that the vanes in the deflection-vane section were visually aligned and that a deflection-vane angle of zero does not necessarily imply truly axial flow (zero whirl). It was principally for this reason that all tests were made with the whirl in one direction (clockwise, looking in the direction of flow).

Roof Slope and Area of Discharge Opening.--The combined influences of the slope of the roof in the horizontal leg and the area of the discharge opening are shown by the test results on figures 12 and 13. For these tests, an adjustable false roof was installed in both models. The false roof was a sheet-metal plate, feathered on the upstream edge to be tangent to the curve of the original roof at its lowest point. The position of the plate was controlled by bolts and lock nuts which supported the false roof from the original roof surface. The end-opening between the false roof and the original roof was closed.

Efficiency tests made on both models covered a range of discharge-opening depth ratios (y/y_o) from 0.4 to 1.0. Similar tests were made on both models for a full range of deflection-vane angles. From the test



GEORGIA TECH HYDRAULICS LAB
Fig. 12
Roof slope
and
Area of Discharge Opening
Model No. 1

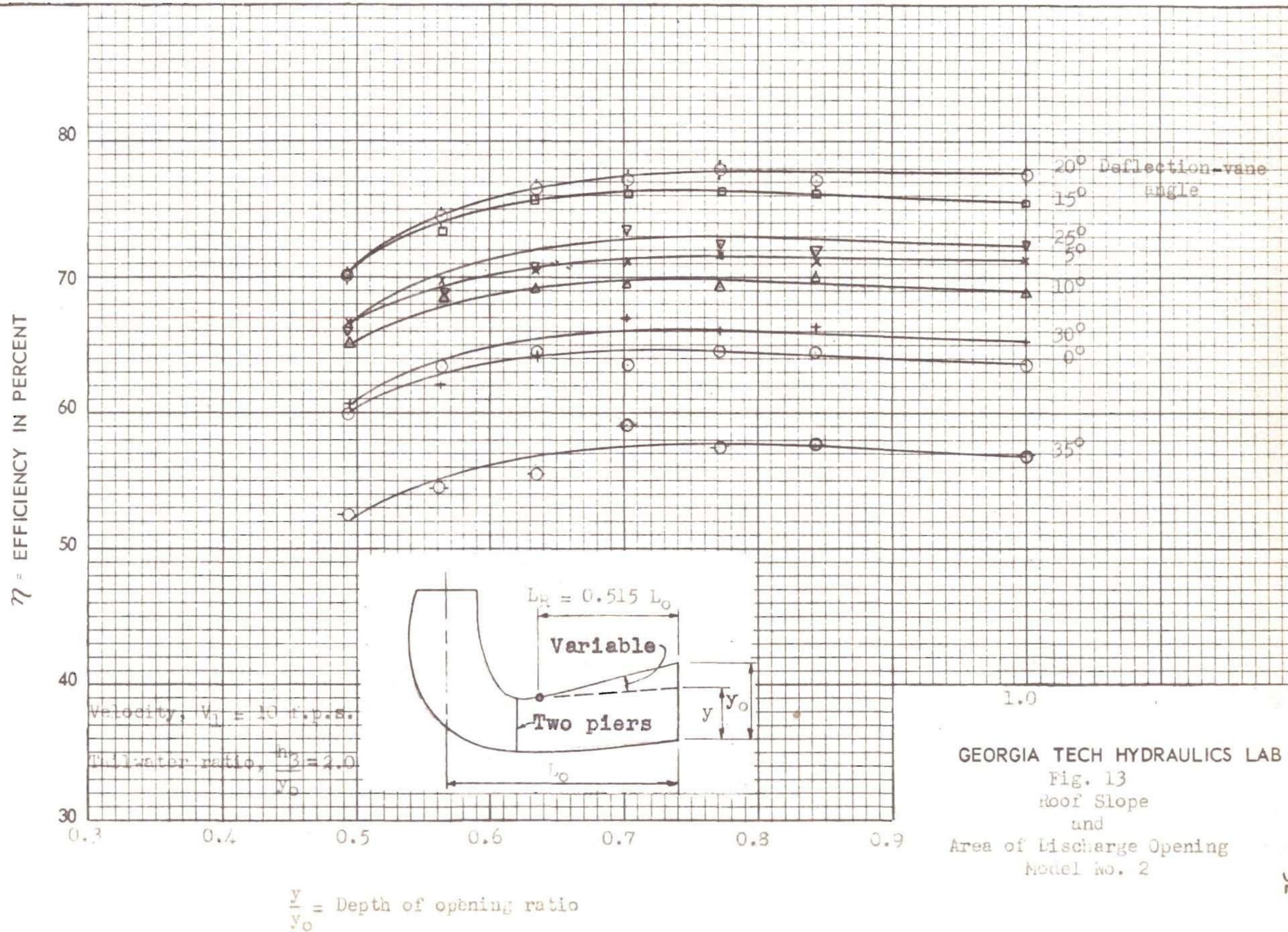


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Fig. 13
Roof Slope
and
Area of Discharge Opening
Model No. 2

results shown on figures 12 and 13, it is apparent that the roofs of both tubes could be appreciably lowered without causing a decrease in the hydraulic efficiency. In fact, for an opening ratio of 0.6, the efficiency of Model No. 1 was slightly better than it was for the original design. This increase in efficiency is associated with the conclusion, based on earlier observations of the flow pattern in transparent models of the same design, that the roof surfaces of the horizontal legs were not effective flow boundaries. It is suggested that a lowered roof, all other details of design remaining as in the original, more nearly comprises an effective boundary and, therefore, prevents the formation of energy-dissipating eddies in the upper areas of the horizontal leg of the tubes.

Number of Piers.--In order to determine the relative influence of the number of piers in the horizontal leg, comparative tests were made with none, one, and two piers in both models. In every case, the piers were located parallel to the centerline of the horizontal leg and were spaced so as to provide passages of equal width at the outlet. The piers for Model No. 1 were made identical with the single pier provided in the original design. As the noses of the two piers provided originally for Model No. 2 were slightly offset toward the center of the tube, however, a single pier with a symmetrical nose was constructed for this model. No other alterations in the shape or size of the original piers were made for this series of tests.

The results of the tests shown on figures 14 and 15 indicate that both models operate most efficiently with the number of piers contained

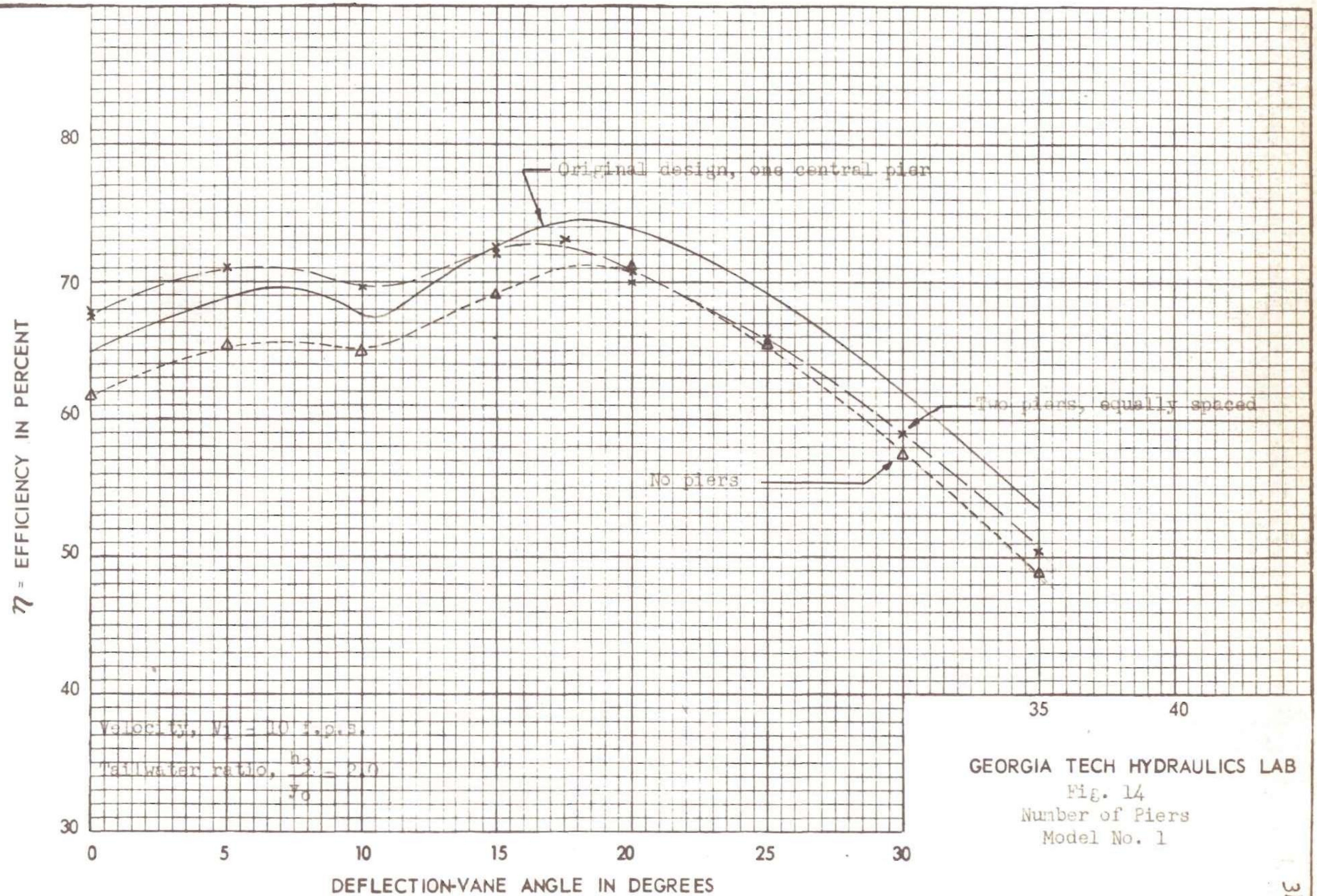

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Fig. 14
Number of Piers
Model No. 1

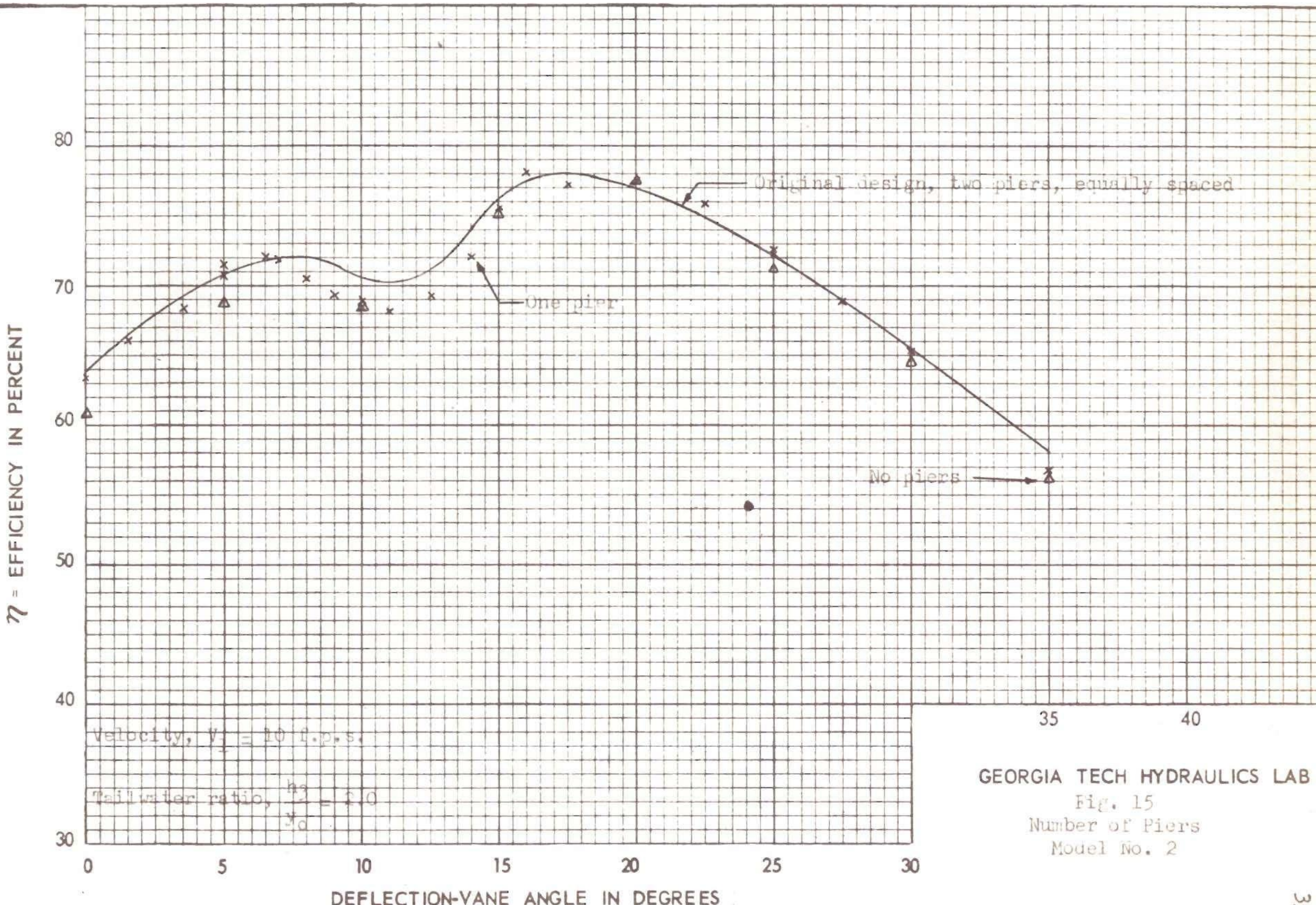

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Fig. 15
Number of Piers
Model No. 2

in the original design. Model No. 1, in fact, showed an appreciable reduction in hydraulic efficiency with no piers in the horizontal leg of the draft tube.

Lateral Position of a Single Pier.--It has been suggested that a single pier offset or turned at an angle with respect to the centerline of the horizontal leg might cause an increase in draft-tube efficiency at high degrees of whirl. Figures 16 and 17 show the results of tests made to determine the relative influence of the lateral position of a single pier. For these tests the axis of the pier was parallel to the axis of the tube. In order to avoid the effect of inherent differences in operation resulting from clockwise and counter-clockwise whirl, tests were made with the pier located in various positions on both sides of the center line, and the direction of whirl was clockwise for all tests.

It is apparent from figure 16 that no general benefit would result from offsetting a single pier in the horizontal leg of Model No. 1. Model No. 2 appeared to be relatively insensitive to changes in the lateral position of a single pier. It should be emphasized, however, that the original design for Model No. 2 included two piers.

Angular Position of a Single Pier.--Figures 18 and 19 show the relative influence of the angular position of a single pier rotated about its downstream end. Each point plotted on these figures represents a particular pier-position angle, θ , and deflection-vane angle. The test results are shown as differences in efficiency, computed with reference to the efficiency of the basic tube with a single, central pier. From figure 18, Model No. 1 appears to benefit slightly from a pier eccentricity

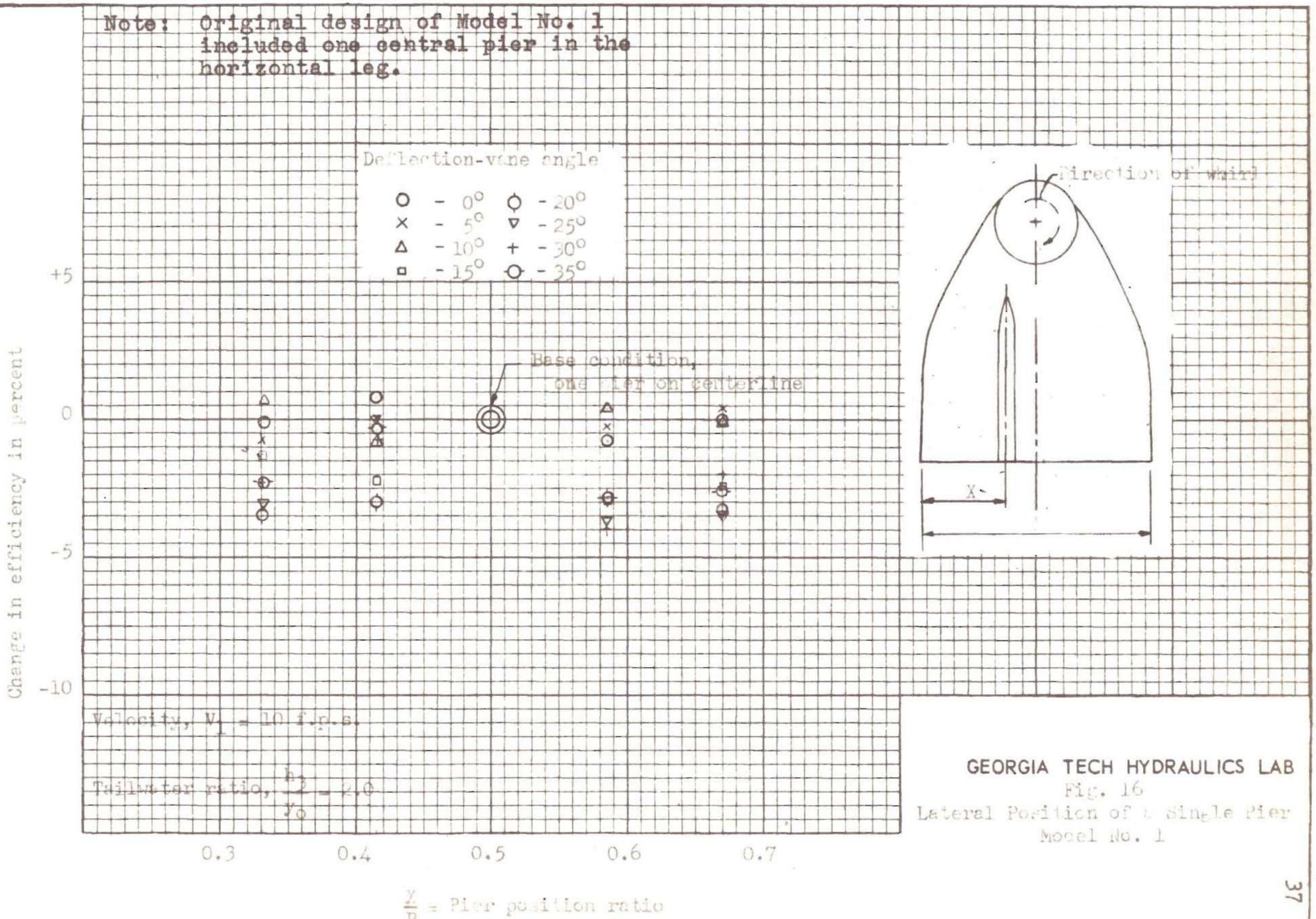

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Fig. 16
Lateral Position of a Single Pier
Model No. 1

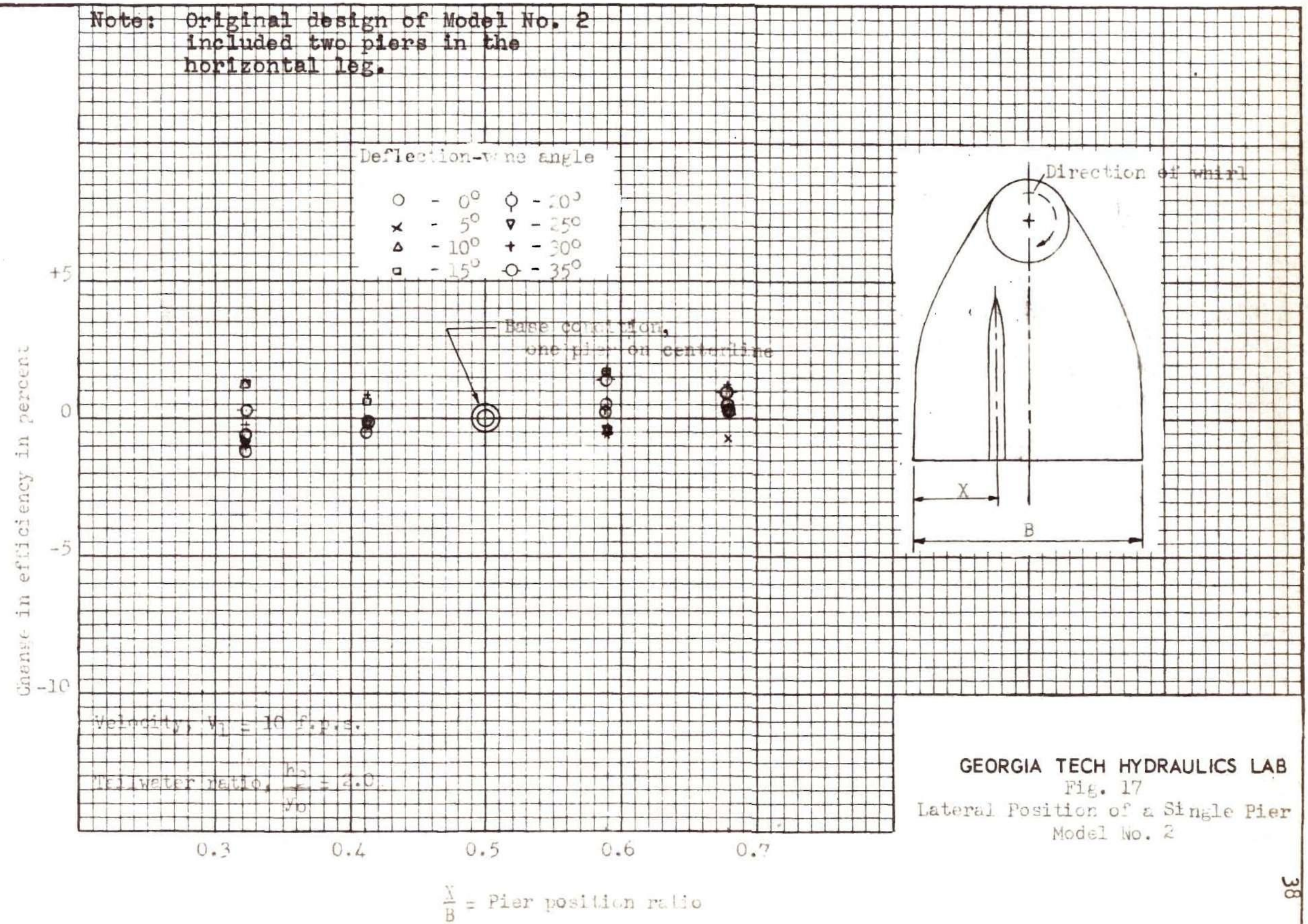

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Fig. 17
Lateral Position of a Single Pier
Model No. 2

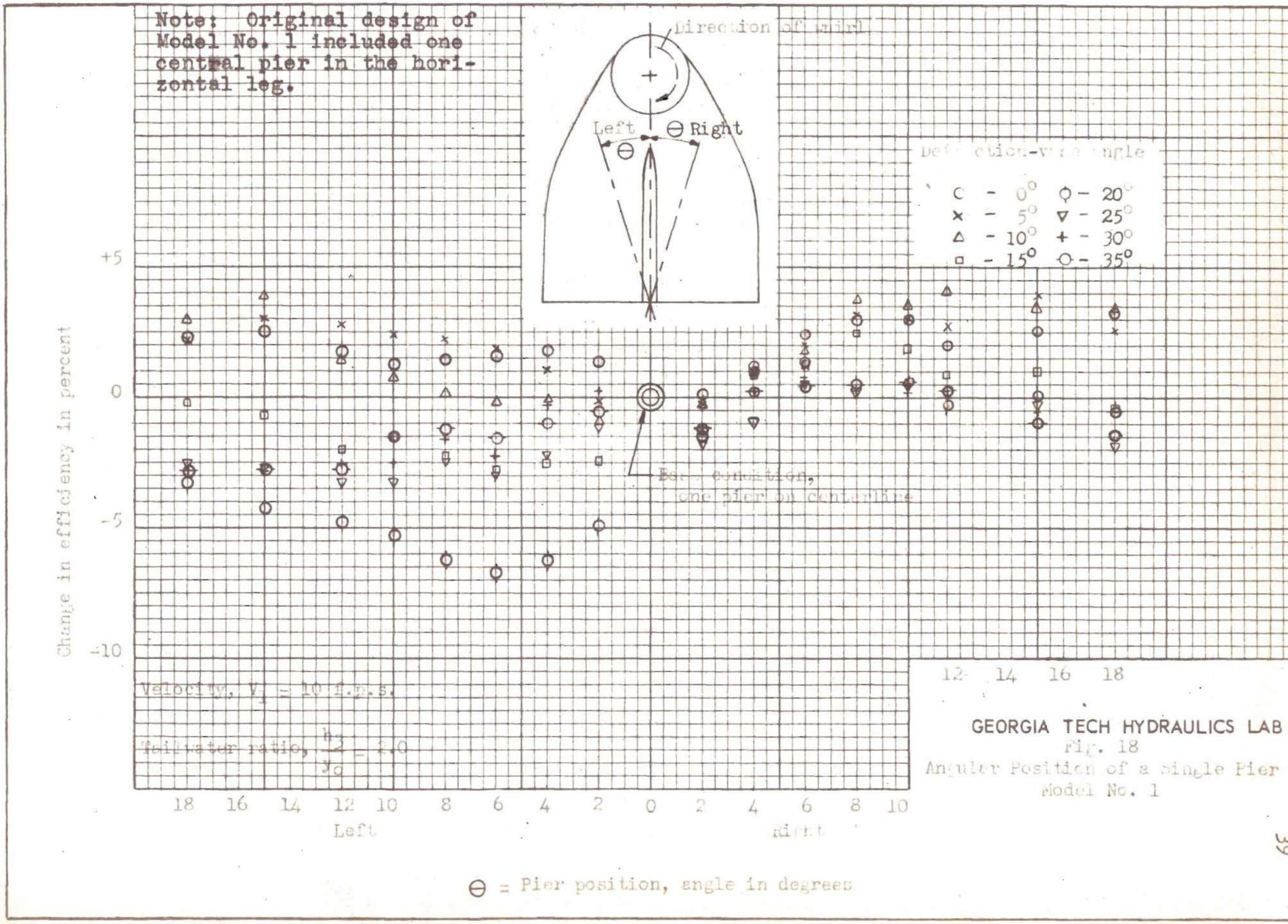

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Fig. 18
Angular Position of a Single Pier
Model No. 1

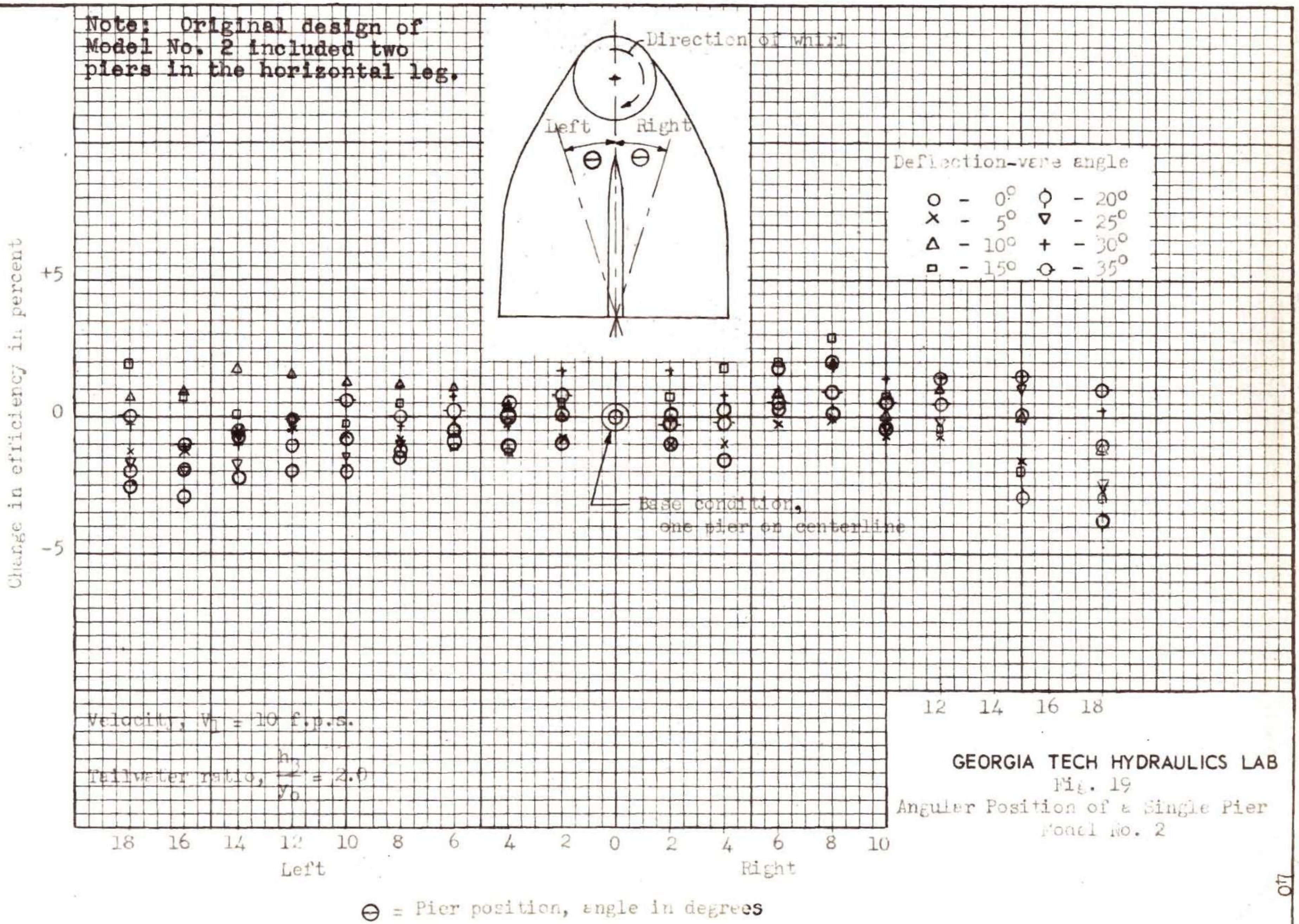

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Fig. 19

Angular Position of a Single Pier

Model No. 2

which corresponds to a moderate value of θ (right). For an eccentricity in the opposite direction (left), the overall performance of Model No. 1 is not benefited. Similarly, for Model No. 2, as shown on figure 19, the influence of the angular eccentricity of a single pier is negligible over the range of conditions tested. It should be noted, again, that the direction of whirl was clockwise for all tests.

Length of a Single Pier.--The results of tests made to determine the influence of the length of a single, central pier are shown on figures 20 and 21. For these tests on both models, pier-nose extensions were used to produce modified central piers of five different lengths, all of which were flush with the downstream end of the draft tubes. It follows that these tests were primarily an indication of the relative influence of the pier-nose extensions. The results are shown as differences in efficiency, computed with reference to the efficiency of the model with a single pier of the original length.

Figure 20 shows that the original pier for Model No. 1 is of optimum length. Figure 21 shows some large but inconsistent effects due to the length of a single pier in Model No. 2. For neither model, therefore, does it appear that a change in the length of a single, central pier would be beneficial.

Length of the Horizontal Leg.--In order to investigate the effect of a reduction in the length of the horizontal leg of the draft tubes, the flume was modified to permit the model tubes to be immersed in the flume instead of being bolted to the end of the flume. Thus, with all other

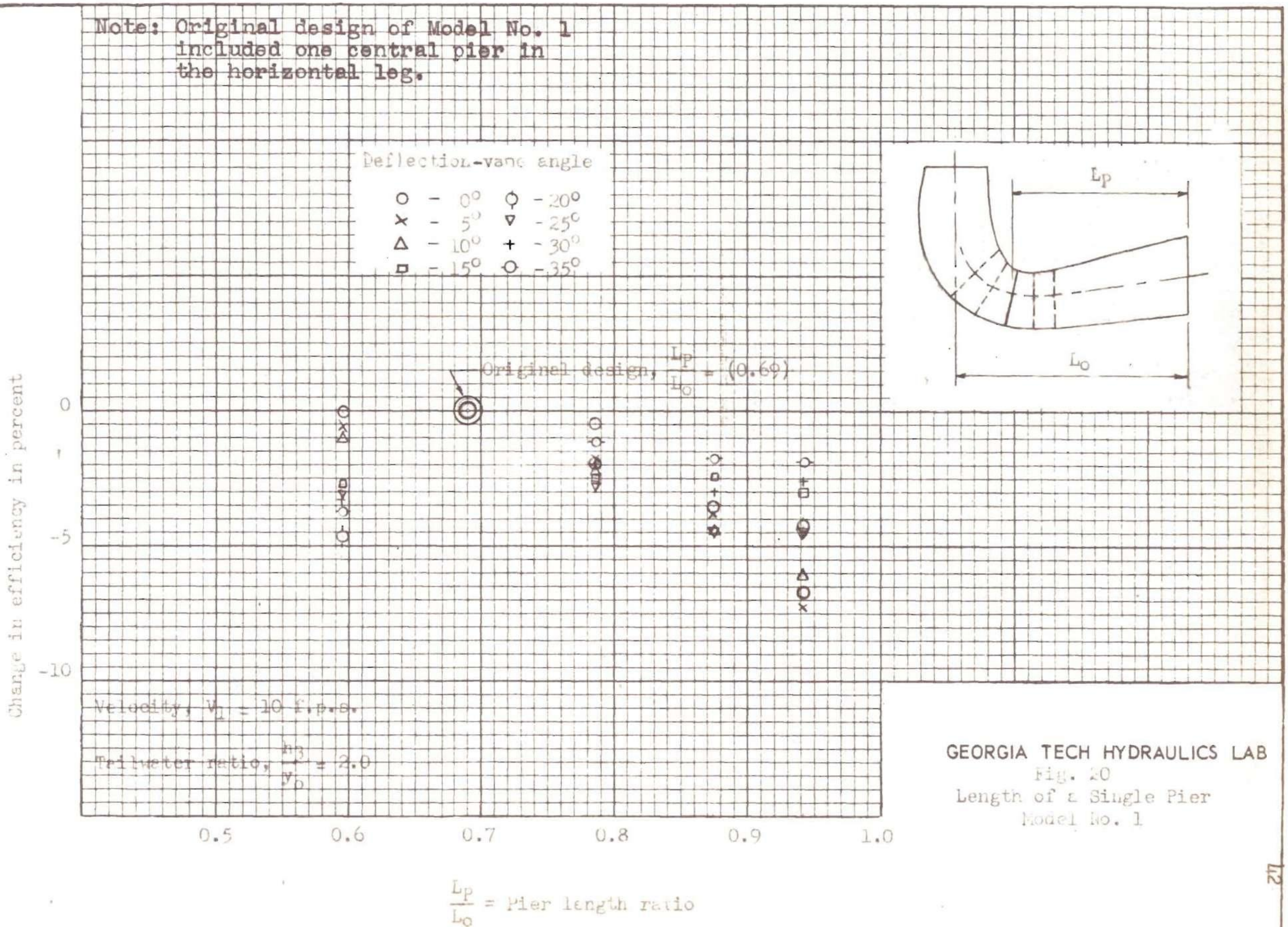

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Fig. 20

Length of a Single Pier

Model No. 1

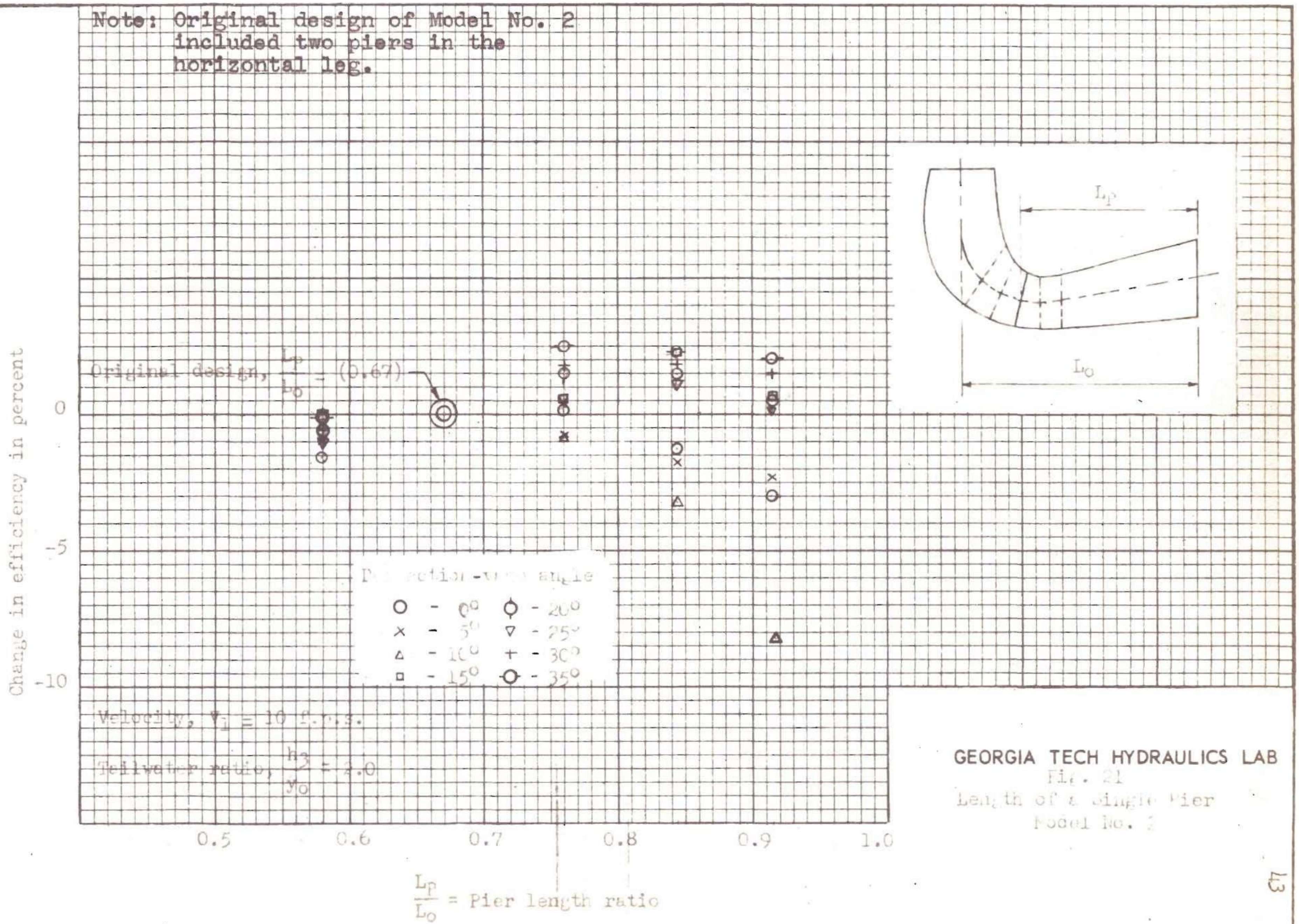

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Fig. 21

Length of a Single Pier

Model No. 2

details of design as in the original, and as a final operation in this investigation, the models were successively shortened by removing a portion of the horizontal leg. It was found necessary, as determined by comparison with tests made in the original set-up, to provide a false end-wall and floor at the outlet of the tubes located in the modified flume. Without these boundary surfaces, eddies surrounding the live jet at the outlet caused an appreciable reduction in the hydraulic efficiency of the tubes.

It is apparent from figures 22 and 23 that the effect of a reduction in length of either draft tube is a reduction in their hydraulic efficiencies. In terms of plant efficiency, however, the indicated reductions in efficiency are not so great as to preclude a consideration of this means of accomplishing a reduction in the cost of the draft tubes.

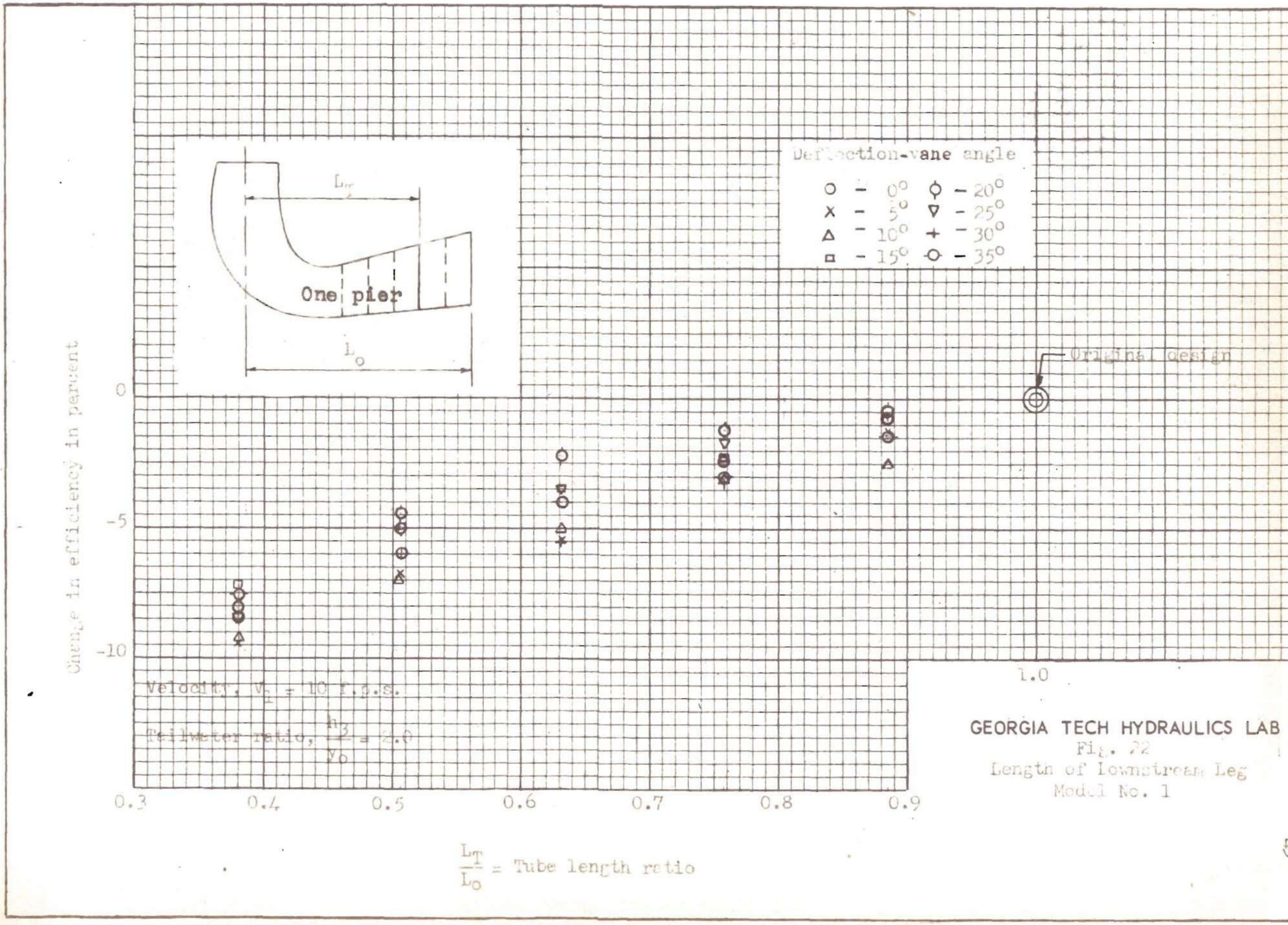

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Fig. 22
Length of Downstream Leg
Model No. 1

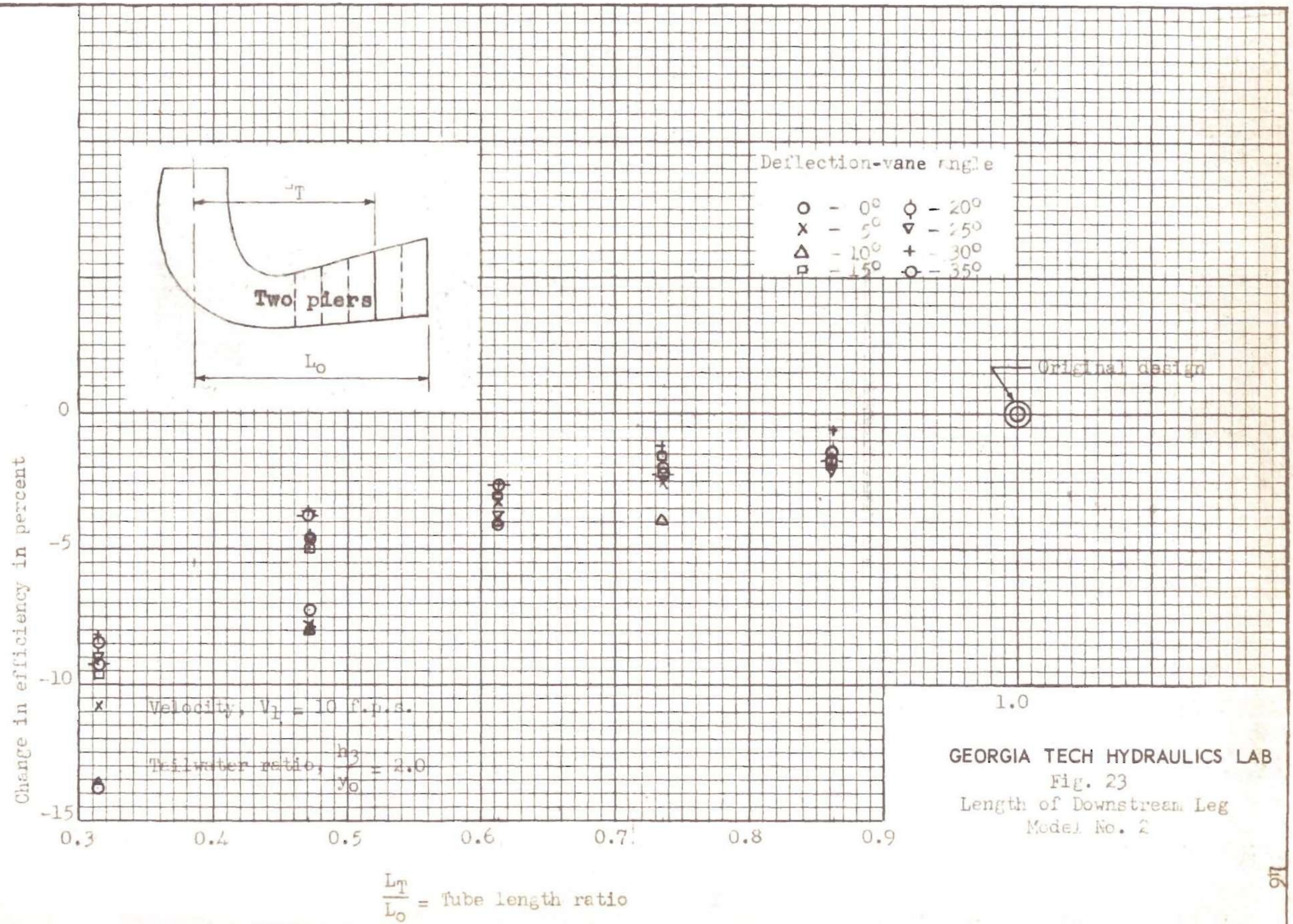

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Fig. 23

Length of Downstream Leg

Model No. 2

CHAPTER V

CONCLUSIONS

1. The relative effect of experimental changes in critical geometric features of hydraulic-turbine draft tubes can be determined from hydraulic model tests.

2. Dynamic similarity of flow patterns in draft tubes can be achieved without attaining identical values of the Reynolds number, provided only that (R) is greater than an experimentally determined, minimum value.

3. Tailwater level has an inappreciable influence on draft-tube efficiency over the normal range of operating conditions.

4. In general, the efficiency of a draft tube as a hydraulic diffusor is a maximum for moderate degrees of whirl in the turbine discharge. Certain types of tubes, including the two tested in this investigation, show a marked drop in efficiency between two points of peak performance.

5. For both draft-tube designs tested, a very slight change in performance was achieved when the roof of the horizontal leg was appreciably lowered, all other features remaining constant as in the original designs.

6. At least one pier in the horizontal leg of either draft tube had a beneficial effect on the hydraulic efficiency of the tube. For Model No. 2, two piers, as contained in the original design, are better than one.

7. Pier eccentricities, involving the lateral position of a single, axial pier and the angular position of a single pier, do not show promise of improved performance as compared with the usual symmetrical arrangement.

8. Pier-nose extensions added to single, central piers of the original length cause, in general, a decrease in draft-tube efficiency.

9. Any reduction in the length of the horizontal legs of both tubes tested causes a reduction in the efficiency of the draft tube.

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