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Technical Paper No. 25, Series B

EXPERIMENTAL STUDIES OF PNEUMATIC AND HYDRAULIC BREAKWATERS

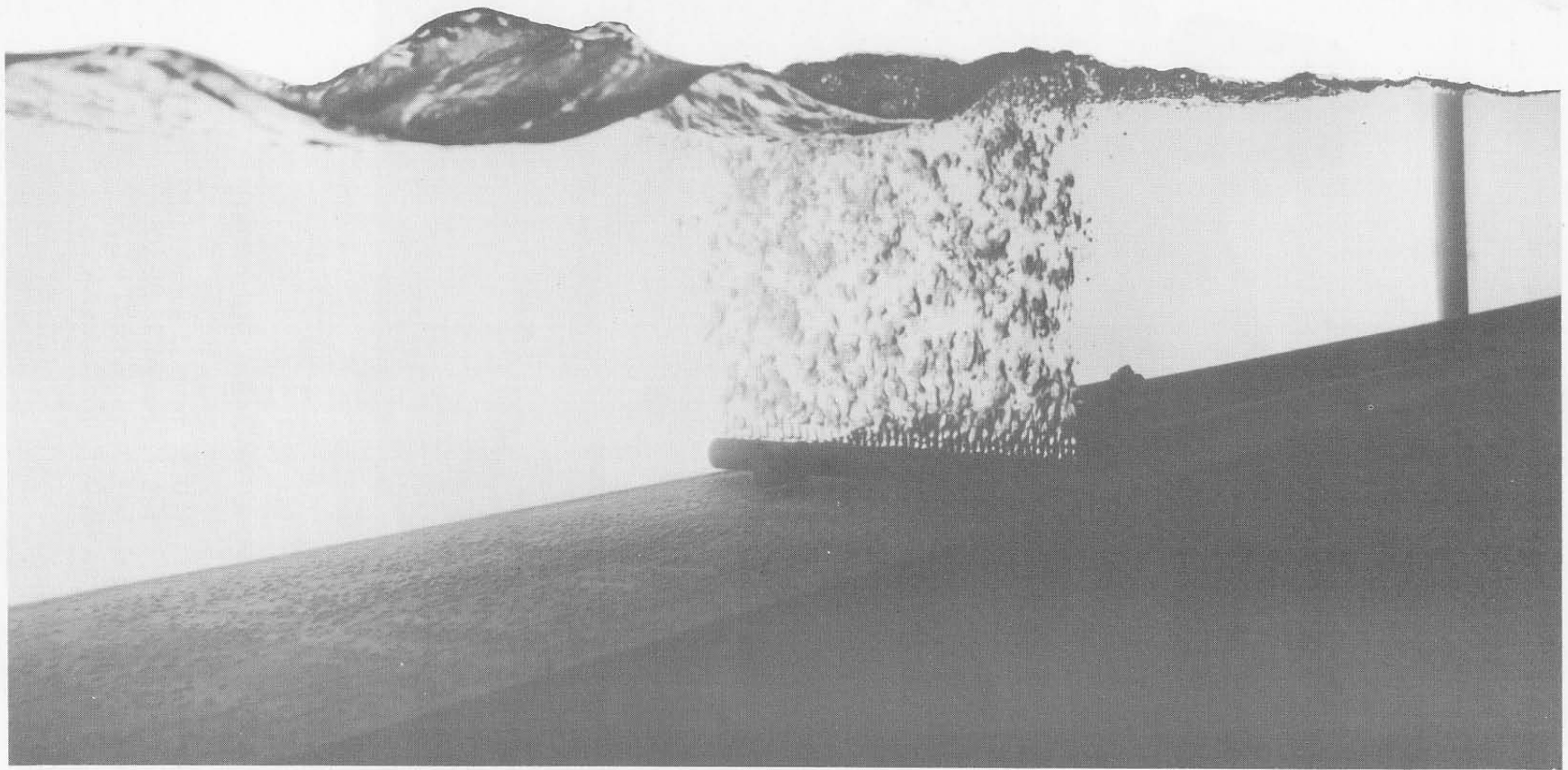
by

LORENZ G. STRAUB, C. E. BOWERS, and ZAL S. TARAPORE



August 1959

Minneapolis, Minnesota



PNEUMATIC BREAKWATER IN ACTION

Wave Length: 1.78 ft
Depth of Water: 1.0 ft
Air Discharge: 0.020 cfs per ft
Attenuation: 70 per cent



HYDRAULIC BREAKWATER IN ACTION

Wave Length: 8.0 ft
Depth of Water: 4.5 ft
Jet Discharge: 0.167 cfs per ft
Attenuation: 100 per cent

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August 1959
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UNIVERSITY OF MINNESOTA
ST. ANTHONY TAILORING LABORATORY
ROBERT E. STRAVA, DIRECTOR

EXPERIMENTAL STUDIES OF
PNEUMATIC AND HYDRAULIC
BREAKWATERS

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P R E F A C E

The experimental and analytical studies described herein were performed at the St. Anthony Falls Hydraulic Laboratory of the University of Minnesota under Contract Nonr 710(10) with the Office of Naval Research. The primary objective of the studies was an evaluation of the performance and behavior of pneumatic and hydraulic breakwaters.

Initial phases of the program involved a survey of literature and preliminary small-scale tests of both pneumatic and hydraulic breakwaters. The results of this work were presented in St. Anthony Falls Hydraulic Laboratory Project Report No. 46 [13].*

A second phase of the program involved both small- and large-scale studies of hydraulic breakwaters. Data were obtained on power requirements and attenuation for various wave conditions of interest. St. Anthony Falls Hydraulic Laboratory Project Report No. 51 presented the results of this phase. The more significant results were summarized in a paper presented at the Sixth Conference on Coastal Engineering [25].

The third and final phase of the studies involved both small- and large-scale studies of pneumatic breakwaters. Objectives of the study included the procurement of large-scale data on the performance of the pneumatic system and an evaluation of possible scale effects.

The project was proposed by and under the direction of Dr. Lorenz G. Straub, Director of the St. Anthony Falls Hydraulic Laboratory. C. E. Bowers supervised the experimental program and correlated the three phases of the studies. The initial small-scale tests were conducted by J. M. Wetzell. Subsequent large- and small-scale tests of hydraulic breakwaters were conducted by John B. Herbich. The concluding phase of the program, large- and small-scale tests of pneumatic breakwaters, was conducted by Zal S. Tarapore. Manuscript preparation was performed by Marjorie Summers under the general direction of Loyal A. Johnson.

*Numbers in brackets refer to corresponding numbers in the List of References on p. 23.

A B S T R A C T

Experimental studies were conducted on two similar models of both pneumatic and hydraulic breakwaters, having a length ratio of 4.5:1.

Tests of the pneumatic system indicated that the horsepower requirements for a given percentage of attenuation depended only on the wave length, the submergence of the manifold, and the depth of water. Multiple-manifold breakwaters with different spacings between manifolds were tried and found to be of no particular advantage over the one-manifold system. An intermittent bubbler device was also tested very briefly, showing very little difference from the one-manifold data.

Tests of the hydraulic system indicated that power requirements varied with wave steepness as well as wave length. Orifice area was a very important parameter as this affected the discharge requirements and the required size of supply piping.

The power requirements of the pneumatic system are somewhat less than the hydraulic for average values of wave steepness, but the maximum attenuation achieved was less than the hydraulic.

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L I S T O F S Y M B O L S

- a_j - Area of jets per foot width (hydraulic breakwater).
- a_c - Area of channel per lineal foot of breakwater or one times the depth.
- A - Wave attenuation $(1 - H_T/H_I)$.
- C - Wave celerity.
- d - Depth of water.
- d_L - Submergence of manifold (pneumatic breakwater).
- D - Diameter of supply pipe.
- E - Energy.
- F - Projected area of air bubbles.
- F_n - Froude number C/\sqrt{gd} .
- g - Acceleration of gravity.
- h - Thickness of surface current.
- h_L - Head loss in supply pipe.
- H - Wave height.
- H_I - Incident wave height.
- H_T - Transmitted wave height.
- hp_L - Horsepower losses in supply pipe.
- k - Head loss coefficient.
- L - Wave length.
- p - Pressure.
- P_I - Power in incident wave.
- P_j - Power in hydraulic jets.
- P_T - Power in transmitted wave.
- q - Discharge per foot width of breakwater.
- Q - Total discharge.
- r - Ratio of energy extinguished by breakwater to the initial energy of the wave.

t - Time.

U - Velocity of the surface current.

v - Horizontal velocity of water particles due to the orbital motion of waves.

V - Velocity in supply pipe.

w - Specific weight.

y - Submergence of manifold (hydraulic breakwater).

z - Vertical distance measured from the water surface.

$$\eta = \text{Efficiency} = \frac{P_I - P_T}{P_j}.$$

ξ - Parameter in Taylor's surface current theory.

ρ - Density of water.

$$\Phi = \text{Dimensionless horsepower ratio} = \frac{\text{hp/ft}}{\rho g^{3/2} L^{5/2}}.$$

$$\Psi = \text{Dimensionless discharge ratio} = \frac{q}{L \sqrt{gd}}.$$

ω - Dimensionless parameter used in Taylor's theory.

EXPERIMENTAL STUDIES OF
PNEUMATIC AND HYDRAULIC BREAKWATERS

I. INTRODUCTION

The problem of protecting selected areas from the destructive power of waves is a difficult one and structures designed for this purpose are usually quite massive and expensive to build. As a result there is continuing interest in new methods or designs which may assist in a solution of some variations of the basic problem.

The pneumatic breakwater was conceived about 50 years ago as a device which would attenuate ocean waves by a curtain of air rather than by reflection or absorption of energy by the massive structures customarily used. It consists of a perforated pipe, usually on the harbor floor, through which compressed air is forced. As the air bubbles rise they impart a drag to adjacent water particles resulting in an upward motion of the air-water mixture. When this mixture reaches the surface, the air escapes, while the flow of water branches into two horizontal currents, as shown in Fig. 1. While the turbulence induced by this system produces some attenuation of the waves, it is usually considered that one of the horizontal currents, opposing the incoming wave, results in breaking of the wave and consequent turbulent diffusion of the incident wave energy. For best results the axis of the perforated pipe should be normal to the direction of wave advance.

The hydraulic breakwater is similar in principle to the pneumatic with the exception that the circulation system is created by jets of water. For best results a perforated pipe or manifold is placed at or near the water surface with the jets of water directed horizontally against the incoming wave. The hydraulic system usually produces only a single circulating current as opposed to the two currents of opposite direction created by the pneumatic system. However, an opposed jet system can be used to reduce thrust on the manifold.

The studies described herein had as their primary objective a determination of the overall performance of both pneumatic and hydraulic breakwaters, including a determination of power requirements, wave attenuation and optimum geometry of the systems. Tests were performed on both large and small

models to assist in extrapolating the test results to prototype installations.

II. REVIEW OF PREVIOUS STUDIES

The initial concept of the pneumatic breakwater is attributed to Phillip Brasher who obtained a patent in 1907 on a method for attenuating waves by forcing compressed air through a submerged perforated pipe. Subsequently several prototype installations were constructed and on the basis of somewhat fragmentary data, described as successful [1, 2]. In the period from 1930 to 1950 some model studies were performed but the results were incomplete and in some instances contradictory. One of the most significant advances during this period was the development by Taylor [7] of a theory for the pneumatic breakwater. This was subsequently modified and published in 1955 [14]. One part of the theory was concerned with the theoretical prediction of the surface-current velocity required to stop waves of given length. The development was based on the assumption of a uniform velocity distribution in a surface current of specified thickness. The second part provided a method of computing the surface currents induced by a curtain of air bubbles.

In 1950 Carr [8] reported results of mobile breakwater studies, including tests with pneumatic and hydraulic breakwaters. His conclusions were that the power requirements were excessively large; however, it appeared that his primary interest was in long, shallow waves, as most of his tests were concerned with L/d values in excess of 2. Conceivably, there are applications involving deep-water waves ($L/d < 2$).

In 1954 Evans [15] prepared a report for the British Transport Commission describing tests on both pneumatic and hydraulic breakwaters. He presented the data on the surface-current velocities required to stop waves of various lengths and provided a general indication of the horsepower requirements. His conclusions were that the complete damping of waves in excess of 200 ft long was probably uneconomical. Initially, he conducted a series of tests on both the pneumatic and hydraulic systems and concluded that both systems would create similar surface currents and attenuate given waves. He then concentrated on hydraulic breakwaters, and apparently the major portion of his data are based on the hydraulic type. He also presented some comparisons with the first part of Taylor's theory.

Evans also discussed an installation of a pneumatic breakwater at Dover, England, to protect the inner dock gate from excessive swell while an outer gate was being repaired. The observations were not very conclusive, probably due to formation of a standing wave or other conditions. However, the operating staff reported that the dock gate was successfully opened and shut by its motor on five or six occasions in wave conditions that would have prevented this action if the pneumatic breakwater had not been operating. Another source [12] reported that waves up to 4 ft high outside the breakwater were reduced to 9 in. high by the time they reached the inner gate.

In 1954 Wetzel [13] prepared a report on the results of preliminary small-scale tests at the St. Anthony Falls Hydraulic Laboratory. Comparisons of the results were made with Taylor's theory and some of Carr's experimental data.

During the period 1955-58 Kurihara [19] conducted model and prototype tests on pneumatic breakwaters in Japan. His major conclusion was that the mechanism of wave attenuation by the model pneumatic breakwater was due chiefly to the surface current whereas the prototype systems were dependent primarily on the turbulence accompanying the surface current. These observations were based on the fact that better efficiencies were achieved in the prototype when the two were compared on the basis of Froude's law.

In 1956 the results of studies on hydraulic breakwaters at St. Anthony Falls Hydraulic Laboratory were reported by Herbich et al [20]; these results were summarized by Straub et al in the Proceedings of the Sixth Conference on Coastal Engineering [25], and are also included in this report.

In 1957-58 several reports were issued on studies performed at the University of California. Snyder [22] reported the results of studies on hydraulic breakwaters in a water depth of 1 ft. The results are in general agreement with those obtained at the St. Anthony Falls Hydraulic Laboratory [20], with the exception of the effect of wave steepness. Snyder concluded that performance did not vary with wave steepness but the narrow range covered by his studies ($H/L = 0.047$ to 0.068) appears inadequate for an evaluation of this effect. Snyder [24] also studied the performance of the hydraulic breakwater when located over a submerged reef; this geometry resulted in a substantial improvement in performance.

Dilley [23] obtained attenuation data on a 1:86.5 scale model of a hydraulic breakwater mounted on a Liberty ship. In order to reduce the thrust on the ship leeward jets were also installed.

Williams [26] studied the comparative performance of four geometrically similar models of the hydraulic breakwater, in a study of scale effect. The results were different for each of 4 values of L/d tested and he concluded that the data were inconclusive as far as scale effect was concerned. Using a Froude extrapolation from the smallest model (water depth 0.56 ft) it appears that there is a trend toward lower power requirements in the larger model (water depth 4.5 ft).

Horikawa [27] studied the refraction around a three-dimensional model of a hydraulic breakwater.

III. THEORETICAL CONSIDERATIONS

Brasher [1] suggested an initial theory that "the air bubbles rising from the perforated pipe have an explosive action, expanding as they do on their upward course, and this function disrupts the wave mass and effectually disturbs the rhythm and the continuity of its particles." It is to his credit that although his concepts of the mechanism of wave attenuation are questionable, he fully appreciated the fact that the best efficiencies are obtained in deep water. Thus he says, "the air breakwater must be placed far enough offshore to intercept the waves in their full swing and before they have reached shallow water. . . for otherwise, the air bubbles have but a trifling effect in combatting the waters."

A. Taylor's Theory

Taking the velocity potential of a deep water wave Taylor [7] superposed on it a uniform current of velocity U and thickness h . Writing the conditions of kinematic and pressure continuity at the free surface and at the current interface, he concluded that wave lengths below a certain critical value could not satisfy all the conditions simultaneously for given values of U and h . This means, that for a given current, it is kinematically impossible for waves shorter than a given length to be transmitted.

Subsequently [14] he modified this theory using a triangular velocity distribution, which is more in accord with the actual distribution. The

results of both methods are shown in Fig. 2. Using these curves one can determine the maximum wave length that can be attenuated by a current of velocity U and thickness h , or alternatively the current velocity required to attenuate a given wave.

In order to relate the velocity and thickness of the current to the air discharge and the submergence of the perforated pipe, Taylor used Schmidt's analogous solution for the convective currents above a horizontal line source of heat. Schmidt showed that the heated zone was a wedge of half angle $\tan^{-1} 0.28$. Assuming that this applies to the case of the pneumatic breakwater, the half width of the current would be $0.28d_1$ where d_1 is the submergence of the perforated pipe. At the free surface, the current turns through 90 degrees; therefore at this point the depth of the current would be $0.28d_1$.

By the same analogy the maximum velocity U of the current is given by

$$U = 1.9 (qg)^{1/3} \quad (1)$$

where q = air discharge per unit width, and the constant 1.9 is dimensionless.

It is seen therefore that the thickness of the current depends solely on the submergence. Thus for attenuating longer waves, where a deeper current is necessary, an increased submergence is desirable. However, it should be borne in mind that this entails larger power requirements, the power being directly proportional to the depth at which the bubbles are released. In addition, there is the factor of velocity diffusion, i.e., for a given discharge the greater the submergence, the smaller will be the current velocity at the surface.

B. A Simpler Counter-Current Theory

In 1942 Unna [6] provided an explanation for the increase in wave height in a tidal estuary. When the waves run into a counter-current, the relative velocity of propagation and hence the wavelength are reduced. Since the rate of energy transfer across a given plane must remain a constant, the wave height must increase.

Evans [10] extended this to breakwater studies, contending that the current created by a pneumatic or hydraulic breakwater increased the wave steepness to breaking point, thus causing the wave to dissipate its energy in the ensuing turbulence.

C. The Turbulent-Viscosity Theory

Since 1952, a theoretical and experimental study has been under way in Japan, under M. Kurihara [19]. The experimental work in the Laboratory was extended to include three full-scale tests of the pneumatic breakwater. It was found that the full-scale experiments required much less power than would be expected from a Froude extrapolation of the model data. The power requirements were also much below those predicted by Taylor's theory. Kurihara was therefore led to propound the "turbulent viscosity" theory.

He concluded from his data that while the counter current was only partially responsible, the attenuation of the wave was also due to the turbulent Reynolds stresses set up by the disturbance of the breakwater. In the model where the scale is small, the counter-current effect dominates, while on the large scale the "turbulent viscosity" is mainly responsible for attenuating the wave motion.

It should be mentioned here that theories dealing with the stopping action of the current or dissipation of the wave due to turbulence in the current are basically also applicable to the hydraulic breakwater, since only the mechanism of generating the current is different.

D. Work Done in the U. S. S. R.

In 1954 Teplov [11] published a summary of the work done in the U. S. S. R. on pneumatic breakwaters, and a translation of this paper has recently become available. His theory will be briefly mentioned here.

Due to a wave motion, the pressure at a given depth fluctuates about a mean. Teplov assumes that this fluctuating pressure does work on the rising bubbles, and the wave energy is thus expended. He assumes that the air bubbles oscillate horizontally with the same velocity as the adjoining water particles. Then the rate of energy dissipation dE/dt is given by

$$\frac{dE}{dt} = \int \frac{pvdF}{F}$$

where p is the local pressure,
 v is the local horizontal velocity, and
 dF is the projection of the air bubbles at a depth z for an element
of area width unity and height dz .

On this basis he obtains a formula for q , the required air discharge

$$q = \frac{0.03r}{1 - r}$$

where q is in $m^3/\text{min}/m$,

r is the ratio of energy extinguished to the initial energy, and the constant 0.03 is determined experimentally.

Alternatively in terms of the attenuation A

$$q = \frac{0.03 A(2 - A)}{(1 - A)^2}$$

The result is immediately seen to be fallacious, in that the air requirement depends solely on the desired attenuation and is entirely independent of the scale of the breakwater. A closer inspection reveals the faulty reasoning. If the bubbles oscillate with the same velocity as the neighboring water particles, then the work done on the bubbles is actually zero and not as stipulated by Teplov.

IV. EXPERIMENTAL STUDIES OF A PNEUMATIC BREAKWATER

A. Experimental Facilities

The experimental work was performed in two wave channels, which for all practical purposes were geometrically similar. The small channel is 2 ft wide, 1 ft 3 in. deep, and 50 ft long (Fig. 3). It is equipped with a pendulum-type wave generator capable of producing waves up to 10 ft long and with a maximum height of 5 inches. The water depth was maintained at 1 ft throughout the tests.

The large wave channel is 9 ft wide, 6 ft deep and 253 ft long (Fig. 4), and is equipped with a hinged-plate generator capable of producing waves

from 2 to 40 ft long; the maximum wave height obtainable is about 1.5 ft for an 18 ft wave. A water depth of 4.5 ft was used in these tests. Both channels are equipped with good wave absorbers.

The test procedure consisted essentially of measuring the incident and transmitted wave heights, with a capacitive wave profile recorder, for various discharges and wave characteristics. Difficulties were sometimes encountered in determining the transmitted wave height, particularly for intermediate attenuations and for wave lengths approximately equal to the channel width. Considerable scatter of the data resulted with intermediate attenuation. For wave lengths approximately equal to the channel width severe transverse waves sometimes developed. In the large channel an average of 20 waves was obtained soon after the breakwater was turned on; an effort was made to obtain the test record after the current system was established but before the transverse wave developed to any appreciable extent.

In the small channel a plate filter was installed between the generator and breakwater to reduce the intensity of transverse oscillations. This also stabilized the transverse wave and measurements were obtained at a node of the transverse system.

B. Experimental Results

1. Effect of Wave Length and Water Depth

Data were obtained in the large channel for L/d values from 1.2 to 2.4, while the range in the small channel was 1.2 to 4.2. The results are shown in Figs. 7 and 8, respectively.

Figure 9 is a summary graph of the above data where the power is expressed by a dimensionless parameter

$$\Phi = \frac{\text{hp/ft}}{\rho g^{3/2} L^{5/2}}$$

The horsepower per foot was computed at the orifices from the formula

$$\text{hp/ft} = \frac{qwd_1}{550}$$

where q = unit discharge at orifices,
 w = specific weight of water,
 d_1 = submergence of orifices,
 ρ = density of water,
 g = acceleration of gravity, and
 L = wave length.

This graph (Fig. 9) affords a comparison between the data obtained from the two models and also may be used for predicting prototype performance. For an attenuation of 75 per cent the agreement between the two models is quite good. With lower attenuations they indicate a decreasing value of Φ with increasing values of L/d , crossing the small-scale tests at about $L/d = 1.8$. As the small-scale tests cover a wider range of L/d and are generally more consistent than the large-scale tests, the curves have been drawn through them. It is probable that transverse waves in the large channel may have contributed to the problem as a wave filter was not used. Also, as may be noted in Fig. 8, the data for attenuations below 50 per cent are somewhat inconsistent.

2. Effect of Submergence

As noted in the section on theory, for efficient operation, the perforated pipe of the pneumatic breakwater should be placed at the bottom. However, in practice, other criteria may dictate the depth at which the breakwater is suspended. In such cases, it would be desirable to evaluate the efficiency of the system for a specified submergence.

Figure 10 shows the increased power required as a function of the relative submergence. The relative submergence is here defined as the ratio of the depth at which air is released to the depth of water. It is seen that the power requirements increase rapidly for decreasing submergence, particularly for the longer waves and higher attenuations. It is interesting to note that for the shorter waves $L/d = 1.22$ and 1.78 , the increase in power required is approximately the same. As the wave goes beyond the deep-water range, the decreased efficiency is more pronounced.

3. Effect of Wave Steepness

The effect of wave steepness was investigated in the small facility for waves of $L/d = 1.22, 1.78, 2.44$. The steepness was varied from 0.02 to

0.08. It was found that the air required for a given attenuation is independent of the wave steepness. Figure 11 shows a typical plot for $L/d = 1.78$.

The effect of steepness is one of the aspects in which the pneumatic breakwater differs from its hydraulic counterpart. As will be mentioned, the power requirements of a hydraulic breakwater increase with wave steepness by as much as a factor of about three for a wave steepness increasing from 0.02 to 0.08 ($L/d = 3.3$). This makes the pneumatic breakwater more efficient for the waves of higher steepness.

4. Effect of Orifice Area

The effect of orifice area was tested in the large facility. Orifices 1/8, 3/16, and 1/4 inches in diameter were tested for $L/d = 1.22, 1.78, 2.44$. Figure 12 is a typical plot for $L/d = 1.78$. It is seen that there is no pronounced change in the air requirements for the different orifices. However, the 3/16-in. orifice gave the most stable flow and was therefore used in all other tests.

5. Effect of the Number and Spacing of Manifolds

It was thought originally that multiple-manifold systems would be advantageous for longer waves, and where high attenuations were desirable. It was suggested that a multiple system might give a deeper surface current enabling it to attenuate longer waves where the orbital motion extended farther down. Also there was the possibility that, if the manifolds were sufficiently far apart, they would each attenuate the wave and thus give very high attenuations.

The above reasoning was not borne out by the experiments. Up to four manifolds at spacings of 12 in. center to center were tested in the large facility, and it was found that there was no advantage to using multiple-manifold systems. On the contrary, for lower discharges the air flow was not uniform and poor efficiencies resulted. Figure 13 shows a typical set of data for $L/d = 1.78$.

The manifold spacing was varied for a two-manifold system in the small wave channel to determine whether this parameter had any effect on the performance. Figure 14 shows the results for a wave with $L/d = 1.78$. Once more it is apparent that the performance is no better than for the one-manifold system. For higher values of L/d , up to 3.11, similar results were

obtained.

6. Comparison with Other Data

Figure 15 shows the 50 per cent attenuation curve taken from Fig. 9. This is compared with results obtained by other experimenters. While many papers have been published on the pneumatic breakwater, the data given are so limited and often incomplete that a comparison of this nature is very difficult.

The Japanese model experiments [18] and one prototype measurement are shown on this graph. Some small-scale data taken at this Laboratory and published in Project Report No. 46 are also shown for comparison. It is seen that the horsepower ratio for all the compared data is about 50 per cent less than that reported herein. This may be due in part to different methods of measuring the attenuated wave and to the effect of transverse waves.

7. Scale Effect

The dimensionless plot of Φ vs L/d shown in Fig. 9 indicates good correlation between data taken in depths of water of 1 ft and 4.5 ft for high attenuations. Thus within this range there is little, if any, scale effect.

If the Froude law relating the model to the prototype holds valid, the value of Φ should be unchanged for a given value of L/d . Very few prototype tests have been conducted, and even among these much of the data is unreliable. However, Kurihara [19] reported that his prototype tests required much less power than predicted from the model tests. The following table shows the Φ values for 50 per cent attenuation for both his model and prototype tests. Comparative values from tests performed at this Laboratory are also tabulated.

It should be noted here that factors like reflection off the quay wall and diffraction around the ends of the breakwater tend to give erroneous values for the transmitted wave. It has been shown [9] that these multiple reflections can either increase or decrease the wave height depending on the geometry of the setup.

Another factor that may affect comparisons of data from various sources is the location of the measuring element with respect to the breakwater. If the measuring element is less than twice the depth from the breakwater the attenuations would be higher, or for a given attenuation the power

SCALE EFFECT OBTAINED IN KURIHARA'S TESTS

(Attenuation = 50 per cent)

Depth of Water, m	L/d	$\Phi = \frac{hp/ft}{\rho g^{3/2} L^{5/2}}$	Equivalent SAF Values of Φ
0.7	2.32	0.206	0.76
0.7	1.73	0.175	0.64
0.8	1.75	0.282	0.64
16.1	1.45	0.067	0.64
15.6	1.51	0.053	0.64
9.5	1.53	0.097	0.64
10.0	2.20	0.215	0.70

requirement lower, than would be the case if the measuring unit were close to the breakwater.

In the St. Anthony Falls Hydraulic Laboratory tests the lee measuring element ranged from 5.5 to 14 times the depth away from the breakwater. Also, the lee probe was traversed over at least one-half wave length to eliminate reflections from the absorber.

8. Velocity Profiles

Velocity profiles of the surface current, produced by the rising bubbles, were taken in both channels. These generally confirmed the qualitative flow pattern shown in Fig. 1. (Typical vertical profiles are shown in Fig. 16.) Figure 17 shows the variation in velocity at three points as a function of air discharge.

Equation (1) in the theoretical section also relates the current velocity to the air flow rate. In the fps system this reduces to

$$q = 0.00454 U^3 \quad (2)$$

where q = unit discharge of air, and

U = maximum current velocity.

If the point of measurement is other than where the velocity is a maximum, the constant 0.00454 must naturally be reduced. However, the exponent of U must remain a constant. It is seen that the experimental data (Fig. 17) fall

parallel to the theoretical line. The values of the velocity at the water surface were obtained by extrapolating the velocity profiles. The latter points fall quite close to the Taylor values. It may thus be said that the maximum velocity measured one water depth from the breakwater is in good conformity with Taylor's theory.

9. Intermittent Bubbles

A device for releasing bubbles intermittently and thereby saving power was tested in the prototype scale at Dover, England [27]. Briefly, the purpose was to generate a zone of turbulence through which the waves would pass and be attenuated.

Such a device was tested very briefly in the small channel and it was found that there was no particular advantage in using an intermittent stream of bubbles. Details of the construction of the Dover installation were not available, so the tests were not necessarily an evaluation of the Dover installation.

C. Summary--Pneumatic Breakwater Tests

The primary observations and conclusions resulting from the tests of pneumatic breakwaters are as follows:

- (1) The dimensionless horsepower ratio

$$\Phi = \frac{\text{hp/ft}}{\rho g^{3/2} L^{5/2}}$$

is the parameter giving the requirements of the pneumatic breakwater. The value of Φ depends only on the wave length, depth of water, and the submergence of the manifold. Figures 9 and 10 illustrate the variation of Φ with these parameters and may be used for design purposes.

- (2) For $L/d < 2$, Φ is a constant for a given attenuation. When L/d exceeds 2, Φ increases rapidly to the point where the breakwater is no longer very effective.
- (3) The breakwater is most efficient when the manifold is placed at the bottom. For decreasing submergence, the

value of Φ increases.

- (4) The power requirements are independent of wave steepness.
- (5) The orifice diameter is not an important parameter. However, if too large, the air flow is unstable, and if too small, the head loss is too large.
- (6) There is no particular advantage to using multiple-manifold systems. On the contrary, the flow tends to be unstable for low discharges.
- (7) The power requirements for pneumatic and hydraulic breakwaters are much the same for equal attenuations, except that the pneumatic system cannot achieve more than 80 per cent attenuation.
- (8) No appreciable scale effect was observed between models having a length ratio of 4.5:1. However, a few full-scale tests conducted in Japan indicate greater efficiency in the prototype. This needs further confirmation before any conclusions are drawn.

V. EXPERIMENTAL STUDIES OF HYDRAULIC BREAKWATERS

A. General Comments

Experimental studies of hydraulic breakwaters were performed prior to the studies of the pneumatic system and have been described in earlier reports [20, 25]. The results are summarized in this paper to assist the reader in comparisons of the two systems.

The major part of the program involved two-dimensional tests with a single manifold, in the same channels used in the pneumatic tests. Test procedures were also similar.

The primary objective was the procurement of information concerning the effect of various parameters on wave attenuation, which is defined as $1 - H_T/H_I$ (H_I = incident wave height and H_T = transmitted wave height). Parameters which were varied include the wave length L , the wave height H , jet submergence y , jet area a_j , and jet discharge q . The horsepower and discharge required under various test conditions were expressed in the form

of dimensionless ratios as follows:

$$\Phi = \frac{\text{Horsepower per lineal foot of breakwater}}{\rho g^{3/2} L^{5/2}}$$

$$\Psi = \frac{\text{Discharge per lineal foot of breakwater}}{L \sqrt{gd}}$$

where Φ is the dimensionless power ratio, Ψ is the dimensionless discharge ratio, ρ is the density of water in slugs per cubic foot, g is the acceleration due to gravity in feet per second per second, and d is the water depth.

The horsepower was normally computed at the jets; for design purposes it would be necessary to add losses in the supply lines and manifold.

Equipment and procedures used in these tests were essentially the same as those for the pneumatic breakwater tests.

B. Experimental Results

The data indicated that the power requirements of a hydraulic breakwater are primarily dependent upon wave length, water depth, wave steepness, submergence of the nozzles, spacing and size (or area) of nozzles, and the number of manifolds.

1. Effect of Wave Length and Water Depth

Experimental data were obtained in the large channel for L/d values up to about 4.2 and in the small channel up to 5.6. Considerable scatter occurred for values less than 1.0. Excluding these values, it appears that Φ , the horsepower ratio, is fairly constant for L/d values up to about 2.0 and that it increases quite rapidly for L/d values in excess of 2.0. Figure 18 illustrates typical data for an attenuation of 100 per cent. In this instance the power requirements for a single manifold increase by a factor of 7 as the L/d value is increased from 2.0 to 5.0.

2. Effect of Wave Steepness

Wave steepness has an important effect on power requirements. Figure 19 presents typical data for three L/d values based on the small-scale tests. Considering the curve for a L/d value of 3.33, it may be noted that

for an increase in wave steepness from 0.02 to 0.08, a factor of 4, the required horsepower ratio Φ is increased by a factor of about 3. However, the true efficiency, that is, the ratio of attenuated wave energy to the jet energy, is considerably higher for the steep waves than it is for the shallow waves.

3. Effect of Jet Submergence

The results presented in Fig. 20 indicate that the zero submergence condition is the most efficient; however, there may be objections to a prototype installation using this value, and the majority of tests were based on a relative submergence value y/d of 0.091. A slight correction can be applied to such data by use of Fig. 20 if zero submergence is required.

4. Effect of Jet Area

One of the most important parameters affecting both the discharge and horsepower requirements is the jet area per lineal foot of breakwater. Jet area is dependent on two variables--jet spacing and jet size. During this study both the jet spacing and jet area were varied; large-scale tests covered spacings of 1.34, 2.56, and 5.11 jets per ft and a number of jet diameters between 0.422 and 1.047 inches. Four L/d ratios (L = wave length, d = water depth) were selected for the tests: 0.72, 1.22, 1.78, and 2.44. A typical summary graph for $L/d = 1.78$ and 100 per cent attenuation is presented in Fig. 21. In this case the dimensionless horsepower ratio Φ and dimensionless discharge ratio Ψ were plotted against the dimensionless jet area which is defined as

$$\frac{\text{area of jets}}{\text{cross-sectional area of channel}}$$

which is equal to

$$\frac{a_j}{a_c} = \frac{\text{area of jets per foot}}{l \times \text{depth of channel}}$$

Figure 21 indicates that the discharge and power requirements are strongly dependent on the jet area. Low values of jet area are associated with a requirement for high power, high jet velocities, and low discharges.

It appears that power requirements at the jets or nozzles will probably decrease as the jet area is increased until the latter is equal to the cross-sectional area of the required surface current. Apparently the turbulent losses in the jets decrease as the jet velocity decreases, resulting in better efficiencies. However, as the jet area is increased, the required discharge is increased; as a result, it would be necessary to use larger manifolds and supply pipes or provide for larger losses in these components.

5. Efficiency

As noted in the section on wave steepness, more power is required to attenuate relatively steep waves than flat waves; however, the efficiency of the system η is much higher for the steep waves than it is for the flat waves. The efficiency is defined as

$$\eta = \frac{P_I - P_T}{P_j}$$

where P_I is the power in incident wave, P_T is the power in transmitted wave, and P_j is the power in the hydraulic jets.

Figure 22 illustrates experimental data on efficiency. For the one-manifold system, the efficiency varies with steepness, attenuation, and relative wave length. For attenuations of 80 to 100 per cent, the maximum efficiency obtained was only about 12 per cent.

6. Scale Effect

The small and large wave channels, being geometrically similar, were ideally suited for a scale-effect study. For the purpose of comparison of small- and large-scale models, it was assumed that gravity and inertia forces are of primary importance in relating the two models and, consequently, that Froude's law governs. The Froude number may be expressed as

$$F_n = \frac{C}{\sqrt{gd}}$$

where C = wave celerity,

g = acceleration of gravity, and

d = depth of water.

Consequently, the length ratio for the two models is $L_r = 1/4.5$, the discharge ratio is $Q_r = L_r^{5/2}$, and the power ratio is $P_r = L_r^{7/2}$.

Figure 23 illustrates comparative discharge data expressed in terms of the small model. The preceding formulas were used to reduce the large-scale data to the appropriate conditions. It may be noted that data in the low-attenuation region exhibited considerable scatter and that the curves had a pronounced discontinuity; the latter may have resulted from a variation in the mechanism of energy loss as the jet discharge was varied. Considering the data for attenuations in excess of 50 per cent and L/d values in excess of 1.0, agreement between the data for two models is quite good. It was concluded that little, if any, evidence of scale effect existed for the 1:4.5 scale ratio of the two sets of tests.

C. Summary Curves

Power requirements at the jets are summarized in Figs. 24 through 26 in dimensionless form. The faired curves were based primarily on the large-scale tests in the region of $1 < L/d < 2.5$ and upon small-scale tests for $L/d > 2.5$. The dimensionless power ratio Φ can be determined for selected values of L/d , H/L , a_j/a_c , and attenuation.

The required discharge per lineal foot of breakwater can be computed by the following formula:

$$q = 58.58 a_j^{2/3} \Phi^{1/3} L^{5/6} \text{ in cfs per ft}$$

where 58.58 is a constant = $1100^{1/3} g^{1/2}$ and L is the wave length.

D. Conclusions--Hydraulic Breakwater

- (1) Horsepower and water-discharge requirements for the hydraulic breakwater are dependent upon wave characteristics such as length, height, and water depth, and breakwater characteristics such as spacing and size of nozzles, submergence of nozzles, and the number of manifolds.
- (2) A single-manifold hydraulic breakwater is quite effective for deep-water waves ($L/d < 2$), but its effectiveness decreases with increasing L/d ratios.

- (3) For high values of attenuation, more power is required to attenuate the steep waves; however, the efficiency of the system based on the ratio of the difference between incident and transmitted wave energy to the jet energy is much higher for the steep waves than it is for the flat waves.
- (4) Zero submergence of the nozzles appears to be most efficient for the range of wave lengths tested; however, the differences in discharge requirements are not large for values of y/d between 0 and 0.1.
- (5) Power and discharge requirements at the nozzles are dependent upon the area of jets per unit length of manifold. As the area is increased, the power decreases, and the discharge increases. As losses in the pumping and supply system are dependent on the discharge, the supply system must be analyzed along with the manifold and jet system in order to determine the optimum jet area.
- (6) Comparative data on the hydraulic breakwater obtained in the large and small channels (scale ratio 1:4.5) agree quite well when compared on the basis of Froude's law for values of L/d between 1.22 and 1.78. This would indicate that little scale effect exists over this range and tends to substantiate extrapolation of the data to the prototype condition.
- (7) There are preliminary indications that a several-manifold hydraulic breakwater producing a thicker surface current might require less power at the jets and higher discharges than a breakwater with a single manifold for large L/d values, but additional tests would be required to determine the optimum efficiencies of several-manifold breakwaters for a given range of L/d values.
- (8) Limited comparisons with Taylor's theory indicated the theoretical value of surface-current velocity usually occurred at a distance of one or two wave lengths from the

breakwater for an experimental wave steepness of 0.01. The comparison is somewhat arbitrary as the theory does not consider wave steepness.

- (9) Snyder's experiments show fairly good agreement with the tests described herein for high values of attenuation. For low values of attenuation Snyder's experiments resulted in somewhat larger discharges than St. Anthony Falls Hydraulic Laboratory data indicate.
- (10) Brief tests with the manifold on the bottom and the water jets issuing vertically required considerably more power than a horizontal jet system located near the water surface.

VI. APPLICATION OF BREAKWATER DATA TO A TYPICAL PROBLEM

In the preceding sections the experimental data were presented in a form which would illustrate the effect of various parameters on some aspect of performance. Summary curves were included (Fig. 9 for the pneumatic system and Figs. 24, 25, and 26 for the hydraulic system) showing a dimensionless power index Φ as a function of various parameters. These curves will be of assistance in extrapolating the results to prototype installations. The model tests did not reveal any significant scale effect between tests in a water depth of 1 ft and those in a depth of 4.5 ft. Scale-effect tests by Williams [26] were considered inconclusive. On the other hand, tests by Kurihara [19] indicated lower power requirements in the prototype than a Froude extrapolation of the pneumatic model data would indicate. As an overall observation it appears that further information on full-scale installations is desirable. For the present, a Froude extrapolation of the model data is recommended, in accordance with Figs. 24, 25, 26, and 9, as the most conservative approach.

With regard to a comparison of the pneumatic and hydraulic systems, the following observations can be made:

- (1) Higher attenuations, approaching 100 per cent, can be obtained with the hydraulic system as compared to maximums in the range from 60 to 80 per cent for the pneumatic system.

- (2) In general, the applied power at the nozzles will be less for the pneumatic system than for the hydraulic, for attenuation on the order of 60 per cent and L/d values less than 2.
- (3) Losses in the supply lines will usually be less for the pneumatic system, or conversely, for the same total power and pipe size a single supply line will accommodate a longer manifold length, resulting in a simplified supply system.
- (4) Installation of the pneumatic manifold on the harbor floor (for shore installations) would be easier and less subject to wave damage than a hydraulic manifold at or near the water surface. On the other hand, silt and sand may present some problems with a bottom installation.
- (5) With both systems power requirements increase quite rapidly with wave length for L/d values in excess of 2.

To illustrate some of these factors the following computations for a specified site are included. The site is one which was of interest to the Navy and for which wave data were available. The wave conditions to be expected in this region were given as follows:

Wave Height, ft	Wave Length, ft
6	70
7	100
8	125
9	150
10	180

The depth of water in the harbor was 40 ft.

Using Fig. 9, the horsepower requirements for a pneumatic breakwater were computed and are shown in Fig. 27. It should be remembered that the wave height is not of interest in this figure since the air required is independent of the steepness. Figure 28 shows the requirements of a hydraulic breakwater for similar conditions. Here the attenuation depends upon both the wave height and the diameter of jets used. The horsepower values are computed

from the dimensionless plots given in Figs. 24 through 26. An intermediate jet area ratio of 0.0035 was used, since for lower areas the applied horsepower was too high, and for higher areas the friction losses were too high.

Figures 29 and 30 illustrate the supply-line losses for the pneumatic and hydraulic systems, respectively. As an illustration a value of 100 hp was chosen for the applied power in each system, and the total power required, including losses, was computed for various supply-line diameters. This, however, is naturally dependent on the length of the breakwater and the length of the supply lines. In these computations the length of supply line was assumed to be 250 ft and the total coefficient of loss for valves and bends was assumed to be 1.0.

L I S T O F R E F E R E N C E S

- [1] Anonymous. "The Brasher Air Breakwater." The Engineer, p. 414. 1916.
- [2] Anonymous. "Severe Test of the Brasher Air Breakwater." Compressed Air Magazine. 1916.
- [3] Bogolepoff, J. A. "Der Pressluft-Wellenbrecher." Bulletin of the Permanent International Association of Navigation Congresses. 1937.
- [4] Platzer, Sven. "Pneumatiska vågbrytare." Teknisk Tidskrift, Vol. 55, pp. 105-109. 1938.
- [5] Schijf, J. B. "Het vernietigen van golven door het inspuiten van lucht (Pneumatische golfbrekers)." De Ingenieur, Vol. 55, pp. 121-125. 1940.
- [6] Unna, P. J. H. Nature, Vol. 149, p. 219. February 1942.
- [7] Taylor, G. I. "Note on Possibility of Stopping Sea Waves by Means of a Curtain of Bubbles." Admiralty Scientific Research Department, ATR/Misc/1259. 1943.
- [8] Carr, John H. Mobile Breakwater Studies. California Institute of Technology, Hydrodynamics Laboratory, Report No. N-64.2. 1950.
- [9] Stoker, J. J., Fleishman, B., and Weliczker, L. Floating Breakwaters in Shallow Water. New York University, Institute for Mathematics and Mechanics, February 1953. 46 pages.
- [10] Evans, J. T. Pneumatic and Similar Breakwaters. British Transport Commission, Docks and Inland Waterways, Report No. 21. 1954. 23 pages.
- [11] Teplov, A. V. The Scientific Principles for the Use of Pneumatic Breakwaters. Moscow, 1954. Translated and issued December 1958 by Technical Information and Library Services, Ministry of Supply, London.
- [12] Anonymous. "Strange Case of the Pneumatic Breakwater." Compressed Air Magazine. Vol. 59, No. 8, pp. 221-222. August 1954.
- [13] Wetzel, J. M. Experimental Studies of Pneumatic and Hydraulic Breakwaters. University of Minnesota, St. Anthony Falls Hydraulic Laboratory Project Report No. 46. May 1955. 61 pages.
- [14] Taylor, G. I. "The Action of a Surface Current Used as a Breakwater." Proceedings of the Royal Society, A, Vol. 231, pp. 466-478. 1955.
- [15] Evans, J. T. "Pneumatic and Similar Breakwaters." Proceedings of the Royal Society, A, Vol. 231, pp. 457-466. 1955.

- [16] Hensen, Walter. "Modelversuche mit pneumatischen Wellenbrechern." Mitteilungen der Hannoverschen Versuchsanstalt für Grundbau und Wasserbau, Franzius Institut der Technischen Hochschule Hannover, Heft 7, pp. 179-214. 1955.
- [17] Anonymous. "Pneumatic Breakwater Trials." The Dock and Harbour Authority, No. 426, Vol. 36, p. 364. 1956.
- [18] Laurie, A. H. "The German Experiments on Pneumatic Breakwaters." The Dock and Harbour Authority, No. 416, Vol. 36, pp. 61-64. 1955.
- [19] Kurihara, Michinori. "On the Study of a Pneumatic Breakwater." Appears in five parts in Reports of Research Institute for Applied Mechanics, Kyushu University, 1955-58. Parts I, II, and III have been translated by K. Horikawa in reports of the Wave Research Laboratory, University of California, Berkeley, Series 104, Issues 4, 5, and 6. English abstracts of Parts IV and V are available.
- [20] Herbich, John B., Ziegler, Jurgen, and Bowers, C. E. Experimental Studies of Hydraulic Breakwaters. University of Minnesota, St. Anthony Falls Hydraulic Laboratory Project Report No. 51. June 1956. 103 pages.
- [21] Hensen, W. "Erprobungen von pneumatischen Wellenbrechern in Model und in der Natur." Mitteilungen der Hannoverschen Versuchsanstalt für Grundbau und Wasserbau, Franzius Institut der Technischen Hochschule Hannover, Heft 10, pp. 183-209. 1957.
- [22] Snyder, C. M. Model Hydraulic Breakwater Studies. Wave Research Laboratory, University of California. 1957. 30 pages.
- [23] Dilley, R. A. Shipboard Hydraulic Breakwater. Wave Research Laboratory, University of California. 1957. 53 pages.
- [24] Snyder, C. M. Model Study of a Hydraulic Breakwater Over a Submerged Barrier. Wave Research Laboratory, University of California. 1957. 56 pages.
- [25] Straub, Lorenz G., Herbich, John B., and Bowers, C. E. "An Experimental Study of Hydraulic Breakwaters." Proceedings of the Sixth Conference on Coastal Engineering, Gainesville, Florida, pp. 715-728. December 1957.
- [26] Williams, John A. Scale Effects of Models of Hydraulic Breakwaters. Wave Research Laboratory, University of California. 1958. 40 pages.
- [27] Horikawa, K. Three-Dimensional Model Studies of Hydraulic Breakwaters. Wave Research Laboratory, University of California. 1958. 43 pages.
- [28] Heath, William A. "Pneumatic Breakwater Development in England." The Military Engineer, Vol. 51, No. 340, March-April 1959.

F I G U R E S
(1 through 30)

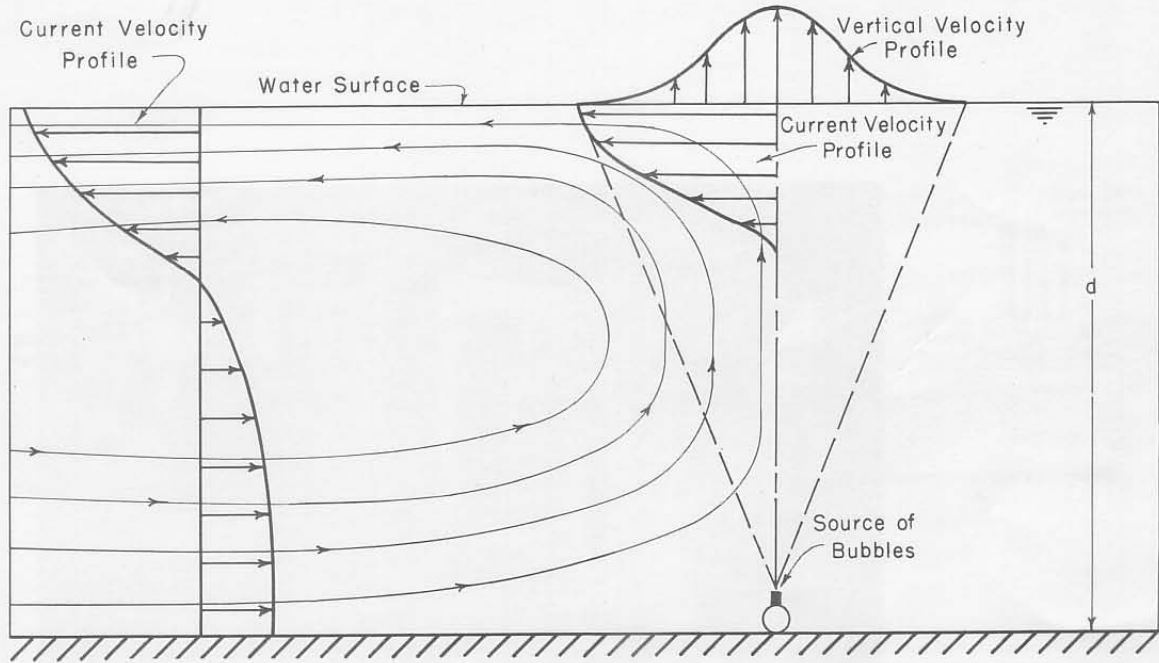


Fig. 1 - Qualitative Pattern of Surface Currents Produced by Air Bubbles. Pneumatic Breakwater.

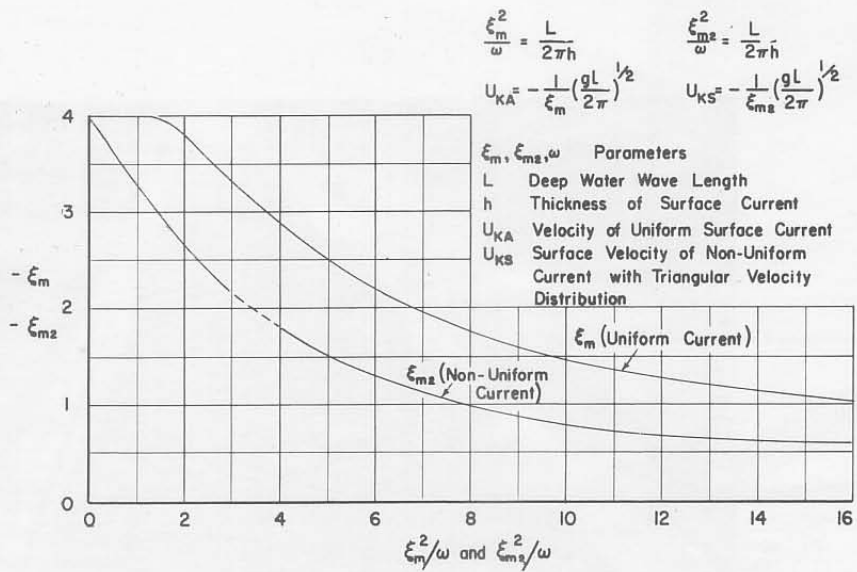


Fig. 2 - Solution for Taylor's Theory for Both Uniform and Nonuniform Velocity Distribution in Surface Current



Fig. 3 - View of Small Wave Channel

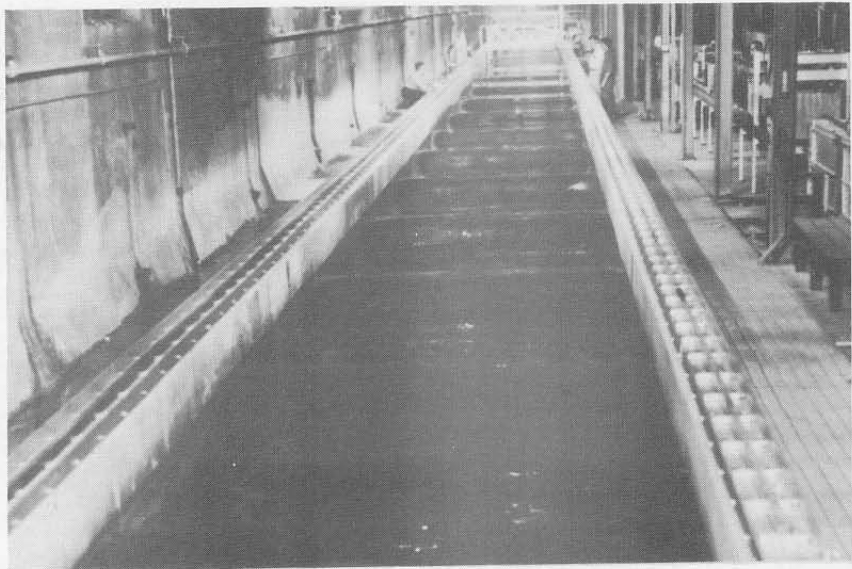


Fig. 4 - View of Large Wave Channel

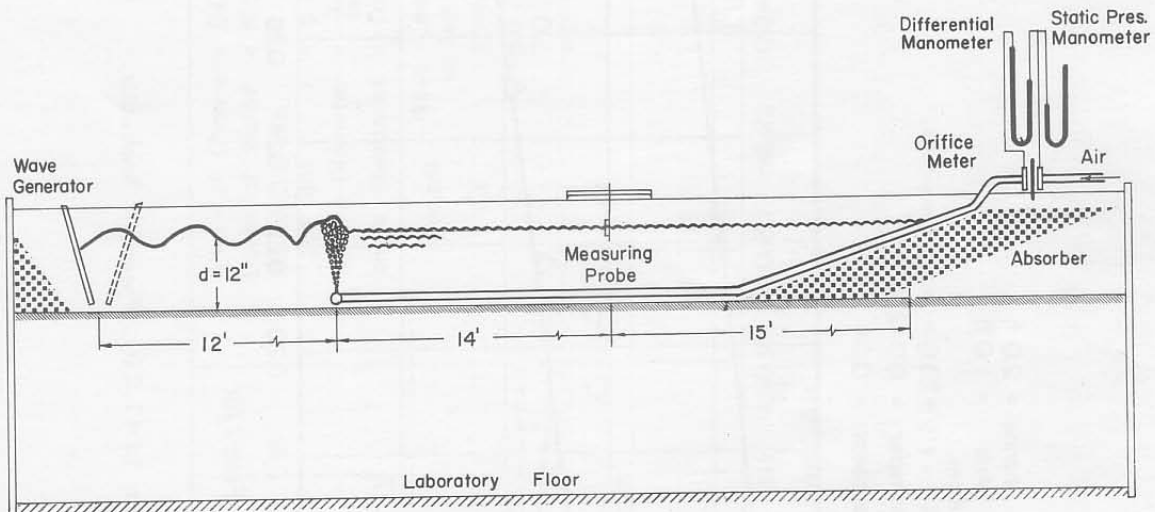


Fig. 5 - Sketch of Test Setup in the Small Channel

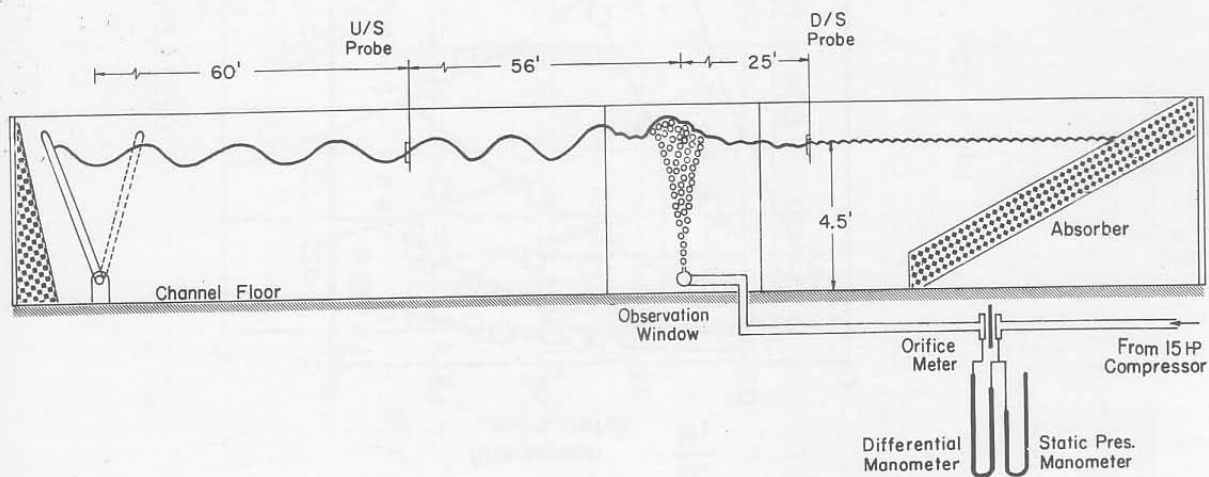


Fig. 6 - Sketch of Test Setup in the Large Channel

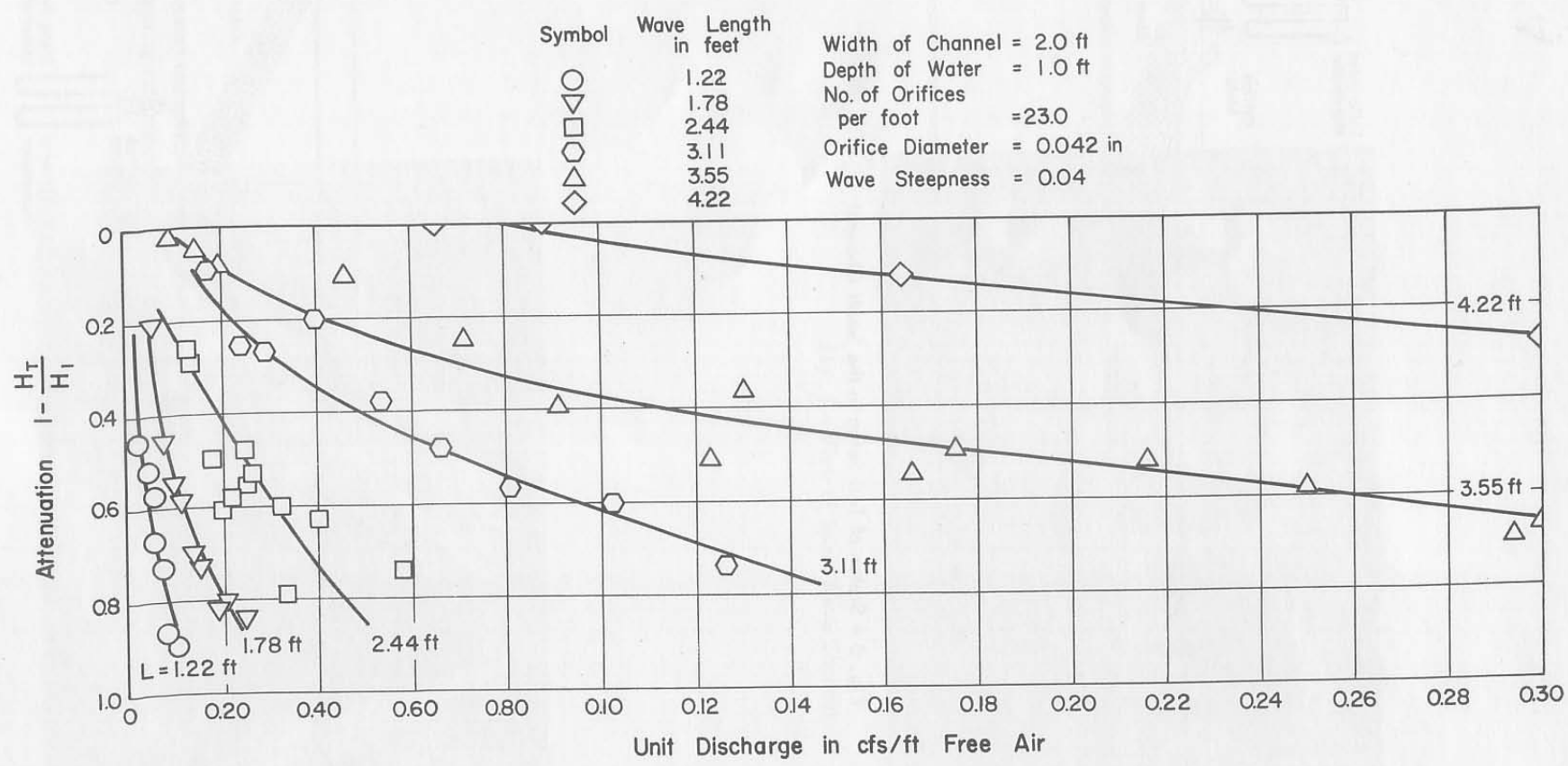


Fig. 7 - Effect of Wave Length on Air Discharge Requirements (d = 1.0 ft). Pneumatic Breakwater.

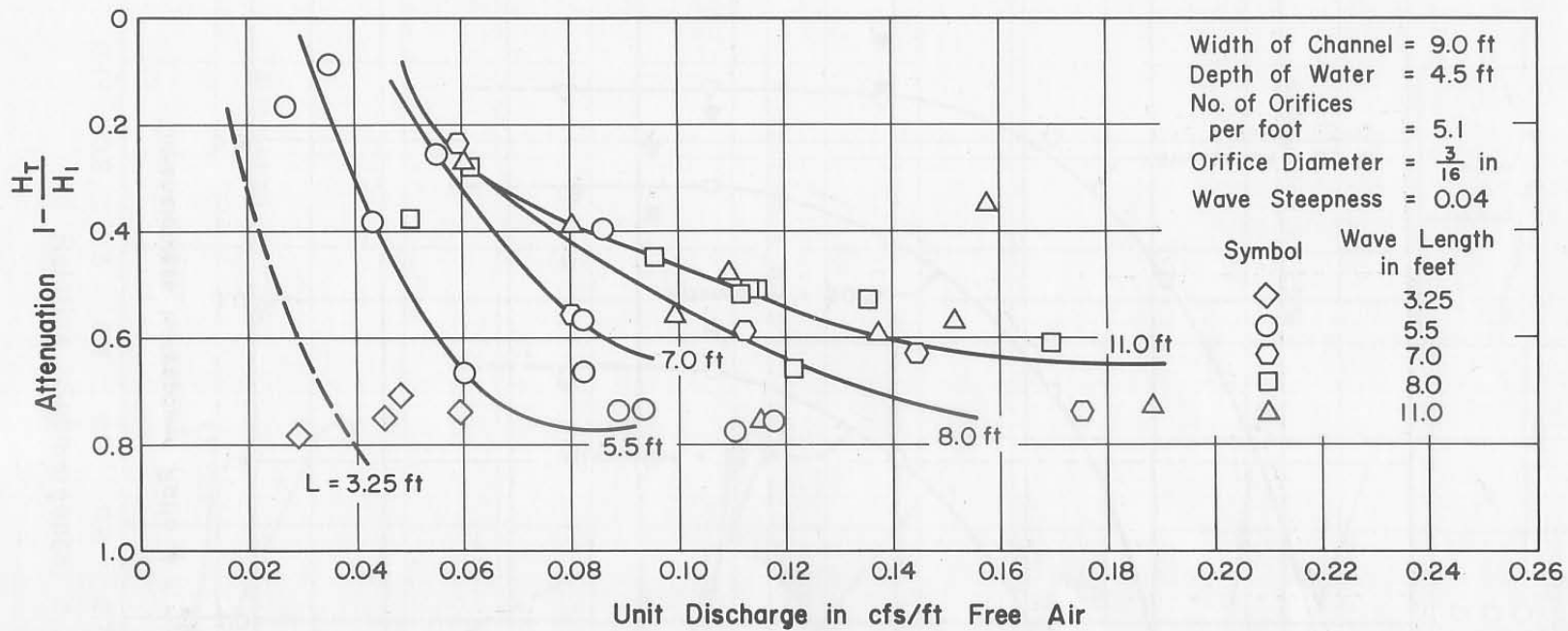


Fig. 8 - Effect of Wave Length on Air Discharge Requirements ($d = 4.5$ ft). Pneumatic Breakwater.

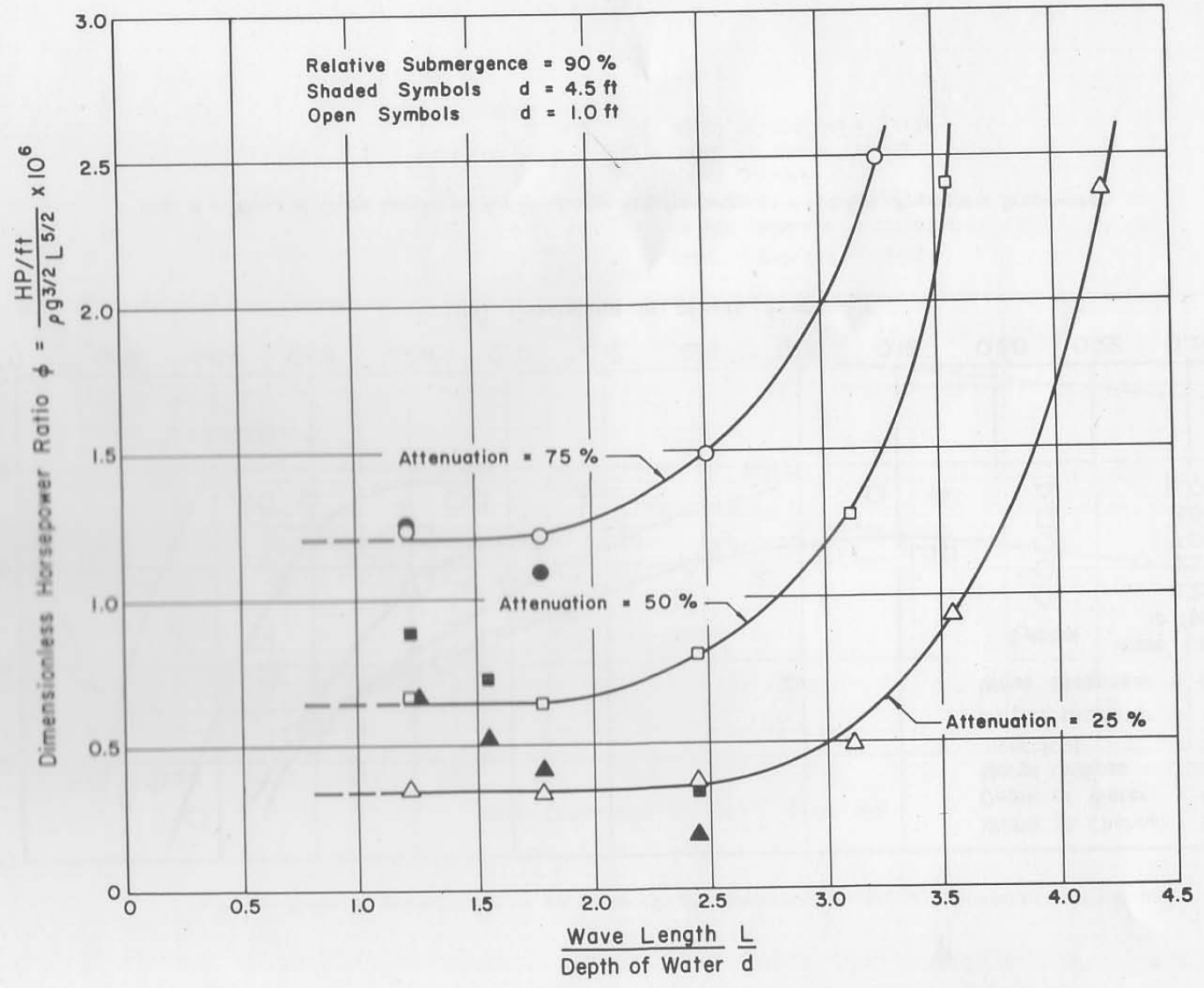


Fig. 9 - Horsepower Ratio as a Function of Length-to-Depth Ratio. Pneumatic Breakwater.

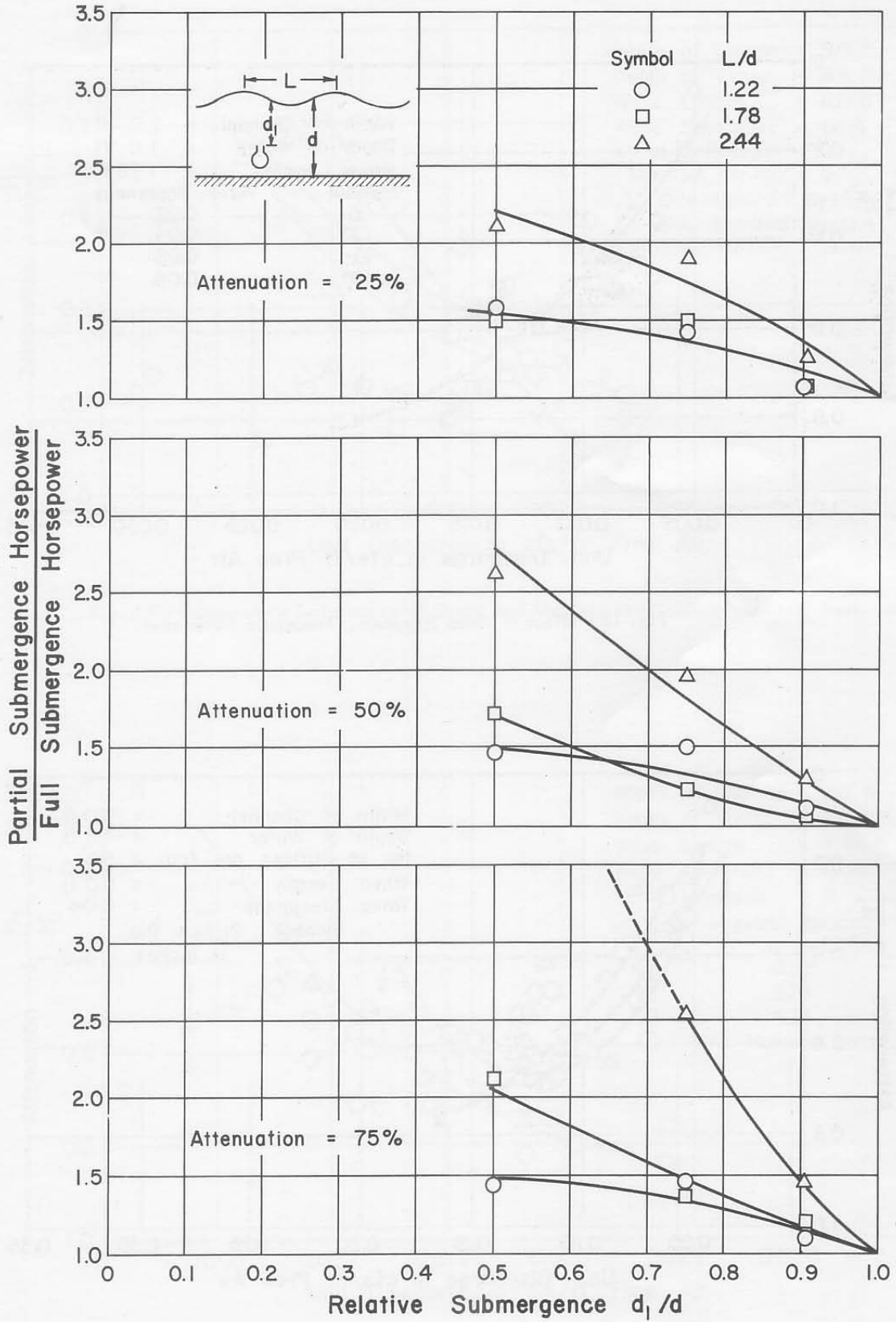


Fig. 10 - Relative Horsepower as a Function of Submergence. Pneumatic Breakwater.

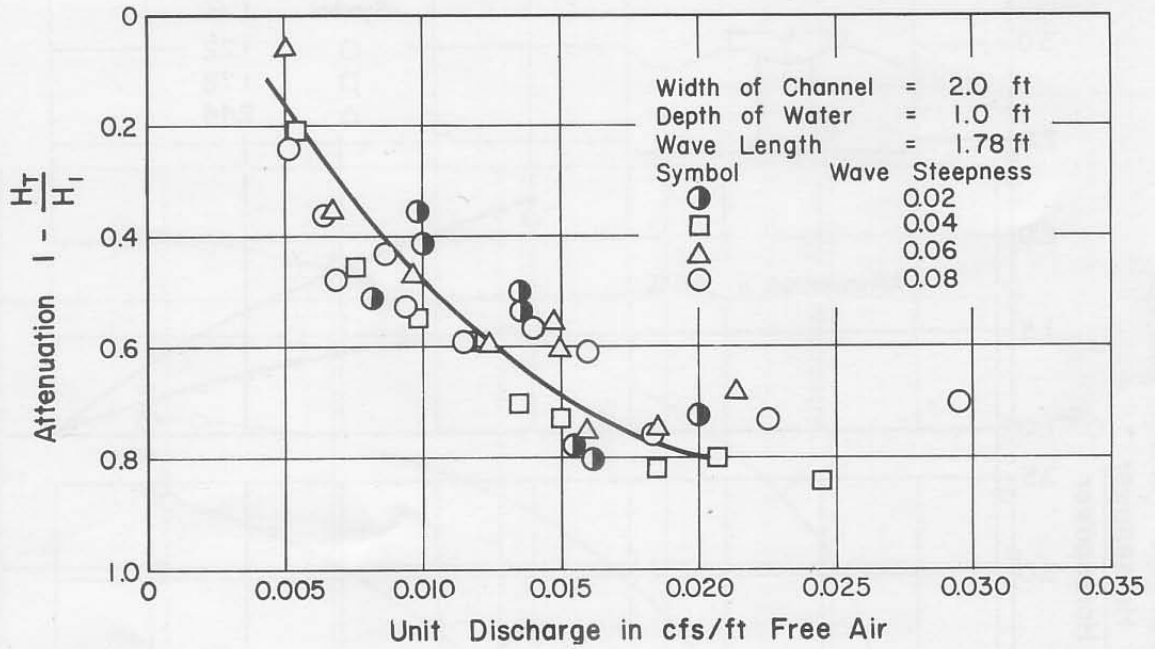


Fig. 11 - Effect of Wave Steepness. Pneumatic Breakwater.

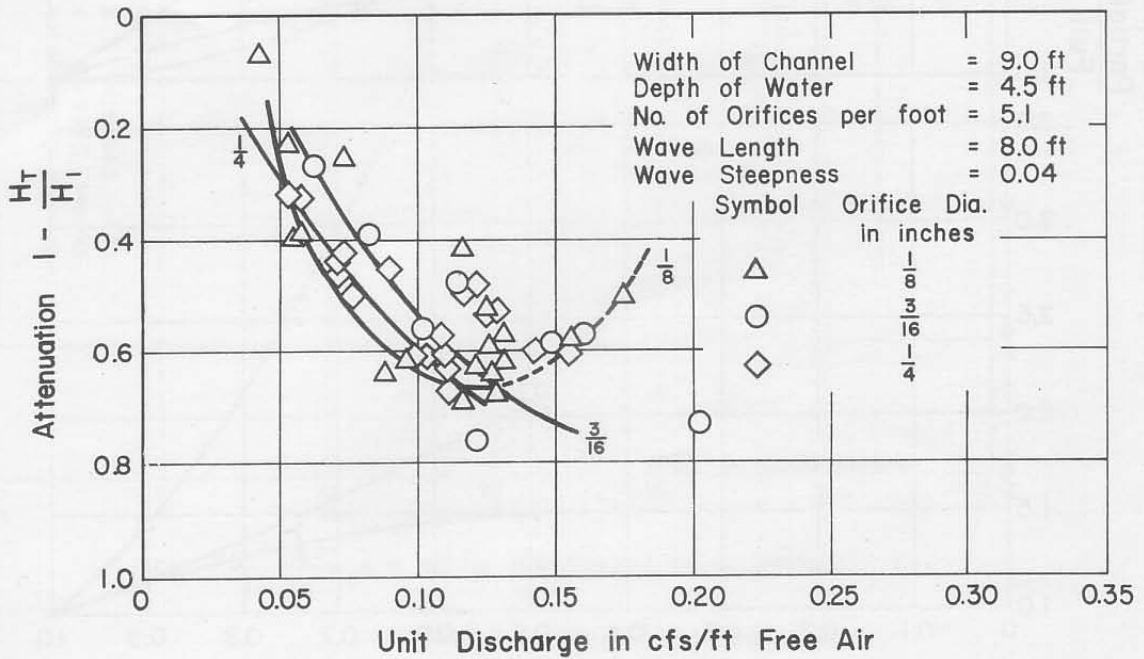


Fig. 12 - Effect of Orifice Diameter. Pneumatic Breakwater.

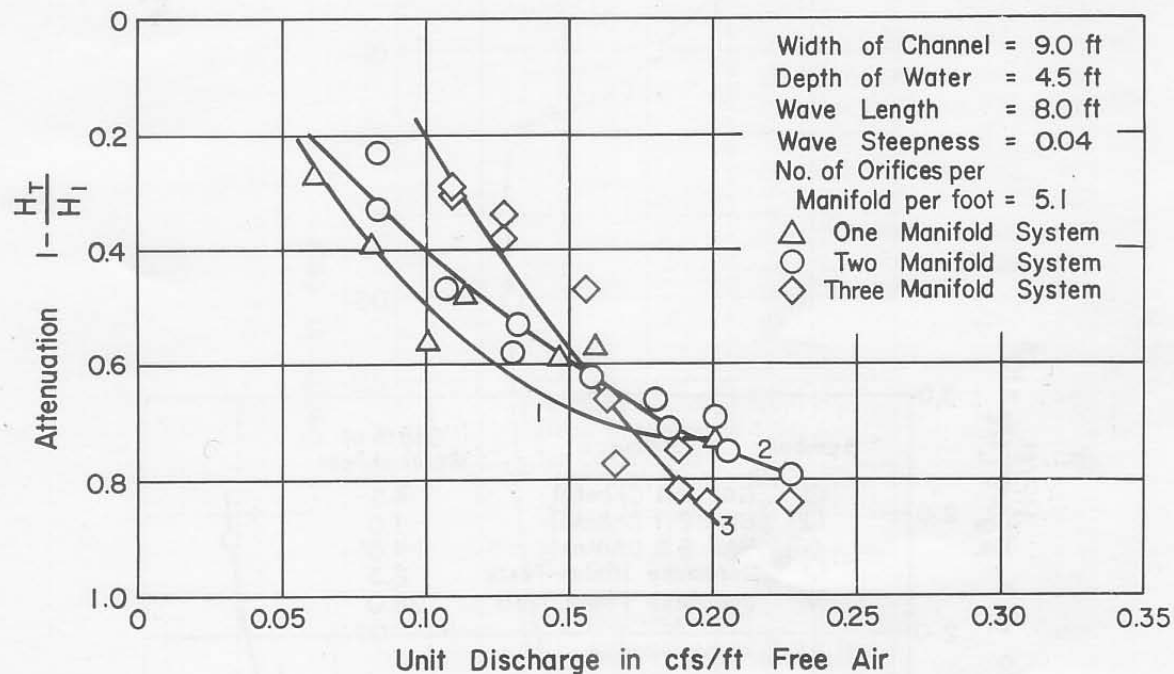


Fig. 13 - Comparative Performance of Single and Multiple Manifolds. Pneumatic Breakwater.

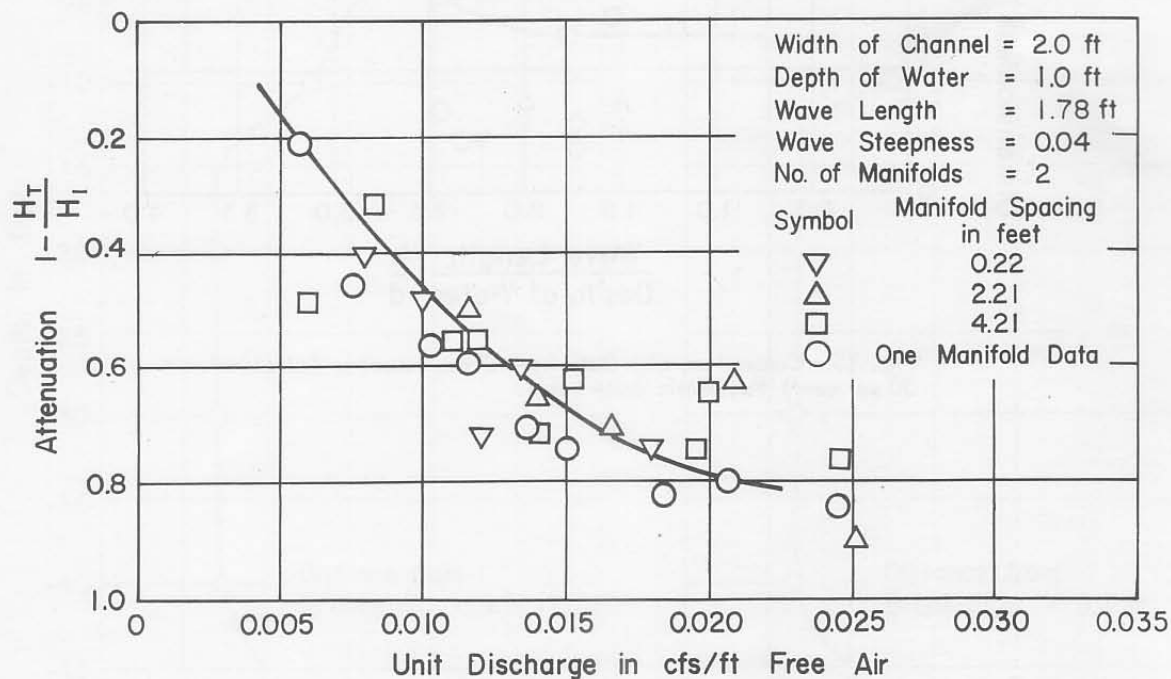


Fig. 14 - Effect of Manifold Spacing. Pneumatic Breakwater.

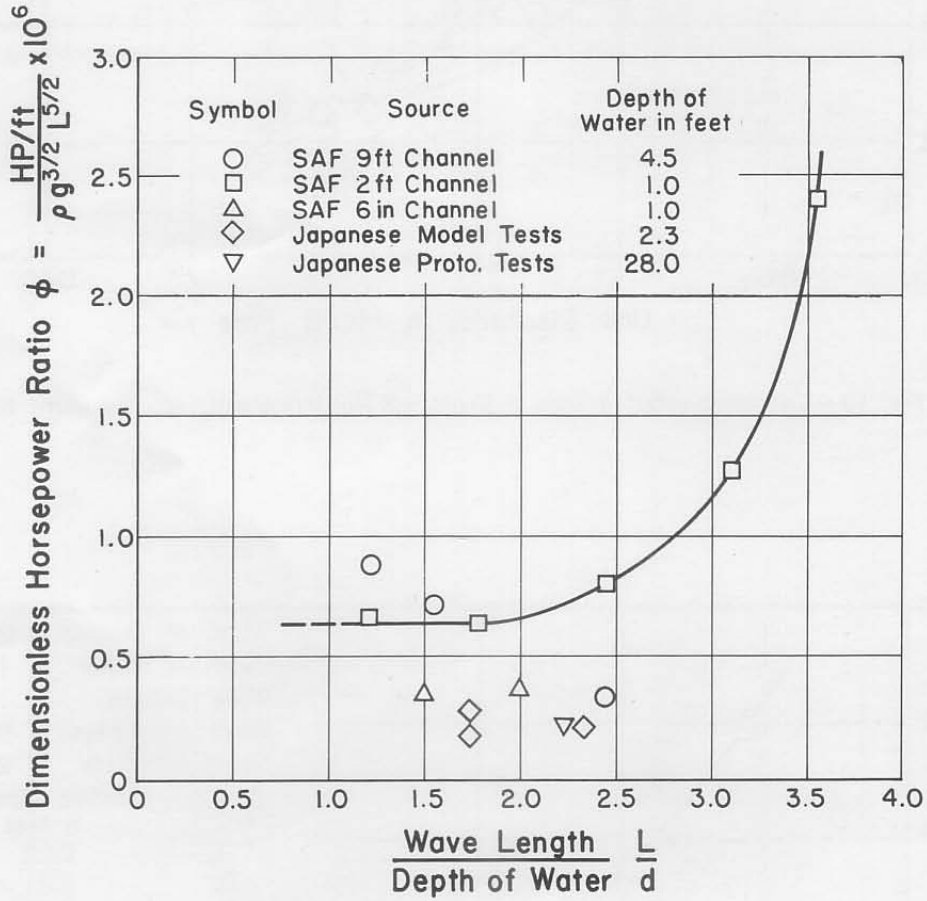


Fig. 15 - Comparison with Data from Other Sources. (Attenuation = 50 per cent) Pneumatic Breakwater.

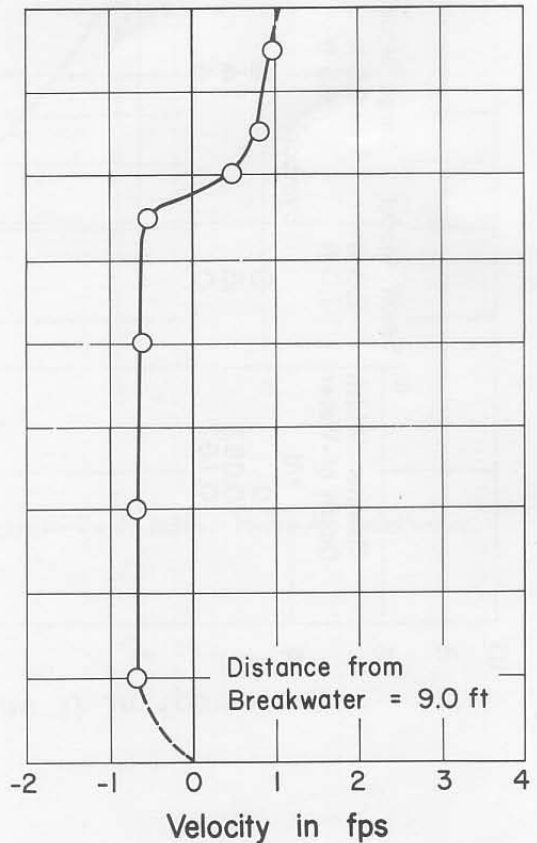
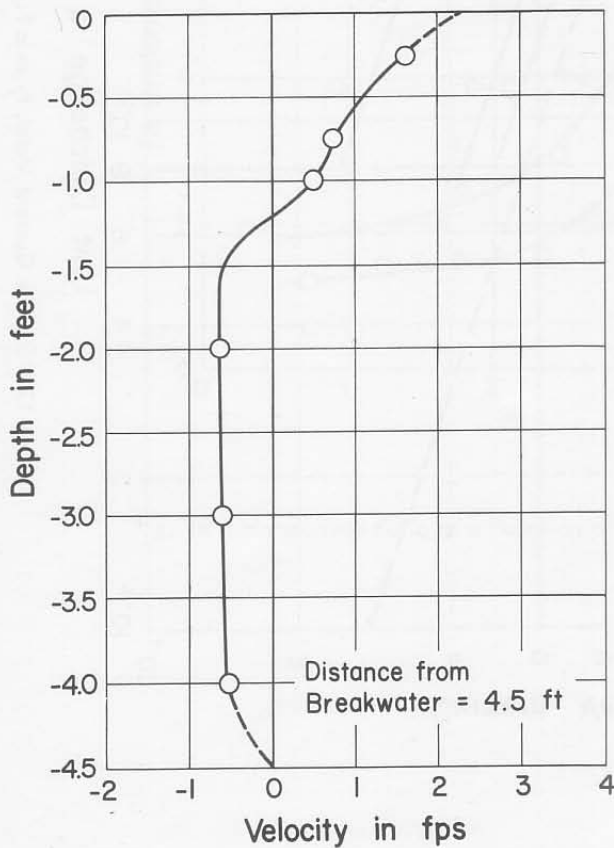
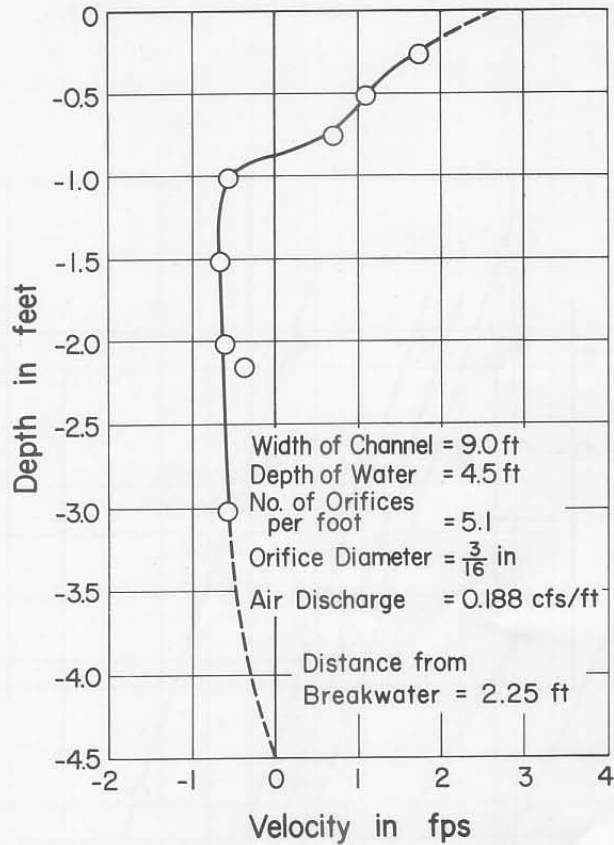


Fig. 16 - Velocity Distribution in Surface Current. Pneumatic Breakwater.

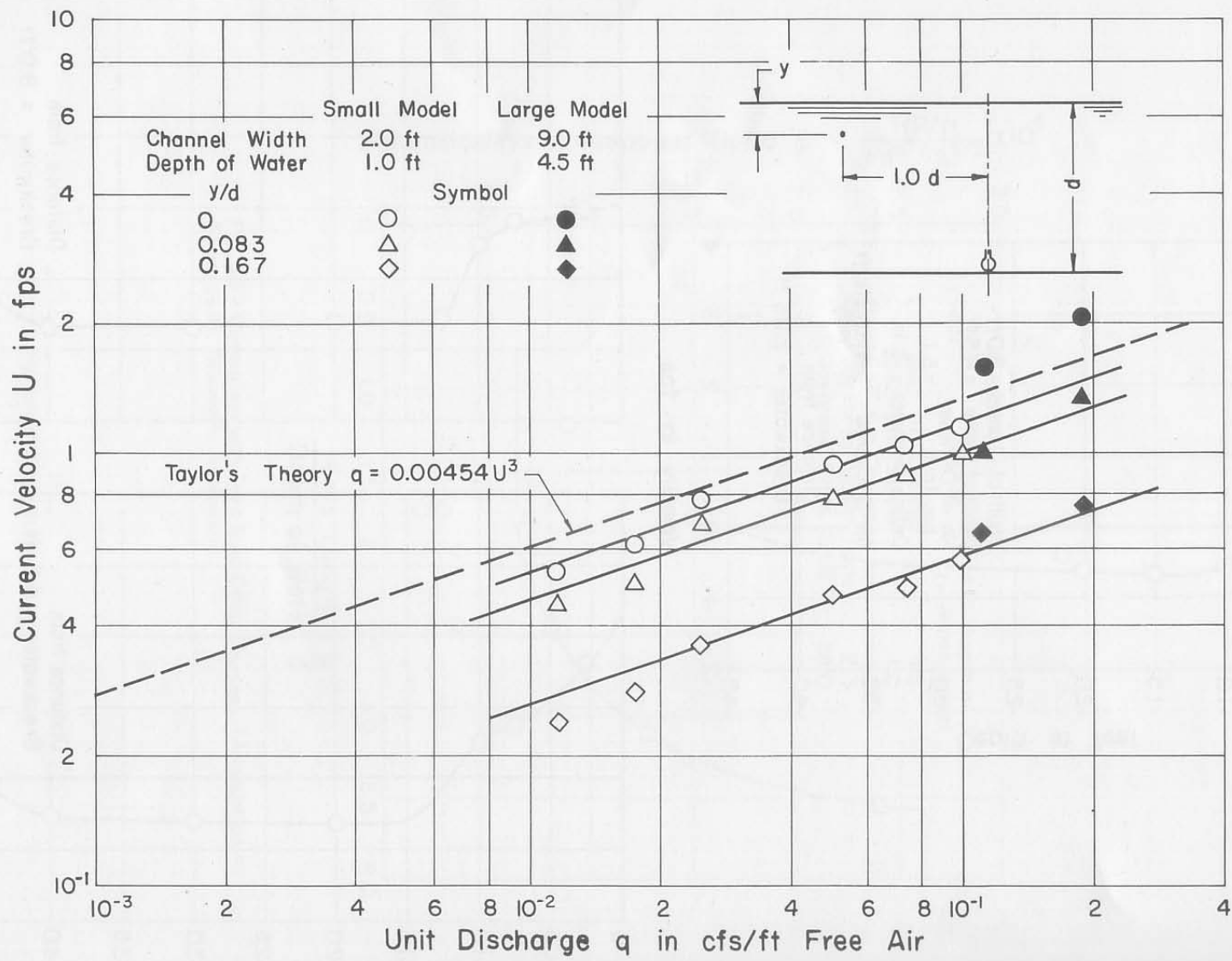


Fig. 17 - Surface Current Velocity as a Function of Air Discharge. Pneumatic Breakwater.

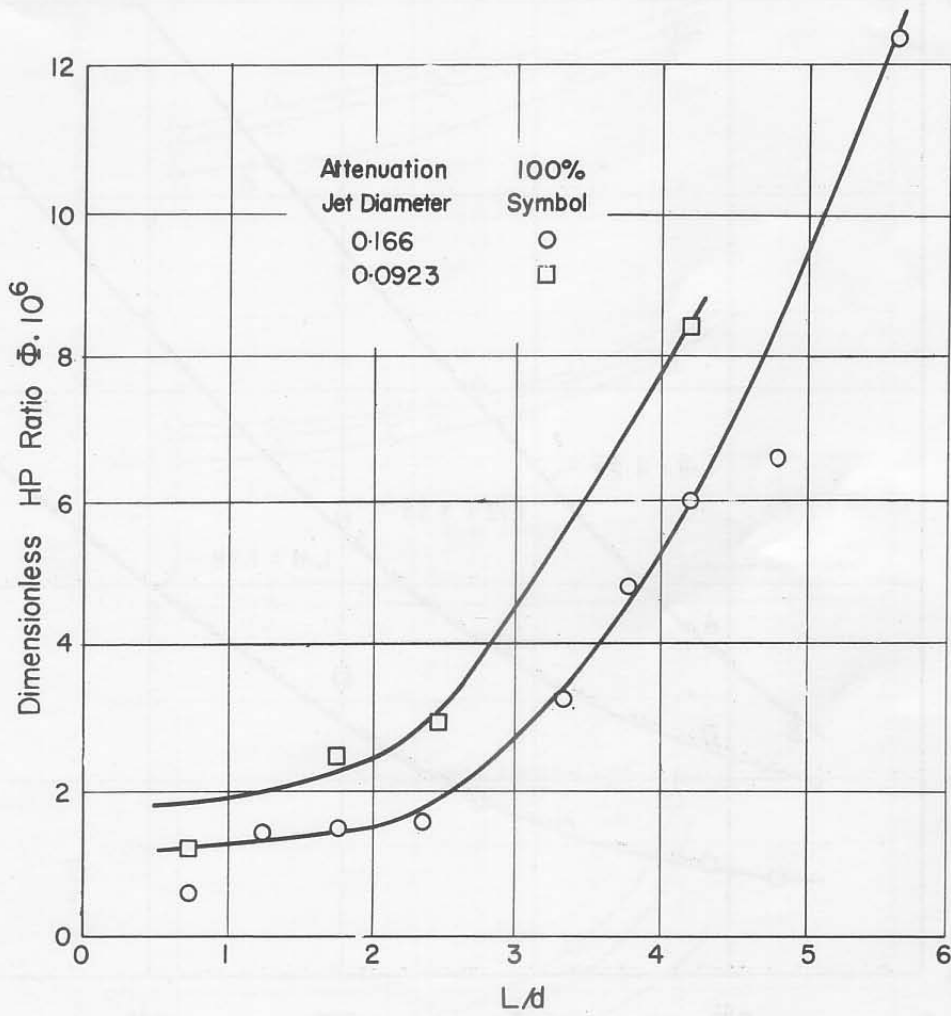


Fig. 18 - Horsepower Ratio as a Function of Length-to-Depth Ratio. Hydraulic Breakwater.

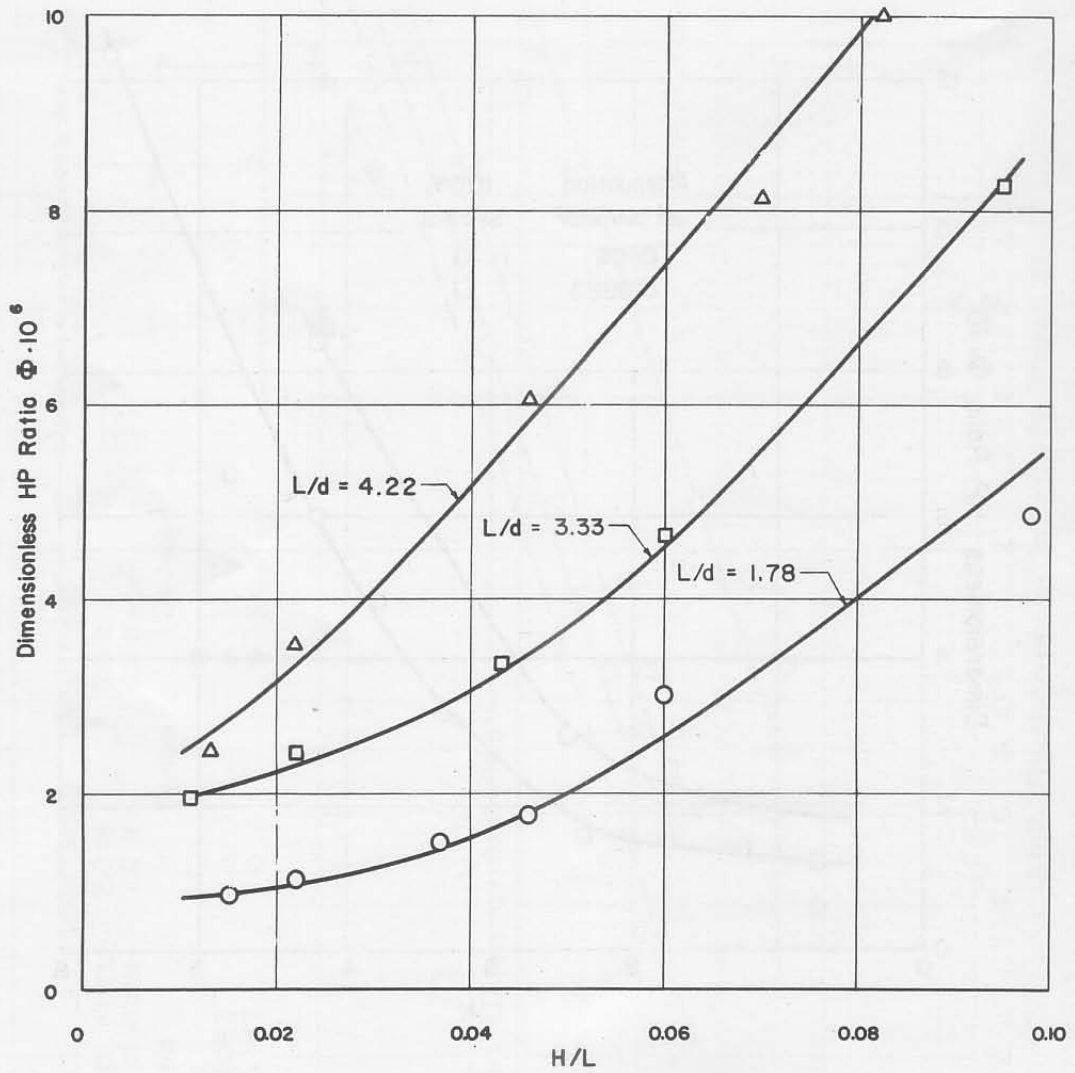


Fig. 19 - Horsepower Ratio as a Function of Wave Steepness. Hydraulic Breakwater.

WAVE CHARACTERISTICS

Channel Width 2.0'
 Water Depth (d) 1.0'
 Wave Steepness 0.04
 Attenuation 90% - ○
 80% - □
 60% - △

BREAKWATER CHARACTERISTICS

Jets per Foot 23.0
 Jet Diameter 0.0923"
 Number of Manifolds 1

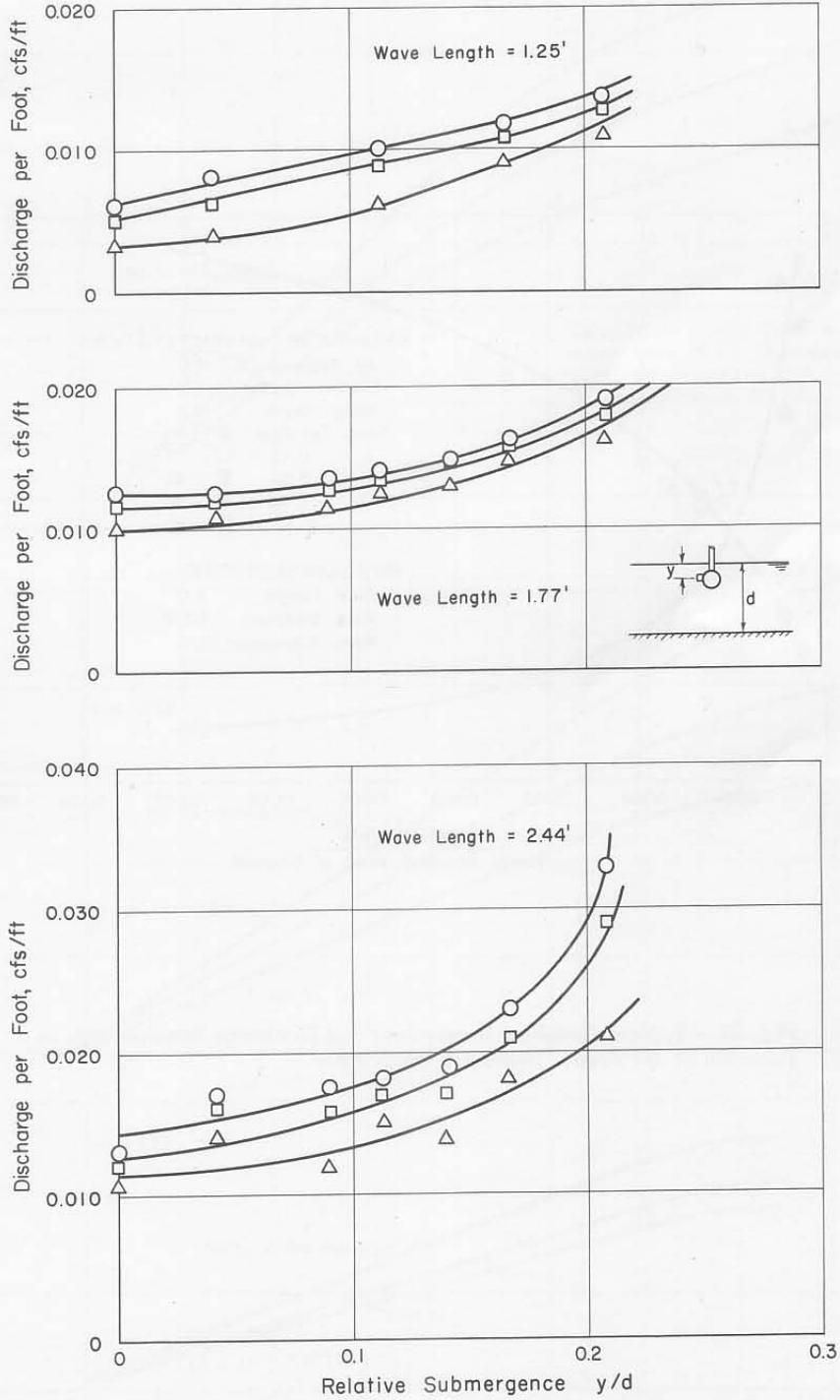


Fig. 20 - Effect of Jet Submergence on Discharge Requirement: Hydraulic Breakwater.

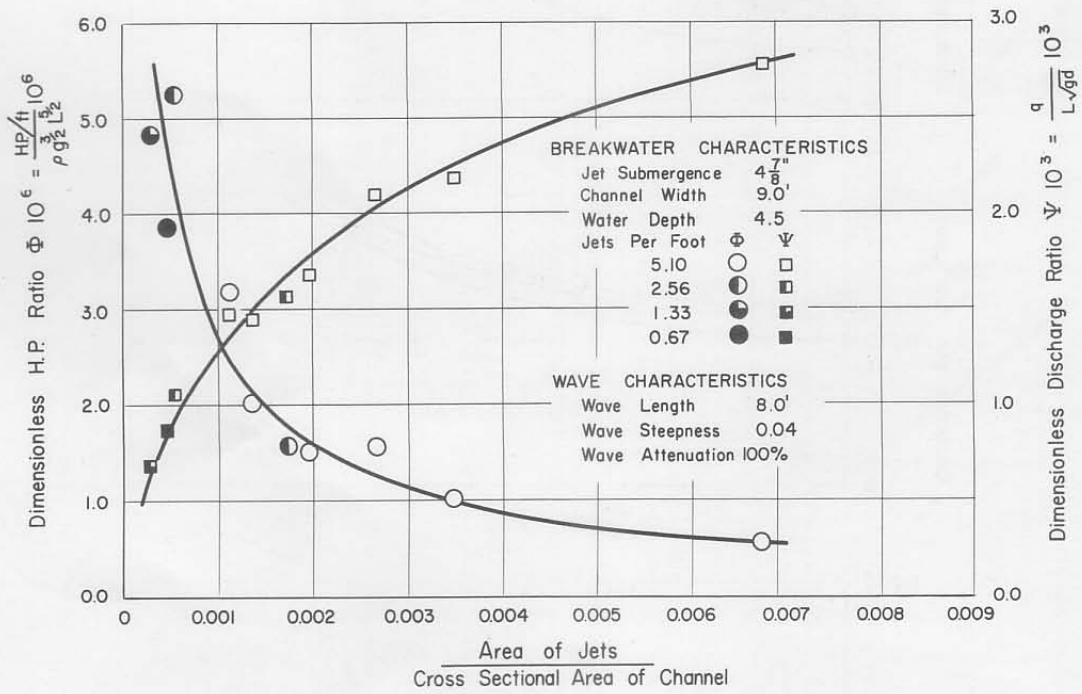


Fig. 21 - Typical Graph of Horsepower and Discharge Requirements as a Function of Jet Area. Hydraulic Breakwater.

WAVE CHARACTERISTICS

Channel Width = 2 ft
Water Depth d = 1 ft

BREAKWATER CHARACTERISTICS

Jets per foot = 23; 1 Manifold
Jet Diameter = 0.166 in.
Jet Submergence = 1.08 in.

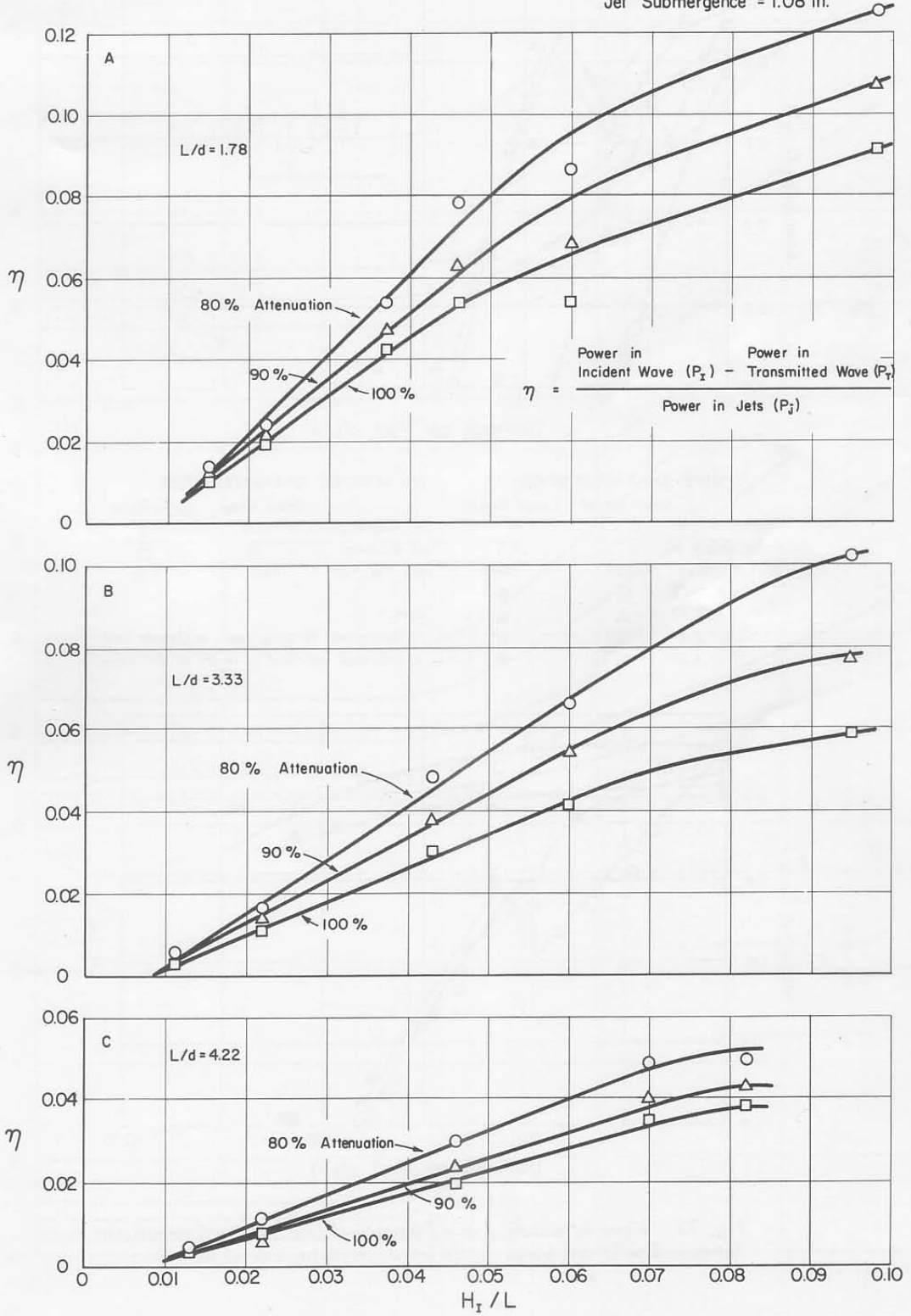
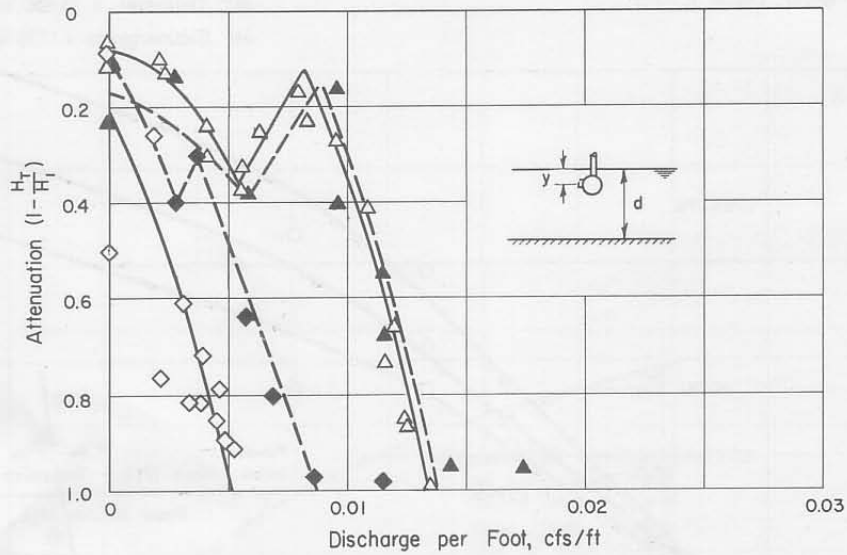


Fig. 22 - Efficiency as a Function of Wave Steepness. Hydraulic Breakwater.



WAVE CHARACTERISTICS			BREAKWATER CHARACTERISTICS		
	Small Model	Large Model	Small Model	Large Model	
Channel Width	2'	9'	Jet Submergence (y)	1.08"	4.7"
Water Depth (d)	1'	4.5'	Jet Diameter	3/32"	27/64"
Wave Steepness	0.04	0.04	Jets Per Foot	23.0	5.10
L/d	0.72	◆	Notes:		
	1.77	▲	1. Comparison of data based on Froude Law.		
	2.44	■	2. Discharge indicated in terms of 2ft model.		
	4.22	○			

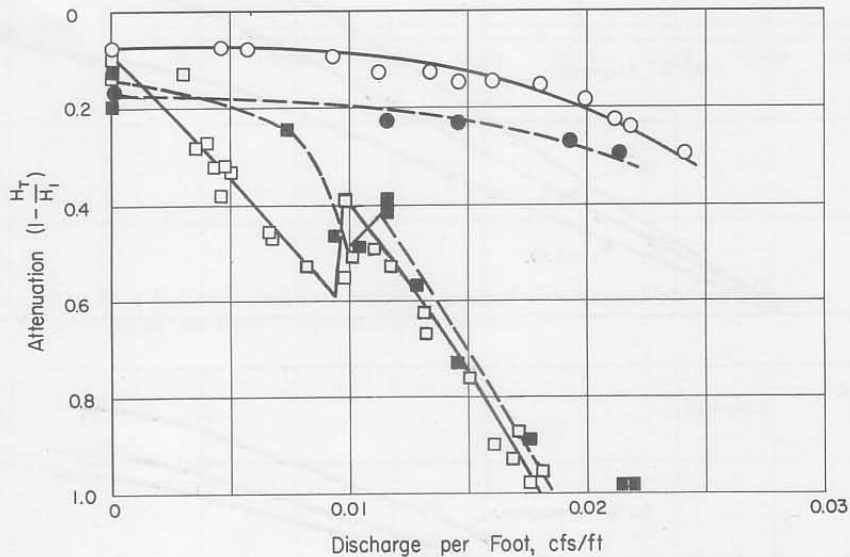


Fig. 23 - Wave Attenuation as a Function of Discharge. Comparison Between Small- and Large-Scale Data. Hydraulic Breakwater.

Attenuation = 100 %

Relative Jet Submergence $y/d = 0.0903$

Jet Discharge per Foot $q = 58.58 \cdot a_j^{2/3} \cdot \Phi^{1/3} \cdot L^{5/6}$

$$\Phi = \frac{HP/ft}{\rho g^{3/2} L^{5/2}}$$

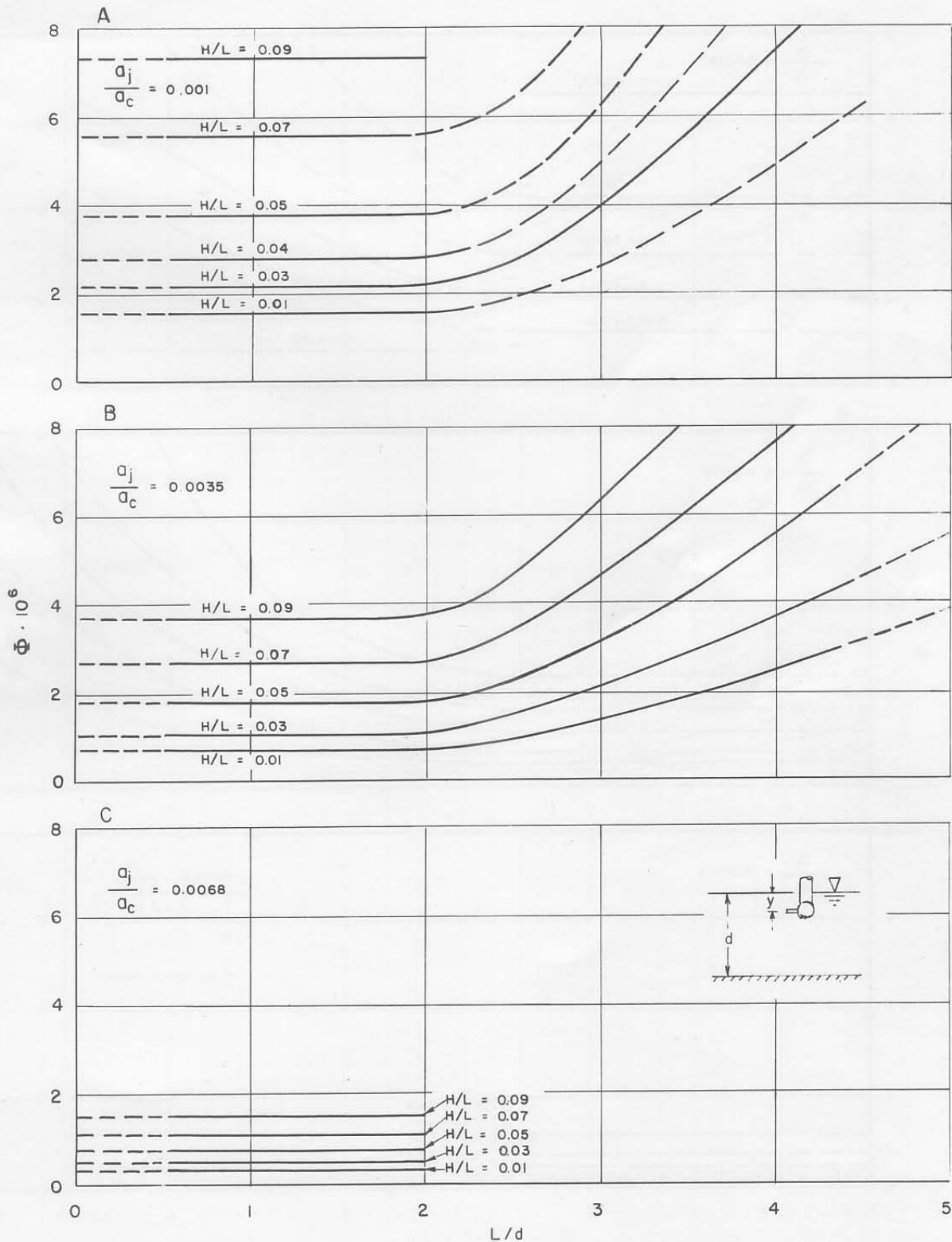


Fig. 24 - Summary Curves--Attenuation 100 per cent. Hydraulic Breakwater.

Attenuation = 80 %

Relative Jet Submergence $y/d = 0.0903$

$$\Phi = \frac{HP/ft}{\rho g^{3/2} L^{5/2}}$$

Jet Discharge per Foot $q = 58.58 a_j^{2/3} \Phi^{1/3} L^{5/6}$

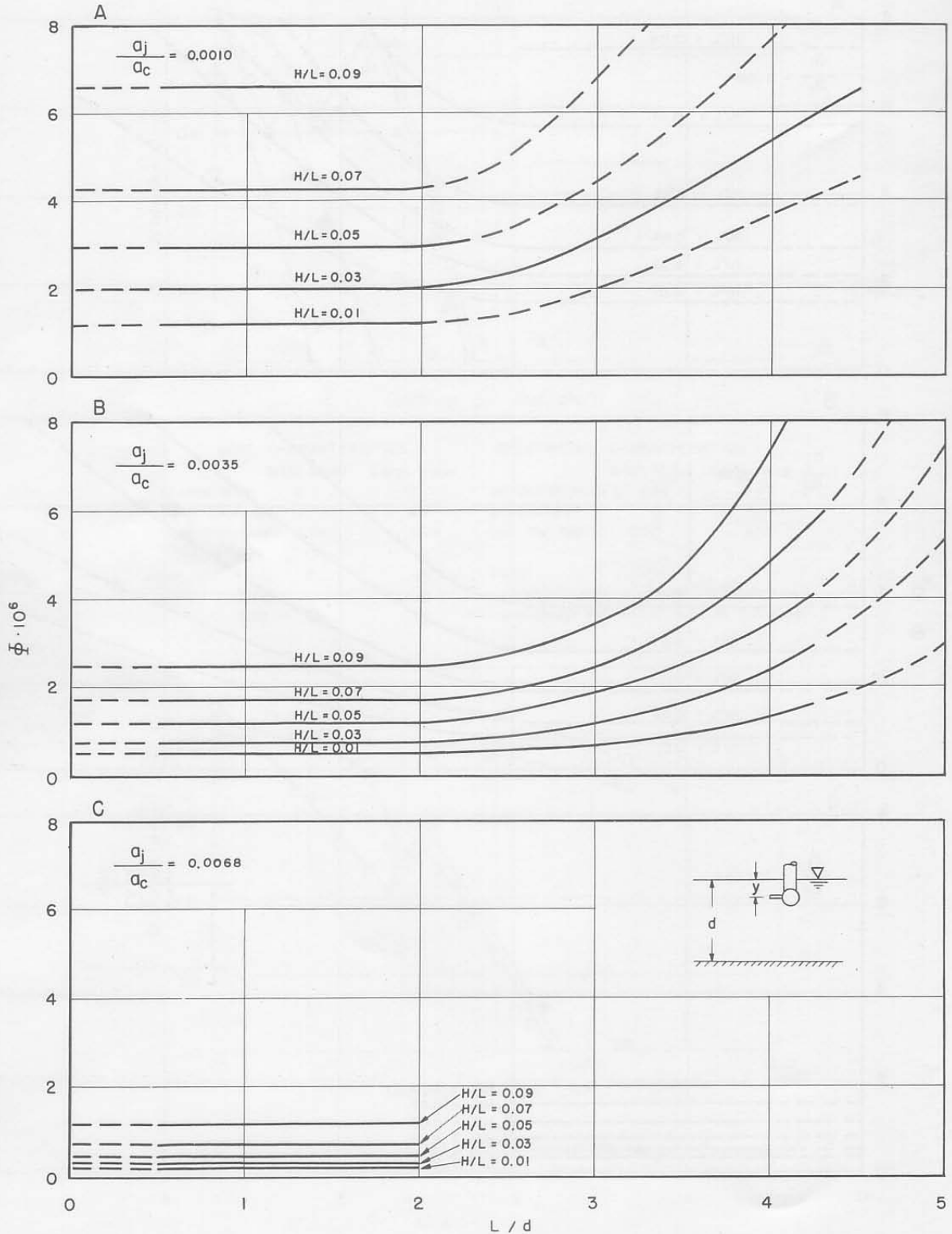


Fig. 25 - Summary Curves--Attenuation 80 per cent, Hydraulic Breakwater.

Attenuation = 60 %

Relative Jet Submergence $y/d = 0.0903$

$$\Phi = \frac{HP / ft}{\rho g^{3/2} L^{5/2}}$$

Jet Discharge per Foot $q = 58.58 q_j^{2/3} \cdot \Phi^{1/3} \cdot L^{5/6}$

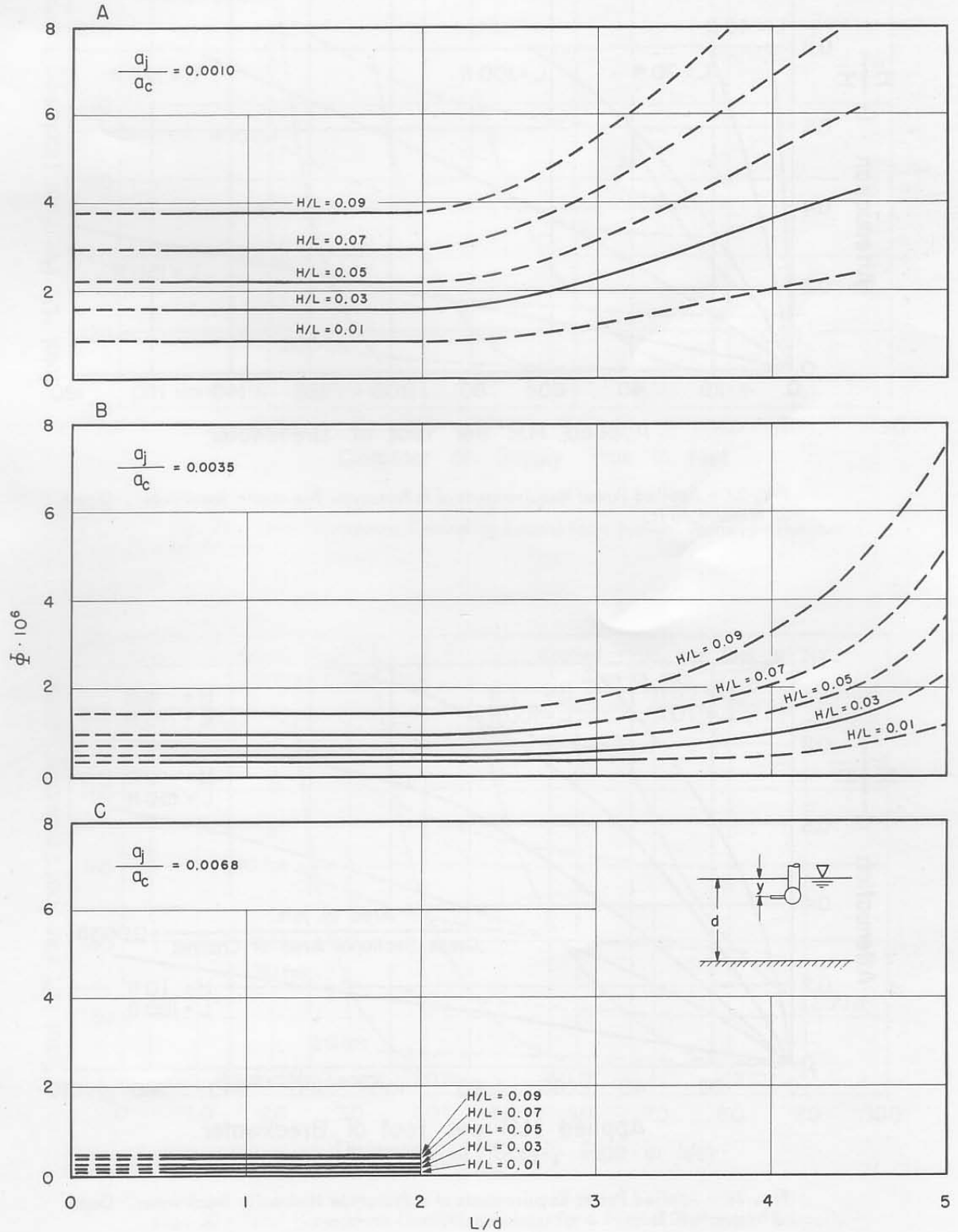


Fig. 26 - Attenuation 60 per cent. Hydraulic Breakwater.

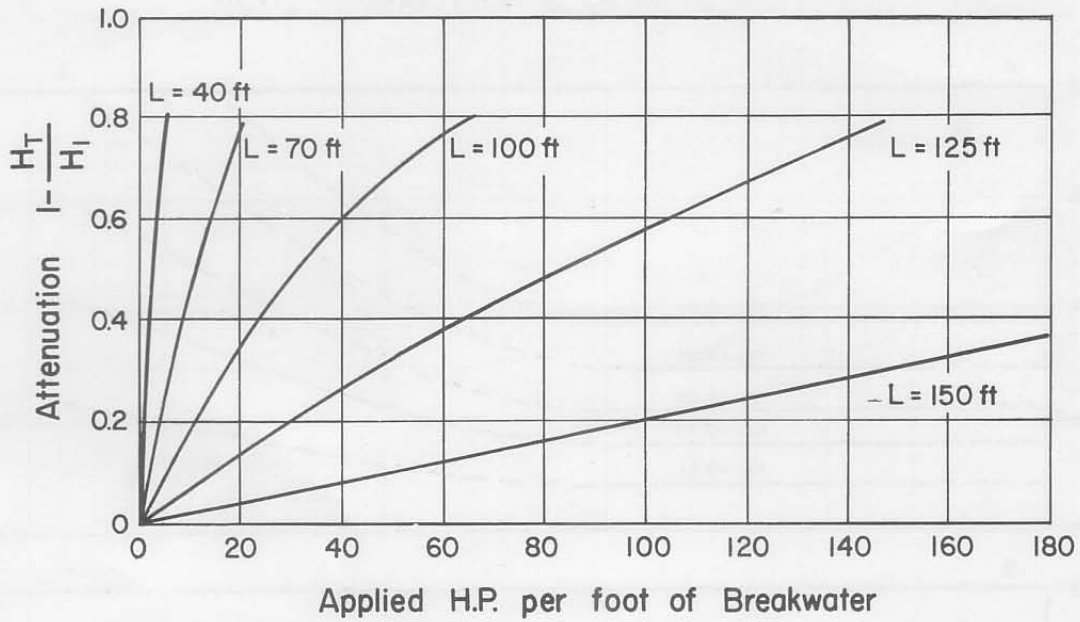


Fig. 27 - Applied Power Requirements of a Prototype Pneumatic Breakwater. Depth of Water = 40 ft.

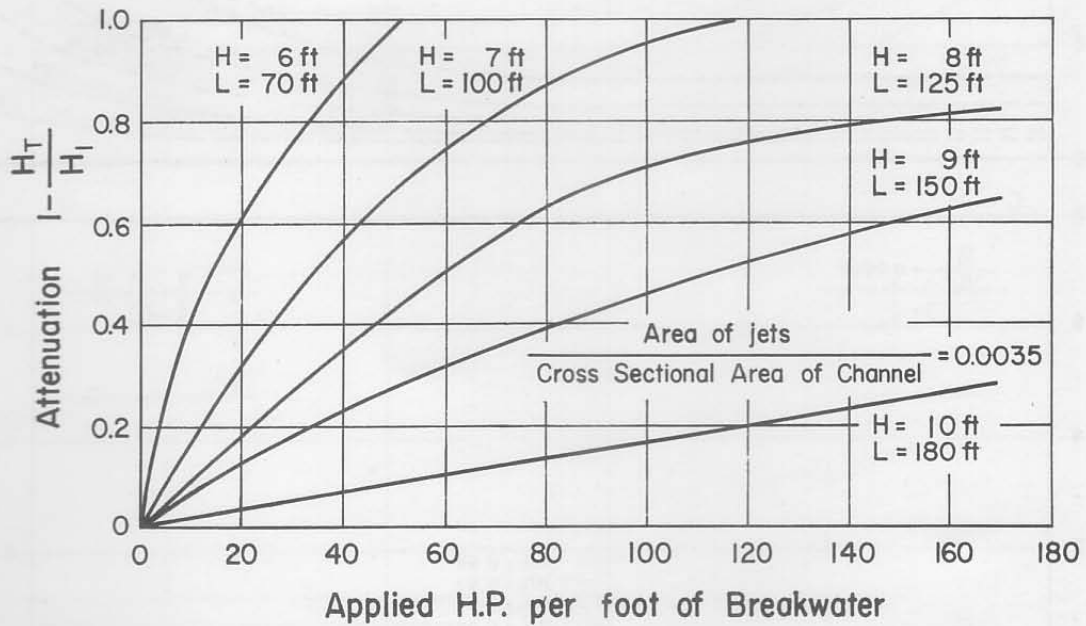


Fig. 28 - Applied Power Requirements of a Prototype Hydraulic Breakwater. Depth of Water = 40 ft.

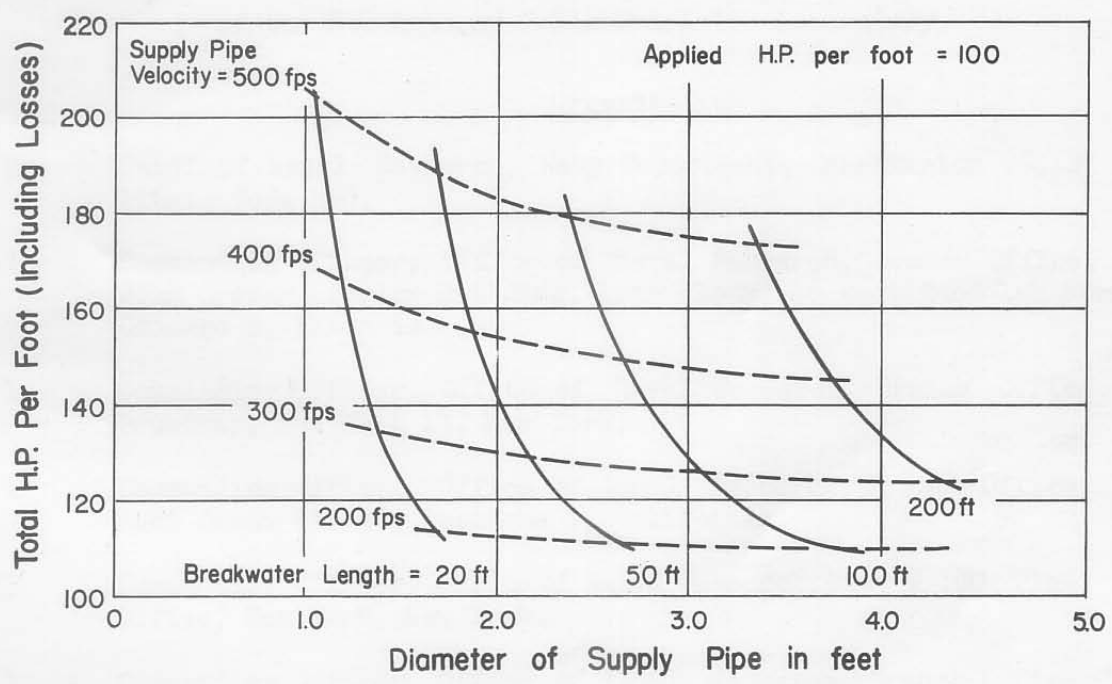


Fig. 29 - Total Horsepower (Including Losses) for a Typical Prototype Pneumatic Breakwater

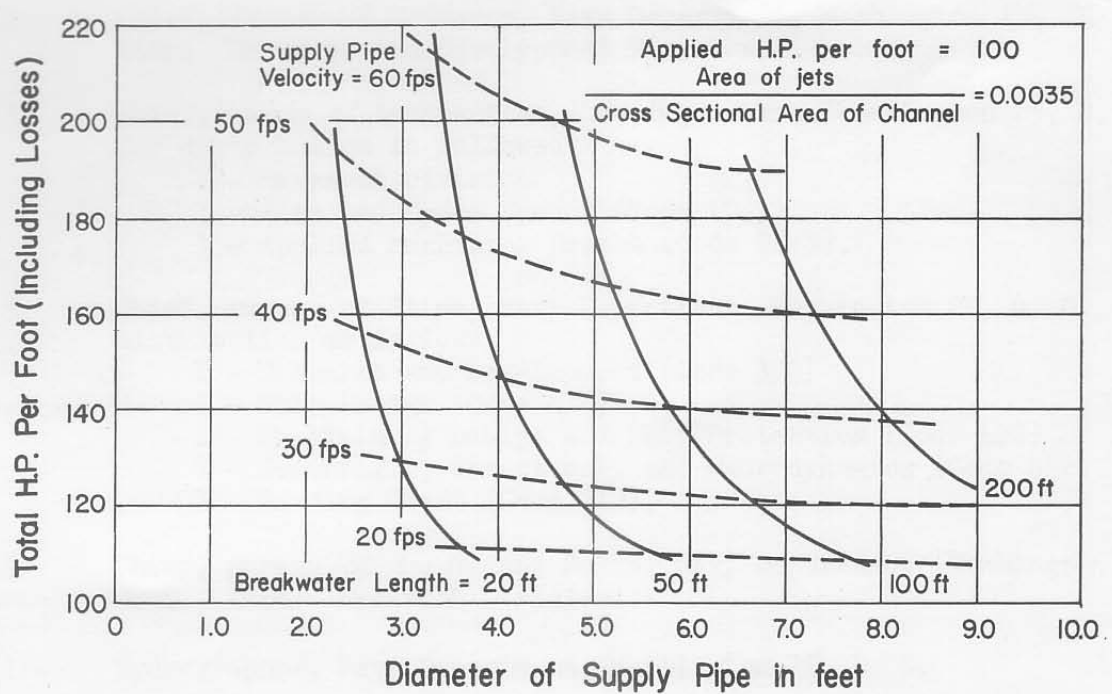


Fig. 30 - Total Horsepower (Including Losses) for a Typical Prototype Hydraulic Breakwater

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