THEORY AND DESIGN OF A WAVE GENERATOR FOR A SHORT FLUME

by

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ABSTRACT

This thesis describes the design of a proposed, new hydraulic laboratory, wave generator for use in a 39'-4¼" long, 30" wide and 36" deep flume, and the re-design of a small wave generator previously built for a 21'-3 5/8" long, 8 3/4" wide and 10 5/8" deep flume. These relatively short flumes are installed in the Hydraulics Laboratory of the Department of Civil Engineering, University of British Columbia, and the installation of the proposed, new wave generator would augment the present limited wave research facilities. The project is supported by an operating grant from the National Research Council of Canada.

The preparatory study of laboratory wave generators in use, presented herein, was made to determine how they function and their design problems. It was concluded that a rigid paddle, double articulation type would be best for generating deep-water, transition and shallow-water waves in a flume of relatively short length.

Biésel's wave generator theory is outlined and was used in estimating wave heights and in determining power and strength requirements.

The existing wave generator for the "small" flume is a rigid paddle, double articulation type. It did not function satisfactorily due to a very irregular paddle motion. The causes were

isolated and a new drive system designed and installed, with good results. The resulting new operating specifications are:

power	= 1. Hp (D.C.)
wave period range	= 0.34 to 2.1 secs.
design water depth	= 6.5"
estimated maximum wave height	≃ 4"

The proposed, new wave generator for the "large" flume is a rigid paddle, double articulation type designed around the adjustable paddle concept of G.D. Ransford (1949) as modified by Lt. C.B. Coyer (1953). The designed operating characteristics are:

power	= 10.Hp (D.C.)
wave period range	= 0.68 to 4.28 secs.
design water depth	= 25"
estimated maximum wave height	≃ 14"

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The author wishes to express his appreciation to the thesis readers for their constructive criticisms and suggestions on thesis presentation; to the staff of the Civil Engineering Workshop for assistance in making modifications to the existing wave generator, and to the National Research Council of Canada for providing financial support for this project.

LIST OF SYMBOLS

WAVE THEORY

C = Wave celerity (velocity of a wave).

L = Wave length.

H = Wave height.

d = Water depth from bottom to still water level.

x, y = Coordinates.

T = Wave period (secs.).

δ = Wave steepness.

g = Acceleration due to gravity.

ρ = Water mass density in slugs/cu.ft. ie. ($lb.sec.^{2}ft^{-4}$)

σ = Water surface tension (lbs./ft.)

α = Full horizontal amplitude of water particle displacement in a wave form.

β = Full vertical amplitude of water particle displacement in a wave form.

z = Depth below still water surface (measured positive downwards).

WAVE GENERATOR THEORY

see Fig. 17 for list of symbols and definitions.

WAVE GENERATOR - RECIPROCATING DRIVE THEORY

See diagram accompanying TABLE II or TABLE IV for symbols and definitions.

THEORY AND DESIGN OF A WAVE

GENERATOR FOR A SHORT FLUME

1. INTRODUCTION - defining the problem.

1.1. Background

The Department of Civil Engineering, University of British Columbia, plans to increase the facilities of the present hydraulics laboratory to permit undergraduate and graduate wave experiments and research. Laboratory space is not presently available for the installation of new wave channels or tanks of relatively large size. Therefore, it was decided to provide wave research facilities utilizing existing flume installations.

The hydraulics laboratory contains a large, fixed, steel and glass flume (Fig. 1) measuring 30" in width, 33" in usable depth and 39'-4¼" in length, and a small, tilting, steel and glass flume (Fig. 2) measuring 8 3/4" in width, 10 5/8" in usable depth and 21'-3 5/8" in length. The large flume could be used as a wave channel for engineering model studies if the flume was equipped with a suitable type of wave generator. The small flume is equipped with a wave generator which was fabricated in the Civil Engineering Department workshop and installed during the years 1965 and 1966. Unfortunately, this wave generator has operating problems which need correcting.

1.2 Thesis Objective

The object of this thesis is:

(a) to isolate and rectify the operating problems of the small flume, wave generator; and

(b) to design a wave generator for the large flume that will produce waves suitable for engineering model study purposes.

No single, complete, source of wave generator design information could be found in the literature. Therefore, a preparatory study was made to determine what wave forms the wave generator must produce, which wave generator type would be best suited for the large, relatively short, flume, the associated design problems, and the hydraulic operating specifications and to present pertinent theory. The final fulfillment of the two objectives is covered in paragraphs 5 and 6, respectively, of the thesis.

Since a complete design procedure for designing a wave generator from start to finish could not be found the basic procedure the author has used is his own.

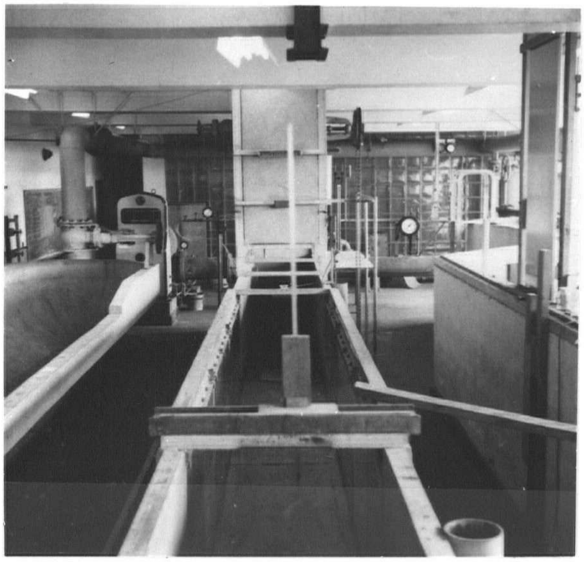


Fig. 1. Three Views of the "Large", 39'-4¼" Long, 30" Wide and 36" Deep, Fixed, Steel and Glass Flume.

(Existing Wave Generator in Foreground)

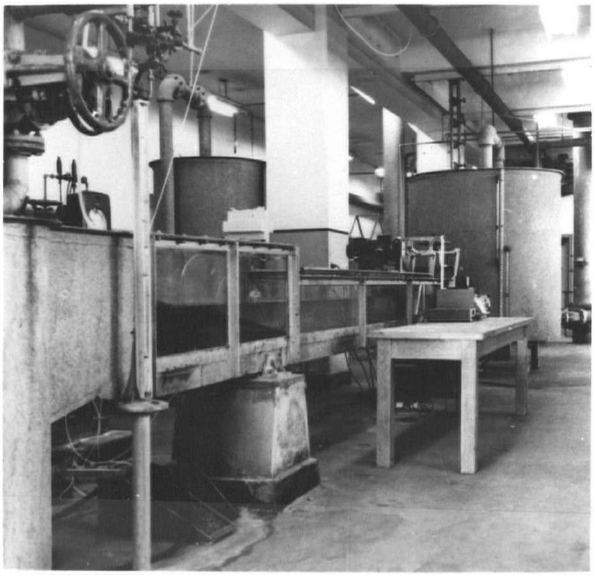


Fig. 2. Two Views of the "Small" 21'-3 5/8" Long, 8 3/8" Wide and 10 5/8" Deep, Tilting, Steel and Glass Flume.

2. WATER WAVES - defining the waves to be produced by the laboratory wave generator.

2.1 The Spectrum of Water-Surface Waves

On commencing the design of a water-wave generator it was first necessary to examine the water-surface wave forms existing in nature and to decide which of these forms were to be reproduced in miniature in the wave channel.

Surface waves (Ref. 7,16 and 18 - Wave Theory) can be classified on the basis of wave period as follows:

Classification	Period
Capillary waves	less than 0.1 secs.
Ultra-gravity waves	from 0.1 sec. to 1 sec.
Ordinary gravity waves	from 1 sec. to 30 secs.
Infra-gravity waves	from 30 secs. to 5 min.
Long-period waves	from 5 min. to 12 hrs.
Ordinary tides	from 12 hrs. to 24 hrs.
Trans-tidal waves	24 hrs. and up.

The classifications of interest are those of capillary waves, ultra-gravity waves and ordinary gravity waves.

Capillary waves in nature are generally caused by the wind but their mode of origin is in question. They are influenced more by surface tension than by gravity and hence are greatly affected by surface-active agents, such as oils and detergents. These waves are subject to rapid damping by viscous forces. Wave lengths are always shorter than about 0.44 inches while the corresponding wave velocity of about 0.95 ft. per sec. is always exceeded since capillary wave velocity increases with decreasing period and length. The behaviour of these waves is considerably different from that of ultra-gravity and ordinary gravity waves.

Ultra-gravity waves in nature are wind-generated waves lying in the transition zone between capillary waves, which are more influenced by surface tension than gravity, and ordinary gravity waves where gravity is the predominant influence and surface tension can be neglected.

Ordinary gravity waves in nature are wind-generated waves for which gravity is the primary restoring force.

The wave generator to be constructed will be used to produce scaled down replicas of <u>ordinary gravity waves</u> for use in hydraulic model studies. Due to the reduction in scale, the equivalent laboratory waves will lie in the <u>short-period end of the ordinary gravity wave band</u> and the <u>long-period end of the ultra-gravity wave band</u>. The lower end of the ultra-gravity wave band, and especially the capillary wave band, should be avoided due to undesirable surface tension effects.

2.2 Physical Characteristics of Gravity Waves

Since the laboratory wave generator will be designed to produce short-period, ordinary gravity waves and long-period, ultra-gravity waves, as miniature replicas of larger gravity waves in nature, it is desirable to review the observed physical characteristics of the latter.

The profile of a gravity wave with its associated terminology is shown in Fig.3. Each wave form moves over the still water surface with a velocity or celerity C. The distance from crest to crest is the wave length L. The wave height H is measured from crest to trough. The depth of water d is measured from the still water level to the ocean bed. The origin of coordinates x and y is a point below mid-crest on the still water level line. The time for two successive wave crests to pass a fixed point is defined as the wave period T.

An important, physical, wave classification depends upon the length of the wave relative to the depth of the water expressed as $\frac{L}{d}$. When a wave moves along the surface it causes water particles beneath it to move up and down as well as to and fro. This motion dies out with depth. If the bottom is far enough away from the still water level, relative to the wave length, it does not interfere with the wave motion and the water particle motion is circular of decreasing diameter with depth (Fig.4). If the bottom is near the surface, insufficient room is left for the water particle vertical motion to develop and the particle motion becomes elliptical (Fig.4). Physically then, the water is either deep or shallow when compared with the wave length. If the value of $\frac{L}{d}$ is small (usually taken as $\frac{L}{d} \leqq 2$), the wave is a deep-water wave. If $\frac{L}{d}$ is large ($\frac{L}{d} \geqq 20$) the wave is a shallow-water wave. This ratio shows that a tsunami with a wave length of hundreds of miles is a shallow-water wave in the deepest parts of the ocean, while ripples may be deep-water waves in a pond a foot deep. Since other aspects of wave motion remain similar for a given $\frac{L}{d}$ ratio, this classification assumes importance in hydraulic model wave studies and is used to relate the motion of the waves in nature to the scaled down replicas produced in a laboratory.

Another significant physical feature of waves is steepness. This is the ratio of the wave height to the wave length, $\delta = \frac{H}{L}$. Waves have a limiting value of steepness where they commence to break, which occurs when the centripetal acceleration of the water particles at the crest is g. This makes sense physically, since, if the acceleration was greater than g then some sort of downward pressure to supplement gravity would be required to keep the wave from flying apart.

Steepness and the $\frac{L}{d}$ ration are the two factors used in establishing geometric similitude between natural waves and the scaled down laboratory waves.

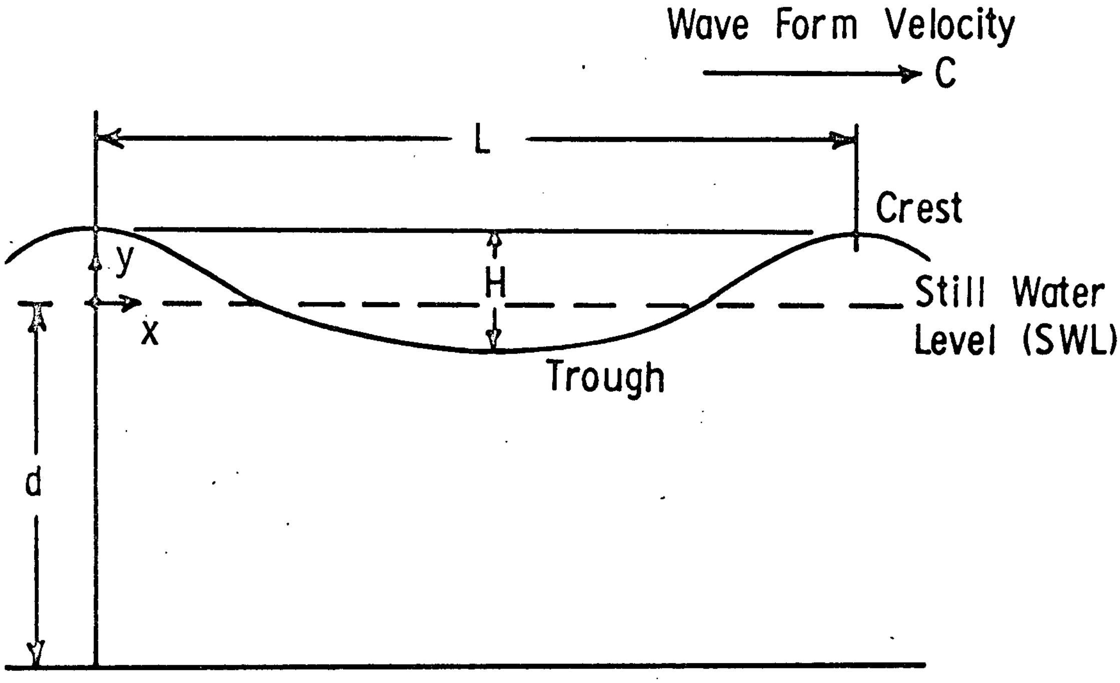


Fig. 3 Wave Profile and Associated Terminology

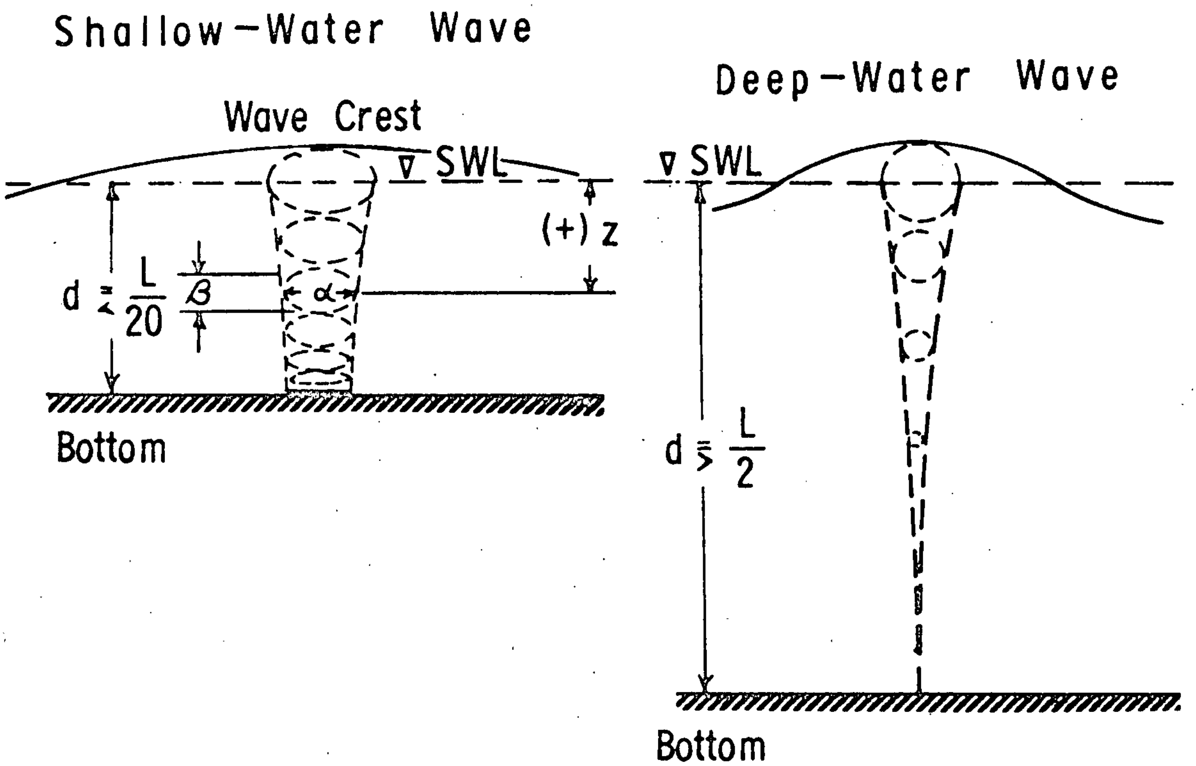


Fig. 4 Deep and Shallow-Water Wave Particle Motion

2.3 Theory of Gravity Wave Motion

Mathematical theories describing the behavious of gravity waves in water have been developed by Laplace (1776), Gerstner (1802), Airy (1845), Stokes (1847), Froude (1862), Rankine (1863), Havelock (1918), Levi-Civita (1925), Struik (1926), Lamb (1932), Biésel (1952), Crapper (1957) and others. Unfortunately, which theories most closely describe the physical characteristics of wind driven gravity waves in nature, are as yet unknown, and the accuracy with which these theories describe waves in a laboratory wave channel changes slightly for waves of different periods. Obviously, the choice of theory to describe gravity wave motion is arbitrary. The results of the theories of Airy and Lamb are outlined below since they are the most widely known and are mathematically easy to use.

If waves are of small amplitude compared to their length and to the depth of the water, the wave profile closely approximates a sine curve. Terminology for this theory is illustrated in Fig. 3.

The rate of travel or celerity (c) of an individual wave form (Lamb, 1932), considering both gravity and surface tension, is:

$$c = \sqrt{\frac{g\,L}{2\,\pi} + \frac{2\pi\sigma}{\rho\,L}\quad \tanh\frac{2\pi d}{L}} \qquad (1)$$

The relative effects on velocity, of the gravity and the surface tension components for deep-water waves, are presented in Fig.5. Since the calculation of the percent difference in Fig.5 is not affected by the wave being either a deep-water or a shallow-water wave type, it is evident that, for any wave more than one foot in length, the effect of surface tension may be safely omitted when calculating the wave form velocity.

Neglecting surface tension effects, the equation for velocity of propagation of gravity waves (Airy, 1854; Lamb, 1932) becomes

$$c = \sqrt{\frac{gL}{2\pi} \cdot \tanh\frac{2\pi d}{L}}\,. \qquad (2)$$

For water deeper than one-half the wave length, that is $\frac{L}{d} \leqq 2$, the term $\tanh\frac{2\pi d}{L}$ is almost equal to 1 and the equation reduces to the "deep-water" wave equation $c = \sqrt{\frac{gL}{2\pi}}\,.$ (3)

For water depths less than $\frac{1}{20}$ of the wave length, that is $\frac{L}{d} \geqq 20$, the term $\tanh\frac{2\pi d}{L}$ approaches the value of $\frac{2\pi d}{L}$ and equation (2) reduces to the "shallow-water" wave equation

$$c = \sqrt{gd}\,. \qquad (4)$$

Actually, there is no abrupt change from "deep" to "shallow" water waves and the depths for which these simplified equations are no longer applicable depends upon the degree of accuracy desired in calculations. But for most engineering studies it has become custom to call waves having $\frac{L}{d} \leqq 2$ "deep water" waves (equation 3) and waves having $\frac{L}{d} \geqq 20$ "shallow-water" waves (equation 4). Waves in between are known as "transition" waves; where equation (2) applies.

The relationship between length, period and velocity is defined by

$$L = CT. \qquad (5)$$

The surface profile is given by the sinusoidal equation

$$y = \frac{H}{2} \cos \frac{2\pi}{L} (x - ct), \qquad (6)$$

where x and y are measured as in Fig. 3.

The profile and celerity equations given above are, mathematically, only first approximations to the theoretical solutions, but are quite satisfactory for most practical purposes. Although the celerity equations only apply strictly to waves of small amplitude, they are sufficiently accurate for most cases except when the wave is becoming large and steep enough to break.

In Fig. 4, α and β are the full horizontal and vertical displacements, respectively, of a water particle as it describes an orbit about its average depth (Z) below the still water surface. Values for α and β can be calculated using the following equations:

$$\alpha = H \frac{\cosh \frac{2\pi(d-z)}{L}}{\sinh \frac{2\pi d}{L}} \qquad (7)$$

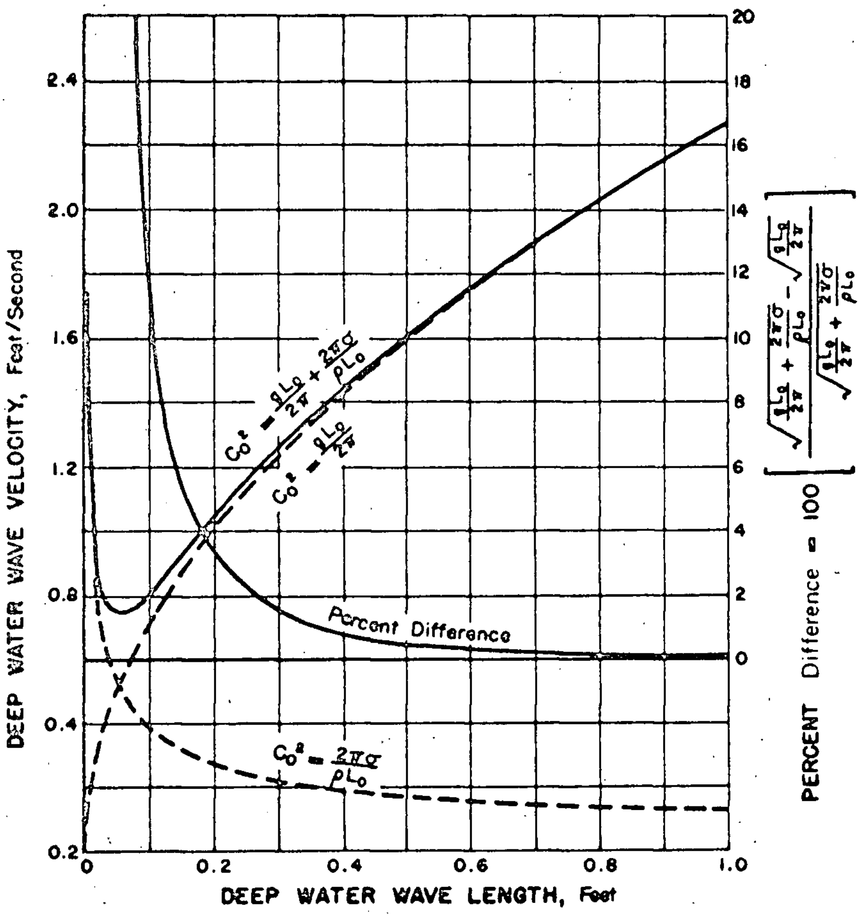


Fig. 5 Effect of Surface Tension on Deep Water Wave Velocity in Fresh Water at 70° F. (Wiegel)

$$\beta = H \frac{\sinh \frac{2\pi(d-z)}{L}}{\sinh \frac{2\pi d}{L}} \tag{8}$$

Waves have a limiting value of steepness. Michell (1893), found found the maximum possible steepness for a Stokes deep-water wave to be

$$\delta \text{ max.} = \frac{H}{L} = 0.142 \simeq \frac{1}{7}. \tag{9}$$

Theory suggests that shallow water waves will break when

$$\frac{H}{d} = .781 \simeq \frac{3}{4} \tag{10}$$

A graphical presentation of theoretical information useful in calibration of wave generators of the double articulation type to be designed, is given in Fig. 6.

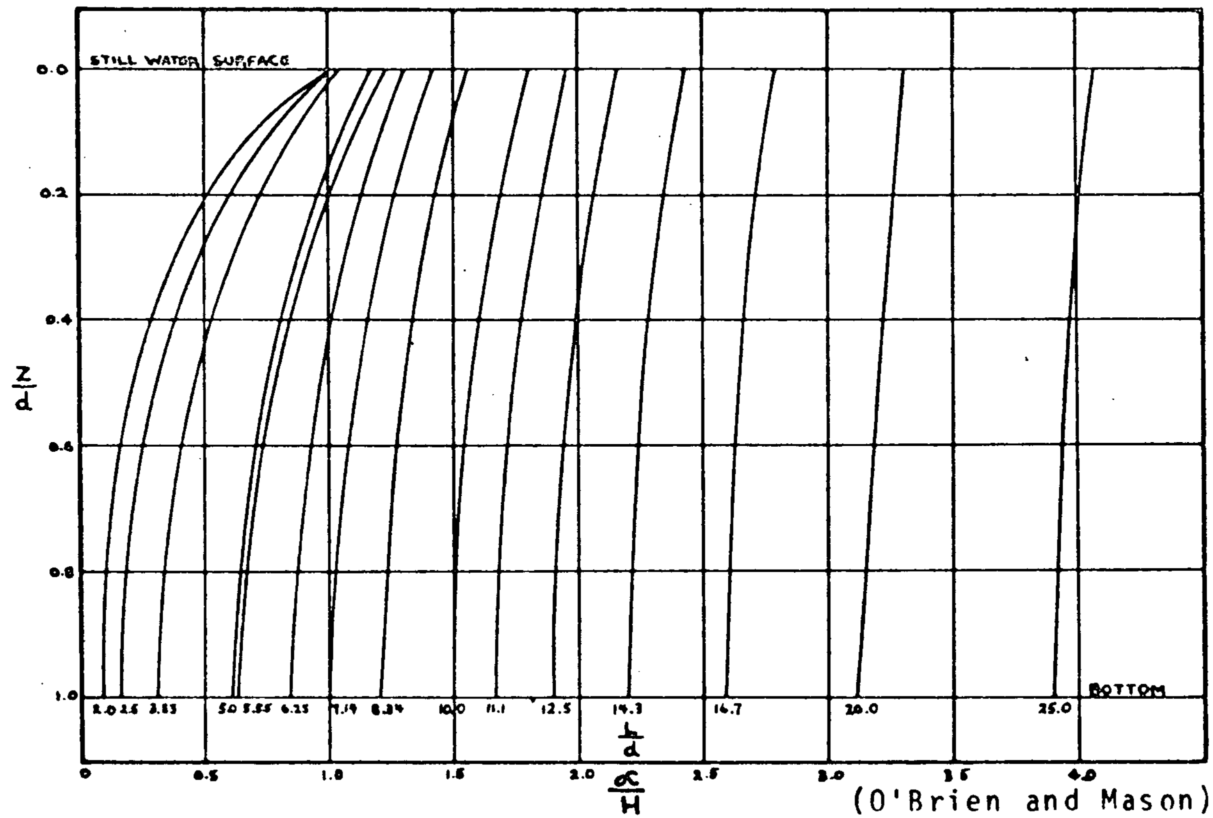
Note: $\frac{\alpha}{H} = \dfrac{\cosh \frac{2\pi}{L}(d-z)}{\sinh \frac{2\pi d}{L}}$ WHERE Z= average depth of orbiting particle below still water surface and measured positive downwards.

d= depth of water

α= full horizontal amplitude of particle oscillation

H= wave height

L= wave length



(O'Brien and Mason)

Fig. 6 Relationship of Ratio $\frac{\alpha}{H}$ to Ratio $\frac{L}{d}$ at Proportional Water Depths $\frac{Z}{d}$ for use in Wave Generator Calibration.

3. WAVE GENERATORS - gaining an understanding of the types in use and their design problems.

3.1 Wave Generator Basic Design Requirements

3.1.1 Kinematics of the Wave Generating Member

There are a number of basic requirements to be considered when designing a laboratory wave generator. The first of these, the kinematics of the wave generating member, is of particular importance when the wave generator is to be designed for use in a flume of relatively short length.

Any periodic disturbance in one end of a wave channel, if of sufficient strength, will originate a train of waves which runs along the channel to the opposite end; but, if the generating member fails to conform with certain kinematic requirements, the distance along the channel at which the wave train stabilizes may be quite long, and the type of wave which results may be wholly undesirable.

The deformation of a vertical plane in water, upon passage of either a shallow-water wave or a deep-water wave, corresponds to the wave's respective envelope of water particle motion (Fig.4). The wave generator must approximate, as closely as possible, this deformation appropriate to the wave desired, if stable waves with correct particle motion are to be produced in the immediate vicinity of the generating member.

Conformance to this kinematic requirement allows a relatively short wave channel to be utilized with resultant compactness and space economy. Also it conserves power required to generate the wave by forming only the wave which is desired. On the other hand, the accurate simulation of the orbital motion of the liquid particles in a wave increases the mechanical complexity of a wave generator, necessitating a degree of compromise. (Ref. 1 - Wave Generator Theory).

3.1.2 Mechanical Requirements

Most wave generators utilize oscillating members to initiate waves. Since the oscillation rate may amount to 2 or 3 cycles per second it is important to keep the inertia of the oscillating members relatively low. High inertia gives rise to two detrimental effects: first, the motor power required to operate the generator is greatly increased, and second, the motion of the generator is made more irregular for any given power input. In the latter case departures from the assigned motion produce unwanted irregularities in the generated waves.

Rigidity of drive members has proved very important in wave generator operation. Any form of flexing alters the drive motion slightly with undesirable effects on the wave forms produced. Therefore, although the weight of moving parts must be kept reasonably light to reduce inertial effects, it is imperative that lightness not be achieved at the expense of rigidity.

Materials used in constructing the wave generator should be resistant to corrosion in water. Also, maintenance will be reduced if moving parts such as bearings, gears and rollers are positioned clear of the water to avoid the damaging effects of water borne sediment.

3.1.3 Wave Reflection

In nature, waves reflected from a structure depart seaward where they are finally dissipated. In a laboratory wave channel, waves reflected from the model of a structure propagate back towards the wave generator. Ideally, the wave-generator should not again reflect these waves but unfortunately, most generators are good reflectors.

Two solutions exist for this problem. The first, and most common, is the use of a filter in front of the generator to reduce the amplitude of the reflected wave to an acceptable height. The second is the use of special wave generators producing either little, or no reflections; or special devices to deal with reflected waves. Examples of such special wave generators are the pneumatic types, to be described later, and those used at the Delft Hydraulic Laboratory in Holland which employ a strong air stream to increase the size of the generated wave and attenuate the reflected waves.

3.1.4 Control Adjustments

Model tests in a wave channel may require the use of either deep-water, transition, or shallow-water waves of different heights.

Therefore, the generator must be capable of adjustment to produce waves of different lengths and heights in a particular depth of water. The adjustments should be simple to make and scales whould be provided which ensure the accuracy of the setting.

3.1.5 Mobility

Mobility is not usually a requirement for a wave generator designed for use in a wave channel.

3.1.6 Water Leakage

Most wave generators use a paddle of some form to generate waves. Leakage of water around the bottom and sides of the paddle can account for the following poor, wave characteristics:

a) decrease in wave amplitude;
b) instability in the wave form; and
c) incorrect wave profile.

On the other hand, if leakages are effectively sealed, the increased friction introduced by gaskets or other sealing devices may result in irregular generator motion. Most frequently a compromise is required.

3.2 Types of Wave Generators

3.2.1 Scope

Wave generators presently used in hydraulic laboratory wave channels assume a wide variety of forms. Fortunately, they may be loosely grouped in a few markedly different categories. Within each category there are as many different variations in the basic principle as there are designers. Some of these variations have been employed to emphasize one or more of the basic generator requirements already mentioned, especially kinematic requirements.

A comprehensive review of different types of wave generators is given in the publication "Laboratory Wave-Generating Apparatus", (Ref.1 - Wave Theory), which is a translation of a series of four French articles in La Houille Blanche by F. Biésel, F. Suquet and a group of engineers at the Laboratoire Dauphinois d' Hydraulique (Neyrpic), in France. A synoptic table from this publication listing types of water wave generators and showing their principles of operation has been reproduced in Fig.7.

3.2.2 The Movable-Wall Type

This general category includes those types whose wave-generating members are immersed in water and oscillated back and forth in accordance with some established law. These machines, although wave reflecting, are very versatile, since by imposing an appropriate law of motion they will generate waves of any desired characteristic.

The first machine of this category shown in Fig.7 is a flexible flap type. It is capable of closely approximating water particle motion resulting in waves with stable and correct motions being produced immediately in front of the flap. This is an important advantage in channels of limited length. Also the machine's inertia is small, contributing to regularity of oscillation. The disadvantage is the complicated mechanism which is not readily adjustable, together with high maintenance due to cranks and bushings being located underwater.

Next shown in Fig. 7 are three versions of rigid flap, wave generators with single articulation. These machines are of simple construction being hinged at one point and driven by a rotating crank and connecting rod. There is the undesirable feature of an underwater hinge in only two cases. They have low inertia, and are easily adjusted. The motion imparted to the water in the first two examples is that for deep-water waves. Therefore, if transition or shallow-water waves are required, a long channel is necessary to permit the correct wave profile and particle motion to develop, and wave height will be limited. Locating the hinge above the channel bed, as in the second example, may be necessary in the case of a model with a movable sand bed. The example showing articulation above the free surface is not very satisfactory since it ignores the prime kinematic requirement of wave generation - namely the decrease in size of water particle orbits with depth. Water leakage under the paddle results in low wave heights and may result in wave instabilities.

The third wave generator type in the movable wall category is the piston. It is a simple machine actuated by a rotating crank and connecting rod. The piston moves back and forth in the channel's longitudinal direction with usually a simple harmonic

Fig. 7 SYNOPTIC TABLE OF VARIOUS WAVE GENERATORS

(Biésel, Suquet and Others)

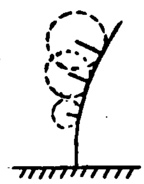
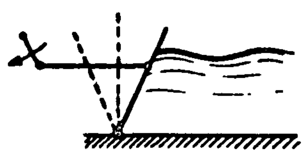
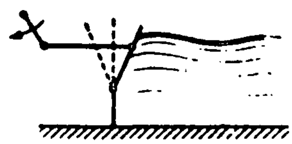
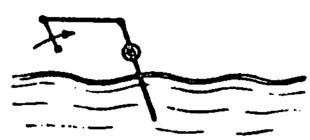
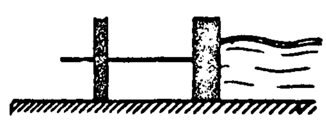
Type of Wave Generator		Diagram of Principle	Amplitude
Flexible flap			
Rigid flap with single articulation	Articulation at channel bed		Calculable
	Articulation above channel bed		
	Articulation above free surface		Not calculable by Biésel's theory
Piston			Calculable

Fig. 7 SYNOPTIC TABLE OF VARIOUS WAVE GENERATORS (Cont'd)

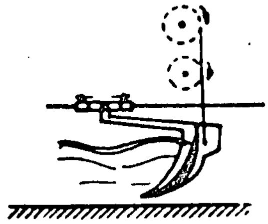
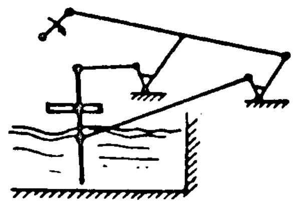
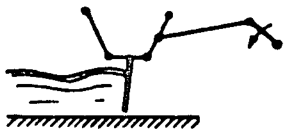
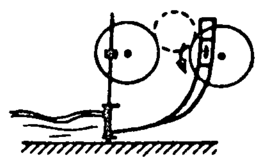
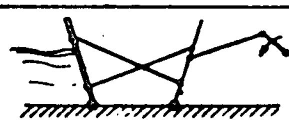
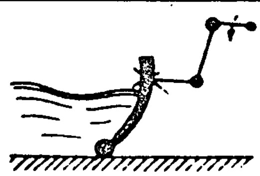
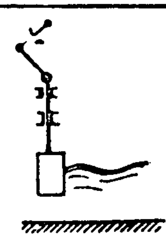
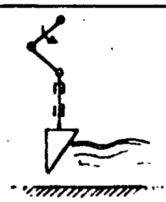
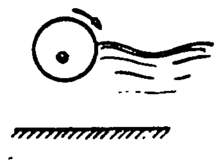
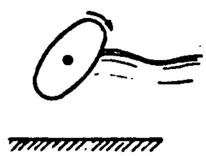
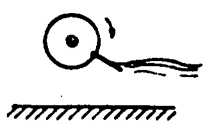
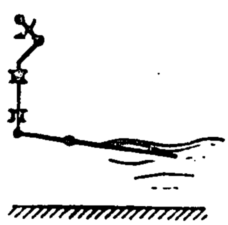
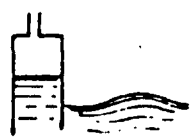
Type of Wave Generator		Diagram of Principle	Amplitude
Rigid paddle with double articulation	Suspended paddle		Calculable if the paddle is plane
			
			
			
	Paddle in contact with bed by means of rollers or slides		
			
Plunger			Generally not calculable by Biésel's theory
			

Fig. 7 SYNOPTIC TABLE OF VARIOUS WAVE GENERATORS (Cont'd)

Type of Wave Generator		Diagram of Principle	Amplitude
Various Devices	Serpent		Calculable
	Eccentric roller		Not calculable by Biésel's theory
	Elliptical cylinder		
	Cylinder with paddles		
	Paddle wheel		
	Surface paddle		
	Pneumatic device		

motion. The piston type is particularly suited for generating shallow-water waves, which means that, when deep-water waves are required, a relatively long channel, to permit the waves to stabilize, is needed.

The fourth group in the movable wall category is composed of rigid paddle generators with double articulation, six examples of which are shown on the second page of Fig.7. The lower end of the paddle of this group can be adjusted to traverse a longitudinal distance different to that traversed by the top end. Assuming that the top and bottom ends of the paddle are adjusted to travel the same distance, the generator then operates as a piston type and is particularly suited for generating shallow-water waves. If the bottom end of the paddle is adjusted for zero travel distance, the generator operates as a flap with single articulation and is particularly suited for generating deep-water waves. If the bottom end of the paddle is adjusted for a travel distance somewhere between zero and the distance travelled by the top end, the motion is then suited to the generation of a transition wave.

Because these machines with double articulation closely approximate water particle motion at the paddle, they are particularly useful when the wave channel is of short length. Furthermore, maintenance is slight because moving parts are out of the water and inertia can be kept reasonably low.

For all machines in the movable-wall category, the length of the generated wave is controlled by the period of oscillation of the movable-wall or paddle while the wave height is controlled by the amplitude of paddle oscillation.

3.2.3 Plunger Types

This second general category comprises those types which operate on the principle of a plunger being periodically thrust into and withdrawn from the water, forming waves by water displacement. The plunger may have various cross-sectional forms, the most common being either a wedge-shape, or one side parabolically contoured with a vertical back. Wave length and height may be controlled by varying the period of oscillation and stroke length (displacement) of the plunger.

Plunger-type machines do not closely approximate water particle motion over the full range from deep to shallow-water waves and therefore relatively long wave channels are required.

Two examples of wave generators in the plunger category are shown in Fig.7.

3.2.4 Pneumatic Types

The third general category is made up of wave generators which produce waves by aspirating the water into a chamber and then letting it fall freely downwards. The chamber is essentially a controlled surge tank. The principle of pneumatic generators is illustrated at the end of Fig. 7. These generators require adequate channel length in front of them to facilitate stabilization of the wave motion. This type is used almost exclusively in the David Taylor Model Basin (U.S. Navy).

In operation, there are limits to the wave periods which can be obtained if the water is allowed to just fall freely in the surge chamber under gravity. To achieve shorter periods, the chamber is connected alternately to the suction and pressure sides of a compressor instead of just the suction side. This procedure also permits higher wave amplitudes to be obtained.

These generators have had a reputation of being difficult to adjust for a particular wave form, but this may no longer be true in view of increased interest and work on this type in the United States. Water flow out the discharge tubes generally gives an initial water particle motion approximating that of shallow-water waves.

The big advantage of pneumatic generators is that they can be designed in configurations (Fig. 8) which do not reflect waves reflected back along the channel by the model.

Various other forms of generators exist which fall outside of the three general categories mentioned. Some of these machines are shown on the last page of Fig. 7. The serpent generator is designed for use in large, rectangular, wave basins rather than in channels. The time at which small sections along its length are horizontally displaced can be adjusted so as to limit the lateral length of a wave crest and thereby, in coastal studies simulate waves advancing to the beach along a stretch of coastline.

The serpent generator is complex and difficult to adjust. The eccentric roller, elliptical cylinder, cylinder with paddles, paddle wheel and surface paddle generators, ignore to varying degrees the kinematic requirements of wave motion thereby necessitating the use of long wave channels. However, the wave generators equipped with paddles do have a very low reflection coefficient.

3.3 Wave Channel Problems Affecting Wave Generator Design

There are other aspects of wave generator design to be considered besides the basic design requirements of kinematics, mechanics, reflection, adjustment, mobility and water leakage already discussed.

A typical wave channel layout is shown in Fig. 9. If the wave generator produces waves in two directions, such as in the case of a rigid flap generator with single articulation, then a length of channel behind the flap must be utilized for a wave absorber. The length required may run in the region of 5' to 15' (Ref. 5 - Correspondence) depending on the size of channel and absorber efficiency.

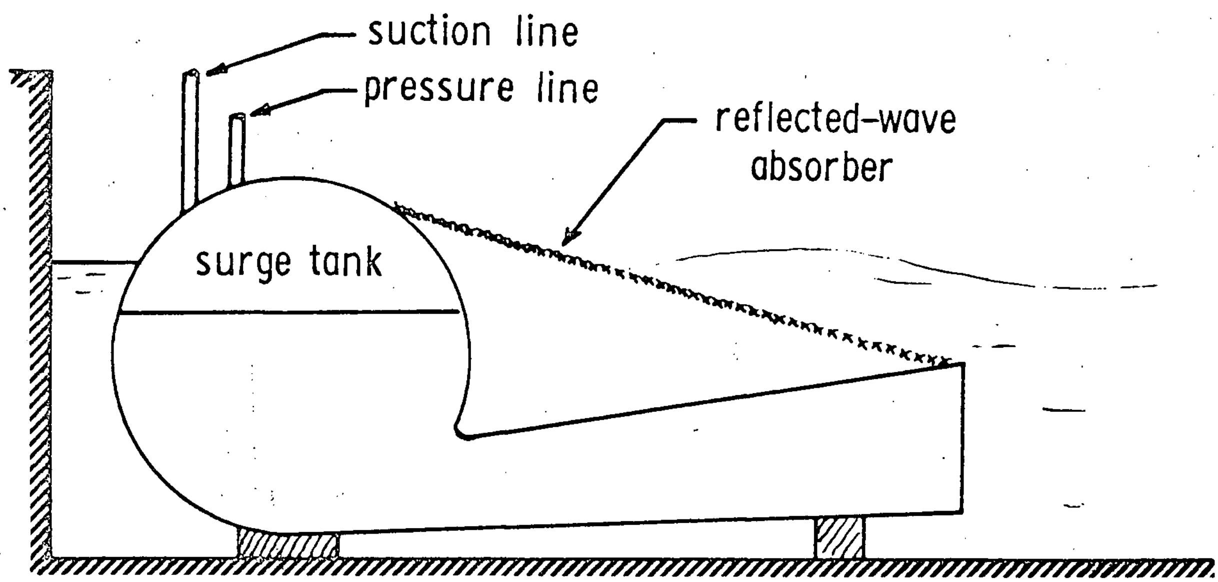


Fig. 8 Schematic of a Pneumatic Wave Generator Designed to Eliminate Wave Reflection.

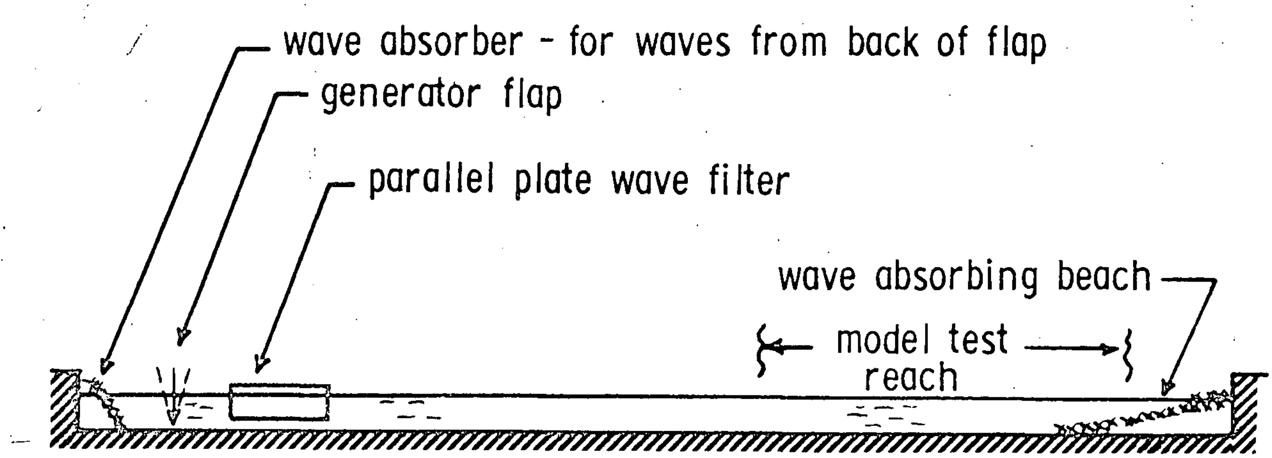


Fig. 9 Schematic of a Typical Wave Channel Layout

When selecting the type of generator to be built, channel length is important. Long channels permit many irregularities in water-particle motion to correct themselves, allowing a choice from a wide variety of generator types. If the channel is of relatively short length, viz. shorter than about 50', then it becomes desirable to produce waves which are initially as perfect as possible. This requires the use of a generator which closely approximates the desired water-particle orbital motion.

If a wave channel is to be designed along with the generator it should be as big as space and finances allow, since the greater the width, length and depth of the channel, the closer it approximates actual prototype conditions.

Generators utilizing flat plates for the wave-generating member may experience difficulties from transverse waves being formed in front of the plate, and causing irregularities in the wave profile. Transverse wave formation, when due to either wave diffraction (Ref.7 - Wave Generator Theory) or local surges induced by water leakage around the sides of the plate, becomes an increasing problem with increased plate width. Ransford (Ref.1 - Wave Generators) recommends the use of unequally spaced "clapotis" plates (Fig.10) to hinder the formation of these waves. The unequal spacing prevents any secondary transverse waves formed in the spaces from reinforcing each other. Other designers also have frequently utilized these transverse oscillation baffles on their wave generator paddles, but there does not appear to be any design information available which would allow the calculation of optimum size and shape, beyond an empirical trial and error procedure. These baffles are particularly useful on very wide wave generating members used in large rectangular wave basins where the use of wave filters is difficult. In wave channels, however, many designers omit the baffle plates altogether and rely entirely on a system of wave filters to smooth out the wave profile.

Transverse waves may create an even bigger problem in a channel when the wave period is such that resonance occurs for wave modes across the channel width. Kravtschenko and Santon (Ref. 6 - Wave Generator Theory) mention that these transverse waves or principle phenomena can produce parasite phenomena, composed of waves with significant harmonics, whose basic period is 2/3 that of the generator paddle.

The net result of transverse waves is an irregular wave profile and water particle motion. To rectify this situation in a wave channel, a filter is used in front of the wave generator. Many types of filters have been tried and their design involves a separate study in itself. However, a series of thin vertical plates parallel to the longitudinal exis of the channel have proven an effective filter for removing initial diffraction ripples and transverse waves (Fig.10).

The required length of most filter systems, for a given attenuation in wave irregularities, increases with increased wave length. Therefore the filter space required in the channel will depend on the maximum wave length to be used.

Other types of wave filters may be required to reduce the height of waves reflected by the model if the wave generator is reflective.

Other transverse wave problems, not easily cured, can result from flexible channel walls. This points to the need for considerable rigidity in the channel structure. Likewise, misalignment of the channel floor and walls can produce an irregular wave form.

Achieving and maintaining large wave heights poses some problems. The use of wave filters to smooth out the waves results in a loss of wave height. In addition, the initial wave height may have been limited by the maximum amplitude of the wave generator plate motion and its kinematics. To illustrate the effect of poor kinematics, which may not be readily apparent, consider a normal deep-water wave. As it steepens it becomes increasingly unstable until it breaks. This occurs when the wave height (H) $\simeq \frac{1}{7}$ L. If the wave generator kinematics do not approximate the wave motion closely, then the instability of the wave form is increased, causing the wave to break sooner, limiting the wave height. This loss of wave amplitude reduces the steepness and may pose a problem in

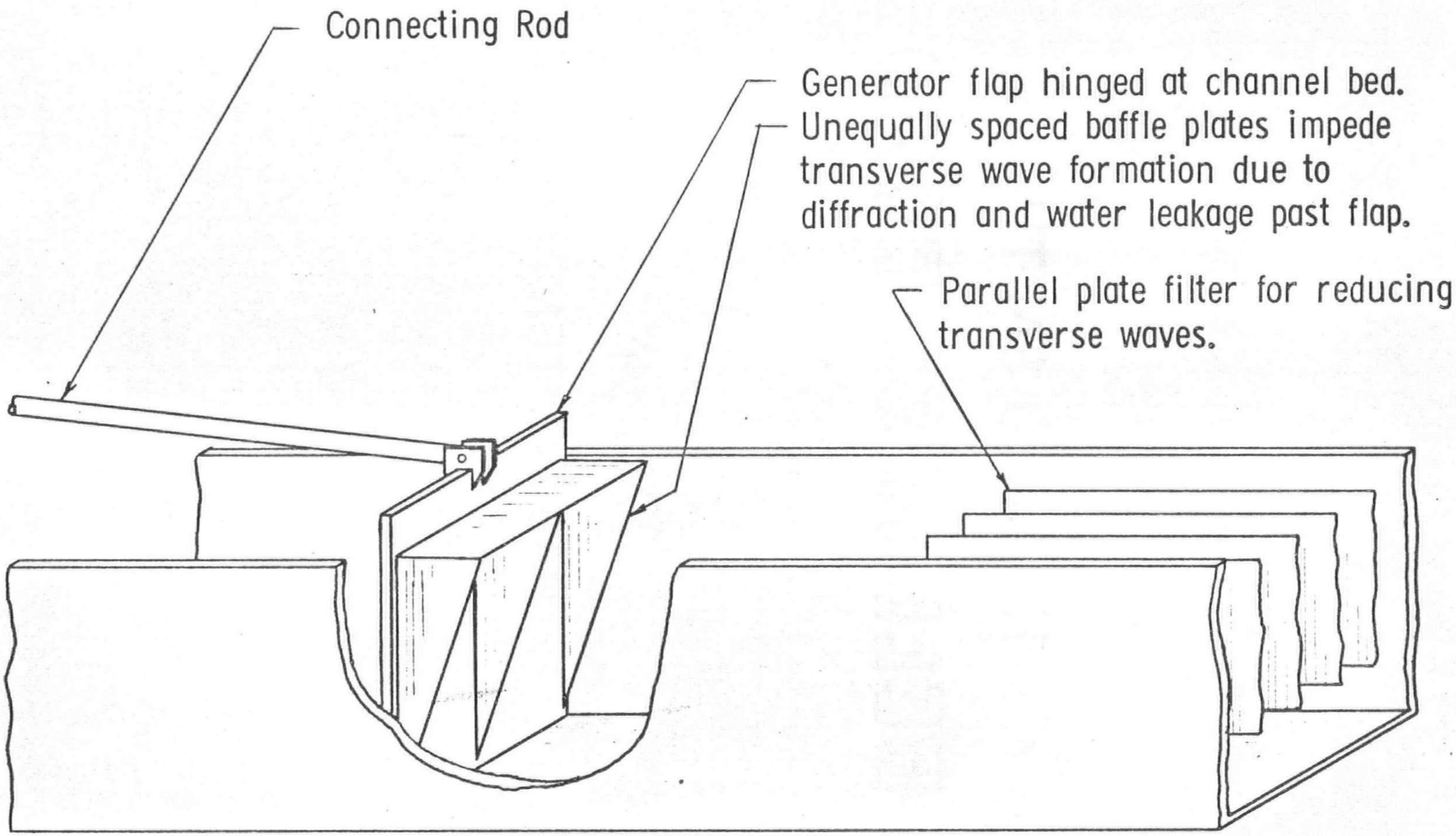


Fig. 10. Methods of Dealing with Transverse Waves in a Channel

in certain tests, usually ones of a theoretical nature.

Adequate wave amplitude may be maintained in two ways. The first method is to have the channel walls converge in plan, downstream of the filters, which will increase the wave amplitude. The second method consists of raising the channel bed by a floor sloping gradually upward away from the wave generator. This has little direct effect on the wave amplitude (the amplitude may even decrease in certain cases) but shortens the wave length (the period remaining the same) which in turn increases the wave steepness. Since this second method leads to appreciable intensification of steepness only when the final ratio of wave length to depth is sufficiently large (shallow water waves), the first method is more generally used. Moreover, the rate of convergence should be low to avoid any wave reflection from the sides. A successful, long-length, channel designed by the National Research Council of Canada, converges from 6' to 4' in width over a length of 40' (Ref.2 - Wave Generator Correspondence).

When wave steepness is of prime importance, consideration must be given to height attenuation due to channel boundary friction. This is especially so if the channel is relatively narrow - under about 2', and sand or pebble bottom layers are to be used. For example, a channel 1.5' wide, 0.75' deep and 60' long with a sand bottom caused a 21% reduction in wave height over its length (Ref.10 - Wave Generators).

At the far end of the channel (Fig. 9), opposite to the generator, a spending beach is required to absorb the energy of incoming waves to avoid reflection. The spending beach comprises some form of wave absorber. Although the design of wave absorbers does not lie within the scope of this thesis, it is worth noting that their efficiency must be high (Ref. 4 - Related Subjects). For an incident wave striking the beach, 99% absorption of its energy still results in a reflected wave going back to the generator with a height 10% of the incident wave height. Fortunately, absorbers can be designed with reflection coefficients (ratio of reflected wave height to incident wave height) in the range of 0.04 to 0.08.

When tests involve sand beaches, the length of channel required for the beach alone may be considerable. Many beach sands are unstable under wave action on slopes steeper than 1 on 10, and equilibrium may require a mean slope of 1 on 20, or flatter. For a 1 on 20 slope, the channel length required by the beach will have to exceed 20 feet for a 1 foot water depth if space is to be left for wave runup on the beach. From the preceding discussion it can be seen that, when designing a wave generator for a relatively short channel, the length of channel assigned to the generator, the wave absorber behind the generator, and the wave filter, must be held to a minimum in order to maximize the length of the channel for wave stabilization and tests, between the wave filter and the beach.

In spite of the great amount of research that has been done, it seems that no particular type of generator has yet been generally accepted as offering the optimum solution to these wave channel problems.

3.4 Specific Problems of Wave Generator Design

3.4.1 Minimum Wave Period

An important decision when designing a wave generator is the range of wave periods it must produce. The shorter periods will apply to deep-water waves while the longer periods will apply to shallow-water waves. The wave length associated with a particular wave period can be calculated using equations (2) and (5) as follows:

wave celerity $C = \sqrt{\frac{gL}{2\pi} \cdot \tanh \frac{2\pi d}{L}}$

and $L = CT$

from which $L = \sqrt{\frac{g}{2\pi} T^2 \tanh \frac{2\pi d}{L}}$ (11)

For deep-water waves the term $\tanh \frac{2\pi d}{L}$ equals 1, and expressing units in feet and seconds, equation (11) reduces to

$$L = 5.12T^2 \text{ ft.} \qquad (12)$$

Using a theoretical approach the minimum wave period is governed by surface tension. Waves with lengths under 2" are controlled by surface tension and should definitely be avoided. To reduce surface tension effects to a negligible level (Fig.5) it is desirable to avoid waves with lengths under about 1 foot. In deep water a 2 inch long wave has a period of about 0.18 sec. while the period of a 1 foot wave is 0.44 sec.

From the practical viewpoint Galvin (Ref. 8 - Correspondence) recommends a minimum wave period of about 0.5 sec. and preferably 0.75 sec. These periods yield corresponding wave lengths in deep water of 15.4 in. and 34.5 in.

The use of Galvin's recommendations, instead of the theoretical minimum wave period, is supported by two practical considerations. First, for wave lengths of 2, 12, 15.4 and 34.5 inches the theoretical, maximum, deep-water wave heights are 0.29, 1.7, 2.2 and 4.9 inches respectively (eqn.9). Due to wave generator imperfections these maximum heights will not be closely approached without using a converging channel. In model work, measurement errors assume increasing significance with decreasing wave size. Therefore, due to their limited height, the generation of very short waves with periods less than 0.5 sec. is not normally justified.

The second point supporting Galvin's recommendations concerns the high oscillation rate at which wave generator paddles must operate to produce these short waves. Periods of 0.18, 0.44, 0.50 and 0.75 sec. require oscillation rates (crank speeds) of

313, 136, 120 and 80 R.P.M. respectively. As shown later in this thesis, mechanical and water inertia loads rise rapidly beyond roughly 100 R.P.M. Hence short wave lengths require wave generators having considerably increased structural strength and motive power, the cost of which is not normally justified by usage due to the very limited wave height and extreme departure in wave size from prototype conditions.

3.4.2 Maximum Wave Period

There are a number of opinions as to the maximum wave period which should be designed into a generator for a laboratory flume of given length. Some researchers suggest that distances of 3 to 5 wave lengths should be allowed between generator and model, to insure stable shape and uniformity of the waves, but have not stated the type of generators used. One commercial hydraulic laboratory visited, which uses a rigid flap generator with single articulation at the channel bed to generate deep-water, transition and shallow-water waves, likes to have at least 10 to 11 wave lengths between generator and model. Galvin (Ref. 8 - Correspondence) suggests a maximum wave length of no more than half the distance between generator and still water line on the beach.

Theory developed by Havelock (1929) and utilized by Biésel (1951) Ref 1 and 2 - Wave Generator Theory) states that if a plate oscillates sinusoidally, then a length along the channel equal to three times the water depth should suffice to provide natural compensation for the defective wave form. This theory would imply no limit to the maximum design wave length. Unfortunately, filters required to smooth out the wave form, become increasingly long with increased wave length, for the same effectiveness. Also, this theory is a first order approximation and is still unproven. Therefore, designing to Calvin's limits would appear to be a reasonable compromise.

The period T for the longest wave length can be computed using the shallow-water form of equation (11) which gives

$$T = \frac{L}{\sqrt{gd}} \quad . \qquad (13)$$

The wave length L will be fixed by the distance from the wave generator to the beach still-water line, assuming Galvin's method. From equation (13) it is seen that lowering the water depth (d) will increase the period, necessitating the choice of a minimum design value for depth. It appears that experiments are rarely done in water depths of less than 6", consequently this value seems reasonable in calculating the maximum period. Galvin feels that, in general, a maximum period of at least 3 secs. is desirable in a wave channel, if sufficient wave variety is to be achieved.

3.4.3 Design Water Depth

Maximum design depth is an important factor in determining the size of waves generated - deep water waves excepted, since their limiting height is dependent only on wave length. For a shallow-water wave of maximum design height (H), the required theoretical depth (d) can be calculated from equation (10) which rearranged gives $d = 1.33\,H$. However, from practical observations Galvin suggests using a design water depth of 1.5 to 2 times the maximum desired wave height, because the waves tend to become unstable and break when the depth to height ratio is less. Used in the converse sense this rule of thumb is handy when estimating shallow-water wave heights to be expected from a wave generator.

When deciding the design water depth and maximum wave height some channel freeboard must be allowed to control splash.

3.4.4 Inertia

Generators of the movable-wall and plunger types produce waves using members which oscillate. Because of the effects of inertia on the assigned motion the weights of the moving parts must be minimized, commensurate with adequate rigidity, necessitating an adequate knowledge of the forces involved when designing the moving parts. Snyder, Wiegel and Bermel (Ref. 9 - Wave Generators) give a method for estimating the forces acting on movable-wall generators from wave energy utilizing experimental results of the U.S. Beach Erosion Board (1949). Force calculations can also bes made using the theories of Biésel (Ref. 1 - Wave Generator Theory). Forces for plunger-type generators, using plunger bodies of various cross sections, can be deduced from the theories of Schuler (Ref 9 - Wave Generator Theory), a resumé of which is available in

"Laboratory Surface Wave Equipment" (Ref. 10 - Wave Generator Theory).

3.4.5 Wave Generator Motion

Departures from the motion assigned to a wave generator create wave problems which can be divided into two groups - first, irregularities in the wave form and second, changes in the wave length.

The first problem, irregularities in the wave form, is due to momentary departures from the assigned motion, caused by load fluctuations. The main load on the wave generator paddle breaks down into wave forces and inertia forces (water and mechanical), both of which fluctuate and are 90° out of phase. Electric motors are the usual source of power in laboratory wave generation and are subject to small speed changes resulting from these load changes. Another source of fluctuating load is friction, due to seals sometimes employed for reducing leakages between the plates of movable-wall type generators and the channel bed and side walls. These seals rub against walls that may be rough or slightly out of alignment, so that the friction forces can be both appreciable and irregular. Most designers omit seals and are content to reduce clearances to a value sufficient to avoid large friction forces.

These fluctuations of load affect the choice of power source and associated speed-changing mechanisms. Hydraulic drives and fluid couplings have been tried but some of these systems were found unsatisfactory due to velocity changes as the load came on and off. Backlash must be considered when utilizing gear-type speed reducers.

When using an electric motor drive the output R.P.M. will drop a certain amount as the load goes from no load to full load. By using a more powerful motor than required, the maximum wave-generator load becomes a smaller percentage of motor full-load capability, with a corresponding reduction in range of speed fluctuation. For this reason many wave generators are overpowered in spite of the increased capital costs and slight loss of motor efficiency.

The second problem, changes in wave length, results from variations in wave generator oscillation period. This problem is caused by long-period fluctuations in line voltage which affect motor operating R.P.M. Some laboratories have used electric motor-generator sets to overcome this problem. The Dauphin Hydraulic Laboratory (Neyrpic), Grenoble, France (Ref. 1 - Wave Generator Theory) relies on the relatively constant power-line frequencies, certainly more constant than voltage, by using synchronous motors of excessive power. In this way, motor slippage is reduced and the motor revolves at nearly its nominal speed. The seriousness of this problem depends on the rate of voltage fluctuation, the magnitude of the fluctuation and the type of test being done. Many wave generators are powered by electric motors connected directly to the normal mains without any special voltage regulating devices.

When considering the problem of irregular motion, the use of a flywheel would appear to be a solution and a number of wave generators have been so designed. Unfortunately, the users have found that flywheels are slow to accelerate and decelerate which results in the channel being filled with a confusion of waves of varying lengths whenever the generator is started or stopped, or whenever a speed change is made. Not only must the model be protected from this confused sea, whose effects are much greater than the wave effects at uniform regime, but also considerable time is lost waiting for stability to be achieved.

Theoretical studies show, that as a first approximation, wave-generator motion should be sinusoidal (simple harmonic motion). Experiments indicate that it would be preferable to assign a more complex pattern to the motion, but as yet this pattern is not know exactly. The generally used sinusoidal motion is usually obtained by actuating the wave generator paddle with a long connecting rod connected to a rotating crank.

3.4.6 Wave Period Control

Wave period is the same as the period of oscillation of the wave generator which in turn is determined by the output R.P.M. of the drive system. From the explanations given in sections 3.4.1 and 3.4.2 on how to determine minimum and maximum wave periods, it is obvious that control of the wave period also gives control of the wave length.

There are a number of ways by which the output R.P.M. of the drive system can be varied. First is the use of a manually shifted gearbox, which is not recommended due to the rather limited speed selection. Next is the use of a variable drive pulley system. These are of two types, belt and chain. The belt-types employing positively controlled sheaves to give speed variations may prove satisfactory, but any units employing spring controlled sheaves, for reasons explained later in Section 5.2 of this thesis, should be avoided. Chain types using mechanically controlled sheaves give positive speed control and are popular.

Direct-current motors give control over a wide speed range and are used in conjunction with a gear box. Immediate speed changes are obtained at the turn of a rheostat. Synchronous motors require reduction gears and a speed-control device which permits adjustment of the frequency to any desired value between two definite limits.

Hydraulic devices and other means of variable speed control are commercially available but when selecting one it must be positive in its operation and reasonably free of backlast in order to handle the fluctuating load of a wave generator, without contributing irregularities to the motion.

It is desirable to be able to effect a change in period quickly while the wave generator is operating. For example, changing the wave generator oscillation rate directly from 60 to 80 R.P.M., without stopping the generator and restarting it, reduces the intensity and duration of the confused sea produced by the change. Also it permits fine period adjustments to obtain a desired wave length.

3.4.7 Wave Height Control

Wave height is controlled by the amplitude of oscillation of the wave generator. The oscillation amplitude of machines driven by a rotating crank and connecting rod can be easily controlled by adjustment of the crank throw (Fig. 11). Some wave generators are designed with additional links between the drive system and the flap and the flap amplitude is controlled by changing the length of the link (Fig. 13). These two examples of amplitude control can be designed for either manual adjustment with the generator stopped, or automatic adjustment when either running or stopped. Schematics for the design of automatic adjustment controls are shown in Fig. 12 and Fig. 14. A study of Fig. 11 and Fig. 13 show that in the case of manual control, the adjustable crank is best, the use of an adjustable link making the drive system unnecessarily complex. However, in the case of automatic controls, Fig. 12 and Fig. 14, the use of a link adjustment can be justified by the simpler automatic mechanism involved.

With reference to Fig. 12, the automatic control of crank throw adjustment functions as follows. The fixed differential causes outer shaft #2 to rotate in the opposite direction to the outer main drive shaft #1. The second differential causes the inner shaft to rotate in the same direction and at the same speed as the outer main shaft #1. By using a reversible control motor to rotate the casing of the moveable differential the inner shaft can be caused to rotate either faster or slower than the outer drive shaft #1. This speed difference causes the threaded guide rod to be rotated in one direction when the inner shaft is rotating faster than the main outer shaft #1, and in the other direction when slower, producing a shift in crank pin position. Either limit switches or a slipping clutch must be incorporated in the system to prevent damage by over adjustment.

The act of changing the wave-generator oscillation amplitude while the wave generator is operating, can produce irregularities in the wave form detrimental to the model; requiring that the model be protected. Since the rate at which these automatic amplitude changers work is relatively slow, especially compared to the time required for a change in motor speed, it is usual to stop the wave generator for amplitude changes. Once the wave generator is stopped, the automatic system offers only a convenience over the manual adjustment. Galvin mentions that in his experience the automatic amplitude control is rarely used with the wave generator running.

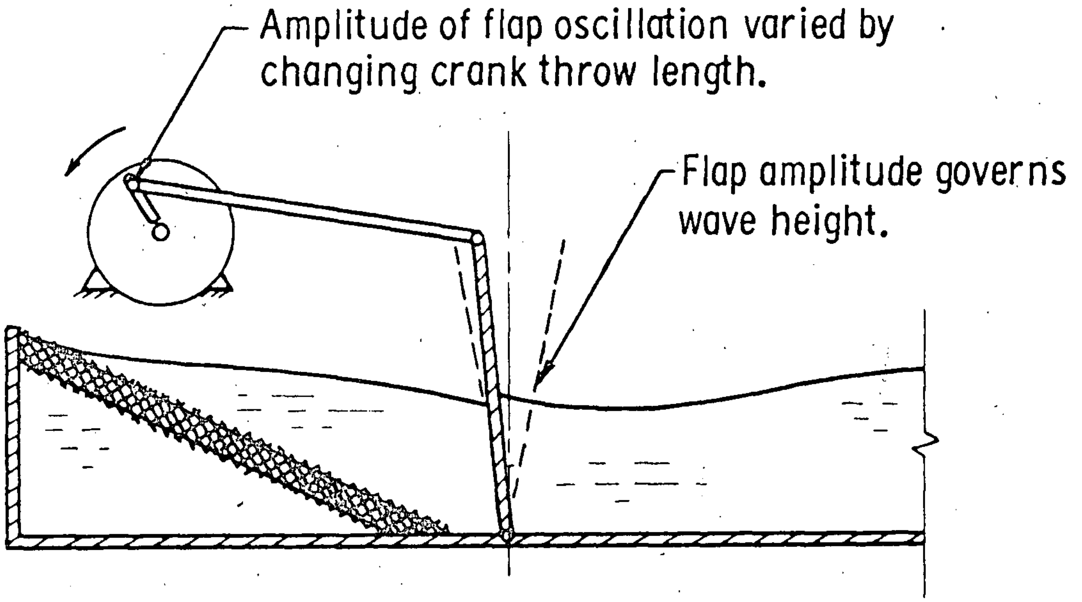


Fig. 11 Principle of Wave Height Control by Crank Throw Adjustment

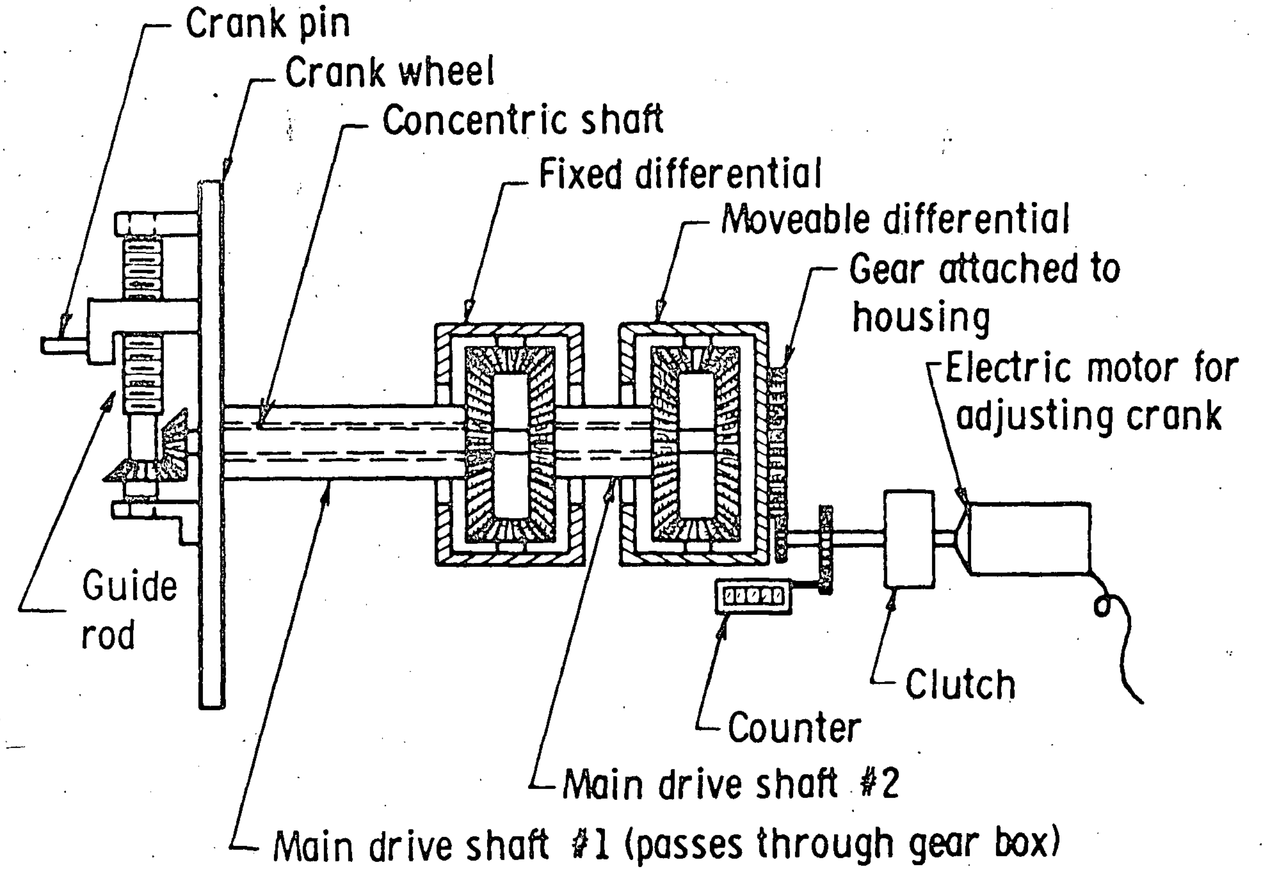


Fig. 12 Mechanism for Automatic Crank Throw Adjustment.

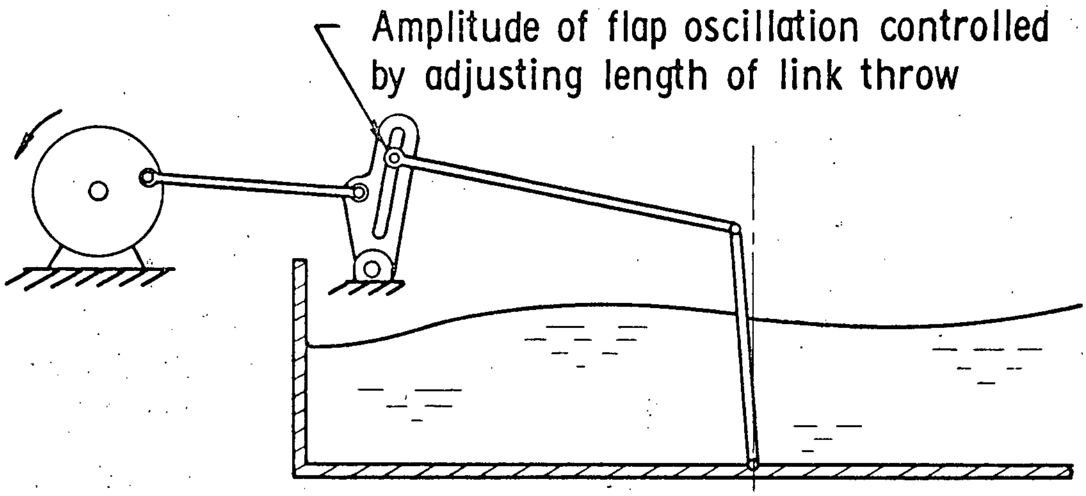


Fig. 13 Principal of Wave Height Control by Link Throw Adjustment.

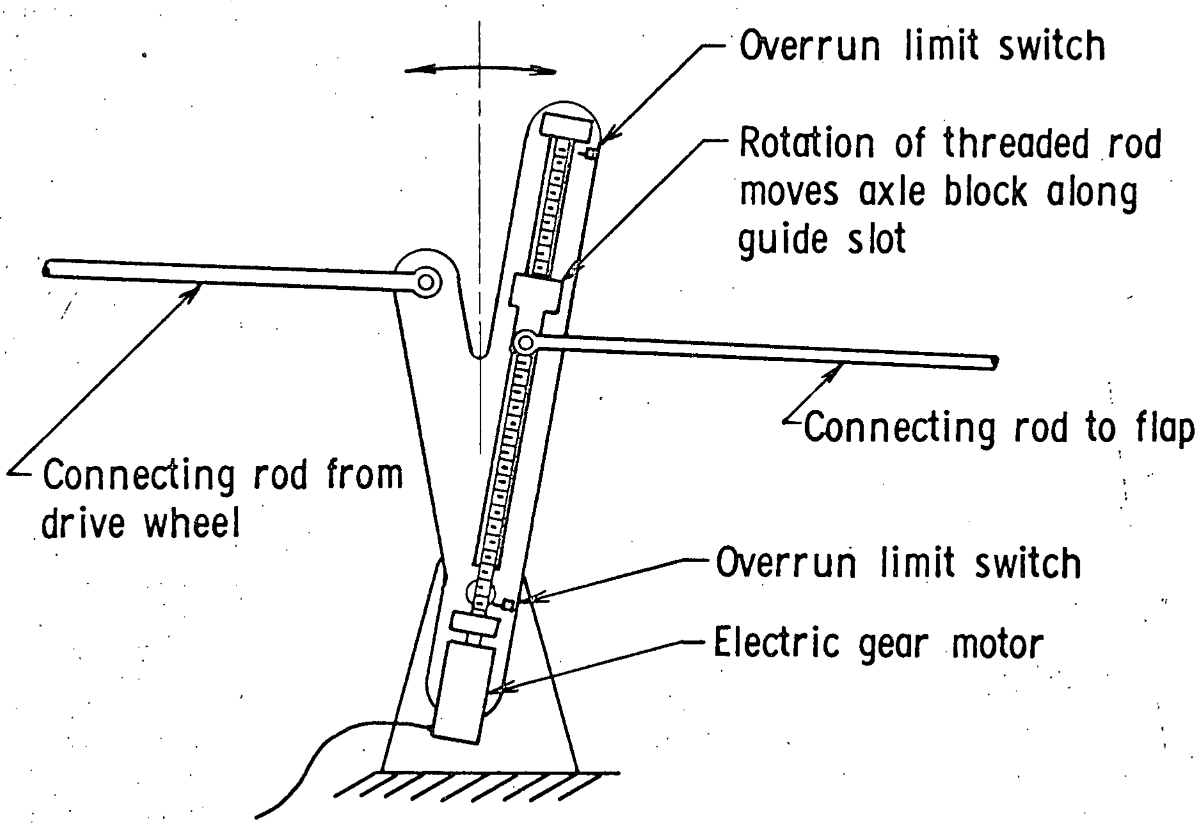


Fig. 14 Mechanism for Automatic Adjustment of Link Throw.

In summary, automatic amplitude controls complicate the wave-generator mechanism and are expensive, particularly in the case of the adjustable crank type. A high degree of accuracy is necessary in their construction since some parts must be able to move smoothly and yet not have enough free play to produce irregularities in the flap motion. Therefore, the provision of automatic amplitude controls over manual adjustment can not normally be justified unless the wave generator is to be used to specifically generate irregular waves for certain studies involving surf, seiches and ship rolling.

3.4.8 Anticipated Performance

Experience indicates that any wave generator can be expected to produce finer waves of small steepness than of large steepness and better short waves than long waves for the same steepness. In fact, generated waves of large steepness are definitely not of uniform quality, even if observed after they have traversed a great length. Biesel (Ref.1 - Wave Generator Theory), offers a brief plausible interpretation of these findings which would suggest that the best wave generator is always the piston-type machine. It is interesting to note that Galvin (Ref. 8 - Correspondence) states that, the Coastal Engineering Research Centre, Washington, D.C., has found the piston-type generator to be the most satisfactory. (Their channels are long and the other types of generators used were not mentioned). In any case, there is no theoretical analysis yet developed which gives a rigorous explanation for these performance results.

3.5 Choice of Wave Generator for a Short Channel

The key problem when generating waves in a channel of short length is the lack of spare channel length to enable irregularities in water particle motion and wave form to correct themselves. It now becomes important to have water particle motion for a particular wave form established right at the wave generator paddle. This kinematic requirement appears to be theoretically best satisfied by a wave generator of the rigid paddle, double articulation type.

Examination of the schematics of rigid paddle generators with double articulation shown in Fig. 7, suggested consideration of the pendulum type due to its simplicity. This type was developed by Ransford (Ref. 1 - Wave Generators) and is shown in Fig. 15. A modification of the geometry of Ransford's generator was made by Coyer (Ref. 3 - Wave Generators) and the arrangement is shown in Fig. 16.

Inertial effects due to water and wave generator mass are a maximûm at the extremes of paddle oscillation. In Ransford's design, the weight of the paddle parts works against inertial effects and helps the motion, whereas in Coyer's arrangement, the weight adds to the inertial effects. On the other hand, Coyer's arrangement offers advantages of compactness and slightly less water leakage under the paddle (due to the geometry of the paddle motion). Experimental data presented by Coyer shows well-established deep and shallow-water wave profiles and correct particle motions existing at a test point only 12' from the generator paddle when the motion of

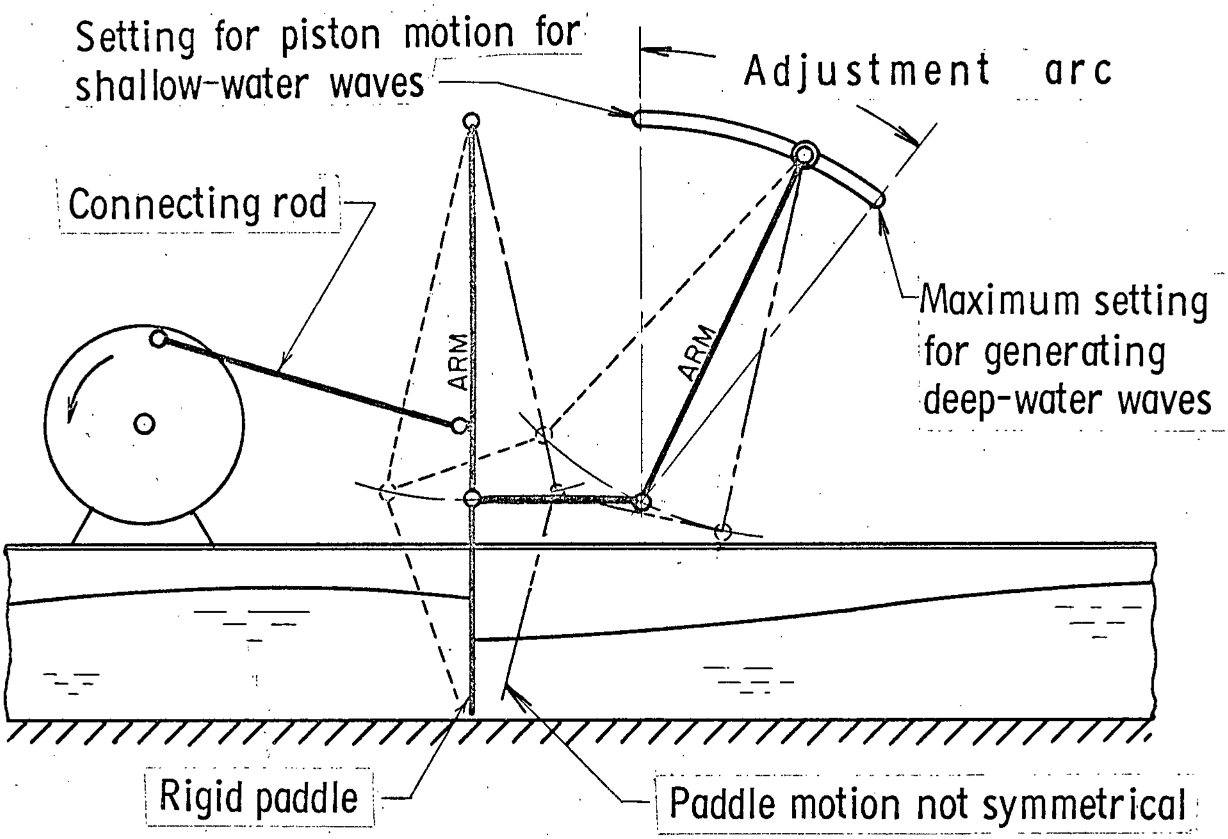


Fig. 15 . Principle of Ransford's Pendulum-Type Wave Generator.

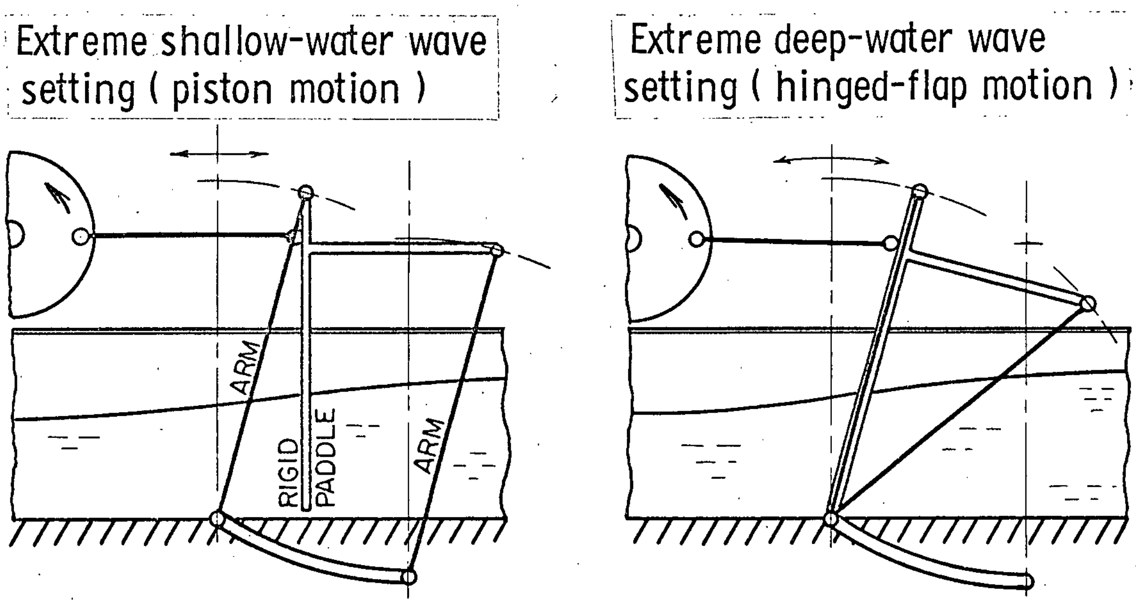


Fig. 16 Coyer's Modified Version of Ransford's Pendulum-Type Wave Generator, Illustrating the OPERATING PRINCIPLE.

the paddle is adjusted to suit the water particle motion of the generated wave.

The variety of wave generator designs and the diversity of opinions, indicate that wave generator design has not yet been optimized. The literature leads one to conclude that most existing wave generators are in the movable-wall category. It is interesting to note that from the practical aspect of actual performance, the opinions of numerous researchers appear to be favouring piston-type wave generators for all types of waves. These researchers were using long channels but it brings up the point of just where is the dividing line between "long" and "short" channels? Since the double articulation arrangements permit the wave generator paddle to function as a piston, use of such a wave generator would permit a future research opportunity, viz., to compare the performance of a piston-type wave generator, producing all wave types in a "short" length channel, with results from a supposedly kinematically superior double articulation type utilizing the same channel, drive mechanism and recording equipment.

For reasons stated, it was decided to design for the large flume aawave generator of the rigid paddle, double articulation type using the geometrical arrangement of Coyer (Fig. 16).

4. WAVE GENERATOR THEORY - a means of determining design forces.

4.1 Theoretical Analysis of a Wave Generator

When designing a wave generator it is necessary to know the water forces involved in its operation, in order to calculate power requirements and to give the mechanism adequate strength and rigidity without unnecessarily increasing the inertia of the oscillating members. Also, some means is required for determining the amplitude of paddle motion required to produce a wave of a certain height. This information can be calculated for rigid paddle wave generators with double articulation, using the theory of Biésel(Ref. 1 - Wave Generator Theory), which is the most "usuable" of available mathematical theories. Biésel's theory (1951) is based on earlier work by Havelock (1929) (Ref. 2 - Wave Generator Theory).

The wave generators considered by Biésel are esentially such that their action is equivalent to that of a membrane or a flexible blade completely obstructing the channel and oscillating with a sinusoidal (simple harmonic) motion about a mean vertical position.

The theory is based upon the assumptions that: the water motion is irrotational; i.e. the water is frictionless and incompressible (ideal fluid); the equations of motion satisfy all the hydrodynamic laws to the first order of approximation and; motion is two-dimensional occuring parallel to the side walls of the channel. The formulae derived are rigorously valid only for infinitely small wave heights and in cases where viscosity and turbulence may be neglected. Water leakage around the flap (paddle), which occurs in an actual machine, is not considered.

Summarizing this theory briefly (Ref. 10 - Wave Generator Theory) boundary conditions are:

(a) $\frac{\partial \phi}{\partial y} = 0$ for $y = 0, \quad x \geqq 0$

where the origin of the coordinates is placed on the channel bed with the ox-axis extending along the bed in the direction of wave propagation (Fig. 17) and the oy-axis designating the mean position of the oscillating member;

b) $\frac{\partial^2 \phi}{\partial t^2} + g \frac{\partial \phi}{\partial y} = 0$

for the surface condition in a first-order theory; and

c) $\frac{\partial \phi}{\partial x} = k \, \S(y) \cos kt$

for the boundary condition at the generator flap where

$k = \frac{2\pi}{t}$ and the motion of the generator is $x = \S(y) \sin kt$.

Biesel's solution for this problem is velocity potential

$$\phi = -\frac{k}{m} c \cosh my \sin (kt-mx) - \Sigma_{n=1}^{\infty} c_n \frac{k}{m_n} \cos m_n y \, . \, e^{-m_n x} . \cos kt \quad (14)$$

$$\text{where} \quad c = c_o = 2m_o \frac{\int_o^d \S(\alpha) \cosh m_o \alpha \, (d\alpha)}{\sinh m_o d . \cosh m_o d + m_o d} \quad (15)$$

$$\text{and} \quad c_n \doteq 2m_n \frac{\int_0^d \S(\alpha)\cos m_n\alpha\,(d\alpha)}{\sin m_n d.\ \cos m_n d + m_n d} \qquad (16)$$

In these expressions, m_o is the positive solution of the equation

$$k^2 = \alpha g.\ \tanh \alpha d \qquad (17)$$

and m_n represents the positive solutions of the equation

$$k^2 = -\alpha g.\ \tan \alpha d \qquad (18)$$

Using the velocity potential ϕ, the displacements of the particles whose mean position is x and y are found to be

$$X = c\cosh my.\ \sin(kt-mx) + \sum_{n=1}^{\infty} c_n.\cos m_n y.e^{-m_n x}.\sin kt \qquad (19)$$

$$\text{and } Y = c\sinh my.\cos(kt-mx) + \sum_{n=1}^{\infty} c_n.\sin m_n y.e^{-m_n x}.\sin kt \qquad (20)$$

The magnitude of the semi-height (a) of the generated wave can be deduced as being

$$a = c_o \sinh md. \qquad (21)$$

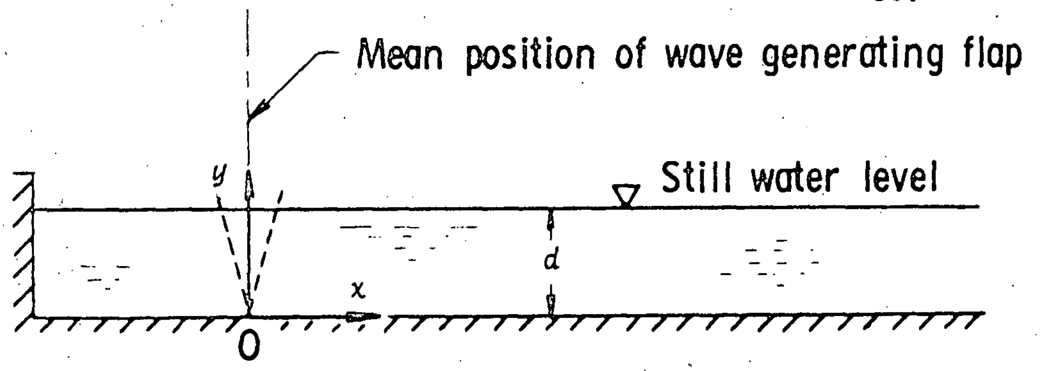
Neglecting terms of the second order it can be proven that

$$\frac{P}{\rho} = -\frac{\partial \phi}{\partial t} + g(d-y).$$

Substituting for ϕ its value from equation (14) yields the general equation for pressure in front of the wave generator flap which is

$$P = \rho g\,(d-y) + \rho g\, a\,\frac{\cosh my}{\cosh md}\ .\ \cos(kt-mx)$$

$$+ \rho \sum_{n=1}^{\infty} g c_n \tan m_n d.\ \cos m_n y.e^{-m_n x}.\ \sin kt \qquad (22)$$



$a = \frac{1}{2}$ *height of the generated wave.*

$m = m_o = \frac{2\pi}{L}$

L = *wave length.*

d = *mean water depth measured from bottom.*

§(y) = *denotes the maximum displacement of the various points of the wave generator flap as a function of their depth y.*

x, y = *coordinates.*

P = *total pressure.*

P_h = *hydrostatic pressure.*

P_n = *wave pressure.*

P_i = *inertia pressure of water.*

ρ = *water density.*

g = *acceleration of gravity.*

$k = \frac{2\pi}{T}$

T = *period of flap ascillation.*

t = *time (secs.)*

n = *1, 2, 3 . . .*

e = *maximum displacement of generator flap from mean position.*

X, Y = *horizontal and vertical particle displacements from mean position.*

β = *a number.*

η = *amplitude of initial water disturbance.*

F_n = *wave force per unit width for water on both sides of flap.*

F_i = *water inertia force per unit width for water on both sides of flap.*

Fig. 17 Definition of Coordinate System and Terms used in Biésel's Theory.

Setting x=0 in formula (22) yields the expression for the pressure acting on the generator flap. This pressure breaks down into three terms.

a) The first term is hydrostatic pressure

$$p_h = \rho\, g(d-y) \tag{23}$$

b) The second term is wave pressure

$$p_n = \rho\, ga \frac{\cosh my}{\cosh md} \cos kt \tag{24}$$

and represents pressure required to form the actual wave. This pressure is in phase with the oscillation speed of the wave generator flap. Energy expended to overcome this pressure is recovered in the wave.

c) The third term is inertia pressure

$$p_i = \rho g \sum_{n=1}^{\infty} c_n \tan m_n d.\ \cos m_n y.\ \sin kt \tag{25}$$

It is the pressure required to overcome water inertia and is in phase quadrature with the wave generator flap speed. This component acts as an augmentation to the inertia of the moving parts of the generator.

When a generator flap emits waves in both directions the hydrostatic pressures existing on either side of the flap balance each other out while the wave pressures and inertia pressures on each side are respectively additive doubling their individual effect.

When using Biesels equations values of the coefficient m_n, or preferably of the dimensionless product $m_n d$, are most easily determined graphically using Fig. 18.

Analysis of equations (19) and (20), which give water particle displacements X and Y, shows that the motion imparted by the wave generator to the water is composed of an ordinary wave (such as would be produced by an ideal wave generator) and an initial disturbance, or transitory in space, which decreases exponentially as $e^{-m_n x}$ with increasing distance x from the generator flap. Biesel shows that, at distance x=d from the generator flap mean position, the maximum amplitude of this disturbance is 21 per cent of the initial amplitude, while at x=2d the amplitude is only 4.3 per cent and at x=3d it is reduced to 1.0 per cent. Biesel and Havelock both concluded that if the motion of a wave generator flap (paddle) reasonably approximates the wave form, water particle motion, then a length equal to three times the depth of the channel will suffice to provide natural compensation for defects in the wave form.

The vertical amplitude of the transient oscillation at the generator flap (x=0) from equation (20) is

$$\eta_n = c_n \sin m_n d \qquad (26)$$

With regard to the height of the generated wave, Biesel suggests an efficiency coefficient of about 70 per cent for relatively large laboratory wave generators. Efficiency will in fact vary widely depending on the ratio of wave length to water depth, the amount of water leakage past the wave generator flap and flap proportions.

4.2 Biésels Theory Applied to a Rigid Paddle, Double Articulation Wave Generator

4.2.1 General

A rigid paddle wave generator with double articulation can be adjusted to make the paddle operate with a piston motion, with a hinged-flap motion, or with an intermediate motion (Fig. 20). The application of Biésels formulae for force calculations will now be examined for each one of these operating modes.

4.2.2 Piston Motion

For piston motion the flap displacement function $\S(y)$ has the very simple form $\S(y) = e$ where e is a constant (Fig. 20a).

From equation (21) the semi-height of the generated wave is

$$a = c_o \sinh md = 2m \frac{\int_o^d e \cosh my \, (dy)}{\sinh md. \cosh md + md} \sinh md$$

$$= \frac{2 \sinh^2 md}{\sinh md. \cosh md + md} e$$

$$= Ke \qquad (27)$$

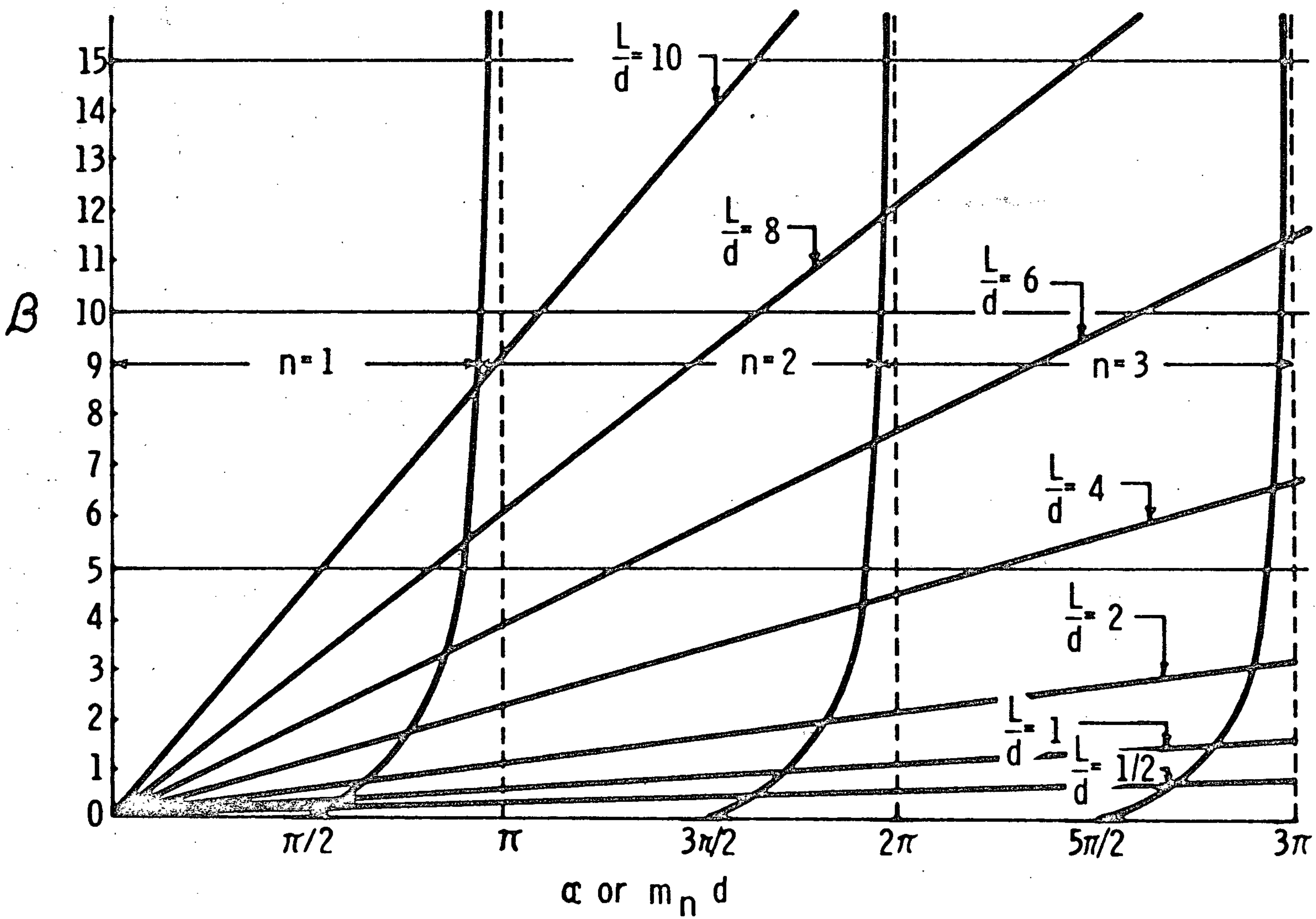


Fig. 18 Graphical Determination of the Values of $m_n d$ for Various Values of $\frac{L}{d}$ (Biésel).

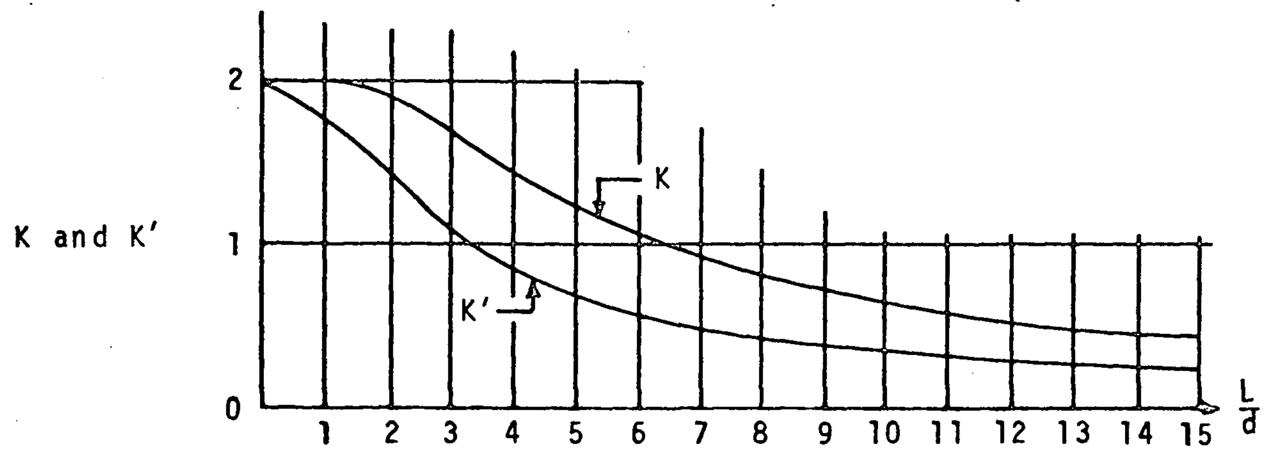


Fig. 19. Variation of Coefficient K (Piston Motion) and K' (Hinged-Flap Motion) for Various Values of $\frac{L}{d}$ (Biésel).

where $K = \dfrac{2 \sinh^2 md}{\sinh md . \cosh md + md}$.

Figure 19 shows the variations of coefficient K as a function of the ratio $\frac{L}{d}$.

Water pressures acting on one side of the flap can be computed. The hydrostatic pressure P_H, given by equation (23) remains as $P_H = \rho g\ (d-y)$. (28)

The wave pressure from equation (24) becomes

$$P_n = \rho g K e . \frac{\cosh my .}{\cosh md} \cos kt \qquad (29)$$

The water inertia pressure from equation (25) is

$$P_i = \rho g \sum_{n=1}^{\infty} C_n . \tan m_n d . \cos m_n y . \sin kt. \qquad (30)$$

where from equation (16) $\quad C_n = \dfrac{2\ e \sin m_n d}{\sin m_n d . \cos m_n d + m_n d}$. (31)

When calculating inertia pressure it is sufficient to compute only the first three terms of the series.

The resultant forces per unit width on the flap, with water on both sides, can be determined from the pressure equations. For water on both sides of the flap the hydrostatic pressures balance out and can be neglected. The wave force for water on both sides of the flap is

$$F_n = 2 \int_0^d P_n\ (dy) = 2\ \left(\frac{Ke\rho g}{m} . \tanh md\right) \cos kt \qquad (32)$$

and will be a maximum when $\cos kt = 1$, that is, when $kt = 0$.

The water inertia force for water on both sides of the flap is from equation (30)

$$F_i = 2\int_o^d p_i\ (dy)$$

$$= 2\rho g\ \{(\frac{c_1}{m_1}.\ \tan m_1 d.\ \sin m_1 d) + (\frac{c_2}{m_2}.\ \tan m_2 d.\sin m_2 d)$$

$$+ (\frac{c_3}{m_3}.\ \tan m_3 d.\ \sin m_3 d)\}\ \sin kt \qquad (33)$$

where values for c_1, c_2 and c_3 can be obtained from equation (31). The inertia force will be a maximum when sin kt = 1, that is, when $kt = \frac{\pi}{2}$.

When using the force equations, values such as $\frac{1}{m_1}$ can be put into the form $\frac{d}{m_1 d}$ and values for $m_1 d$ read from figure 18.

4.2.3 Hinged-Flap Motion

For hinged flap motion (Fig. 20b) the flap displacement function § (y) has the form $§\ (y) = \frac{e}{d}\ y.$ (34)

From equation (21) the semi-height (amplitude) of the generated wave is

$$a = C_o \sinh md = 2\ m\ \frac{\int_o^d \frac{e}{d}.\ y.\ \cosh my\ (dy)}{\sinh md.\ \cosh md + md}.\sinh md$$

$$= 2\ e\ \frac{1-\cosh md + md.\ \sinh md}{md\ (\sinh md.\ \cosh md + md)}.\ \sinh md$$

$$= K\ e \qquad (35)$$

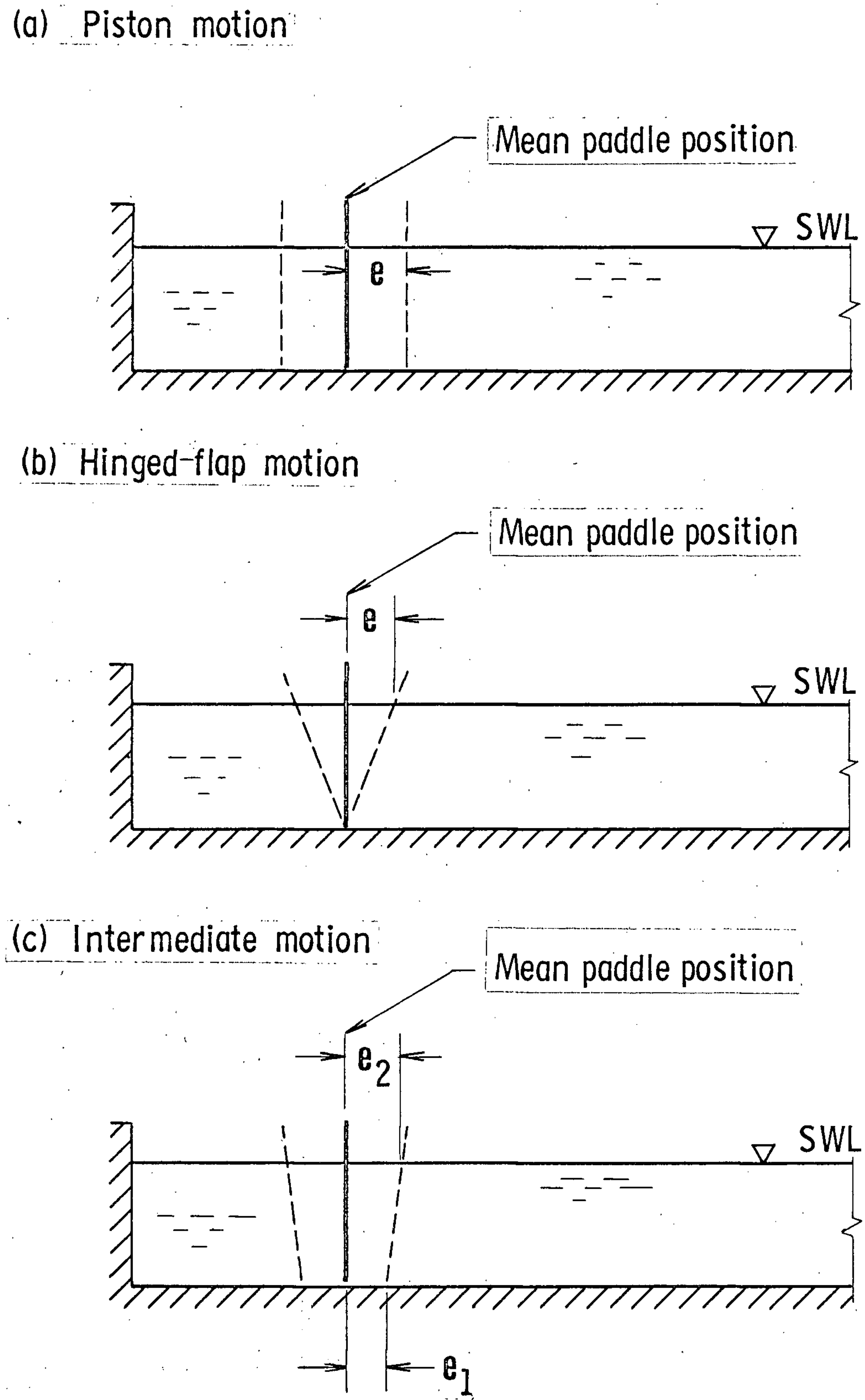


Fig. 20 - Paddle Displacements for the Three Operating Modes of a Rigid Paddle Wave Generator with Double Articulation.

where $K' = 2 \dfrac{\sinh md\ (1 - \cosh md + md.\ \sinh md)}{md\ (\sinh md.\ \cosh md + md)}$.

Values of K' for various values of $\frac{L}{d}$ are given in Fig. 19.

The value of C_n is given in equation (16). Substituting for § (y) and integrating by parts gives

$$C_n = 2_e \frac{m_n d.\ \sin m_n d + \cos m_n d - 1}{m_n d\ (\sin m_n d.\ \cos m_n d + m_n d)} \tag{36}$$

Force equations can now be determined if required using the same procedure as for piston motion (Ref. sub-section 4.2.2, equations 32 and 33).

4.2.4 Intermediate Motion

For intermediate motion (Fig. 20c) the flap displacement function § (y) has the form

$$§\ (y) = e_1 + (e_2 - e_1)\ \frac{y}{d} \tag{37}$$

where e_1 is the maximum displacement of the bottom of the flap from mean position and e_2 is the maximum displacement of the top of the flap at still water level from mean position.

From equation (21) the semi-height (amplitude) of the generated wave is

$$a = C_o \sinh md = 2m \frac{\int_o^d \{e_1 + (e_2 - e_1)\frac{y}{d}\} \cosh my\ (dy)}{\sinh md.\ \cosh md + md} \sinh md$$

$$= 2m \frac{\int_o^d e_1 . \cosh my\ (dy)}{\sinh md. \cosh md + md} . \sinh md$$

$$+ 2m \frac{\int_o^d (e_2 - e_1) \frac{y}{d} . \cosh my\ (dy)}{\sinh md. \cosh md + md} \sinh md$$

$$= \frac{2me_1 \int_o^d \cosh my\ (dy)}{\sinh md. \cosh md + md} . \sinh md$$

$$+ \frac{2m\ (e_2 - e_1) \int_o^d \frac{y}{2d} . (e^{my} + e^{-my}). dy}{\sinh md. \cosh md + md} . \sinh md$$

$$= \frac{2e_1 \{\sinh my\}_o^d}{\sinh md. \cosh md + md.} . \sinh md$$

$$+ \frac{2m\ (e_2 - e_1) \frac{1}{2d} \{ \frac{e^{my}}{m^2} (my-1) + \frac{e^{-my}}{m^2}(-my-1) \}_o^d}{\sinh md. \cosh md + md} . \sinh md$$

$$= 2e_1 . \frac{\sinh^2 md}{\sinh md. \cosh md + md}$$

$$+ 2(e_2 - e_1) \frac{(1 - \cosh md + md \sinh md)}{md\ (\sinh md. \cosh md + md)}$$

$$= Ke_1 + K' (e_2 - e_1) \tag{38}$$

where K is as previously defined in equation (27) and K' is as defined in equation (35).

From figure 19 it can be seen that when the wave generator is operating with intermediate motion the heights of the generated waves will lie between the heights achieved using piston motion and flap motion, assuming other conditions equal.

The value of C_n for intermediate motion is obtained by substituting for § (y) in equation (16) and integrating:

$$C_n = 2m_n \frac{\int_0^d \{e_1 + (e_2-e_1)\frac{y}{d}\} \cos m_n y. (dy)}{\sin m_n d. \cos m_n d + m_n d}$$

$$= 2m_n \frac{\int_0^d e_1 \cos m_n y \ (dy) + \int_0^d (e_2-e_1)\frac{y}{d}. \cos m_n y \ (dy)}{\sin m_n d. \cos m_n d + m_n d}$$

$$= 2e \frac{\sin m_n d}{\sin m_d n . \cos m_n d + m_n d} + 2(e_2-e_1)\frac{m_n d . \sin m_n d + \cos m_n d - 1}{m_n d(\sin m_n d . \cos m_n d + m_n d)} . \quad (39)$$

Force equations can now be determined using the same procedure as for piston motion. (Ref. sub-section 4.2.2, equations 32 and 33).

4.2.5 Discussion

A rigid paddle, double articulation, wave generator, of the type to be designed, is normally adjusted to operate with an intermediate paddle motion (Fig.20c) suited to the water particle motions of the generated wave. At one extreme of adjustment this motion becomes a piston motion suited to generating shallow-water waves and at the other extreme a hinged-flap motion suited to generating deep-water waves.

Biesel (Ref.1 - Wave Generator Theory) presents sample calculations which show that the transitory wave deformation (Eqn.26) occuring in front of the paddle is minimized when the paddle motion closely approximates the water particle motion of the generated wave. Therefore, in a flume of short length, it would appear particularly important that the wave generator paddle motion always be correctly adjusted.

With this type of wave generator it is still possible for the operator to generate all types of waves using either a piston motion or a hinged-flap motion. For a deep-water wave having $\frac{L}{d} = 0.5$, Biesel shows that for piston motion the ratio of water inertia to wave force is $\frac{F_i}{F_n} = \frac{35}{1}$, whereas for hinged-flap motion $\frac{F_i}{F_n} = \frac{13.6}{1}$. It becomes evident then, that more power is required to drive the wave generator when the paddle motion is not suited to the water particle motion of the generated wave.

The freedom which the wave generator operator has to adjust the paddle motion, brings up the question as to which operating mode (Fig.20) will produce the greatest loads on the paddle. From Fig. 19, it is evident that the wave amplitude coefficient K is always greatest for piston motion, resulting in the wave pressure (P_n) acting on the paddle, being greatest for this case (Eqn. 24). Wave inertia pressure (P_i Eqn.25) increases with decreasing wave period (length). If the water depth is sufficient to make the short-length wave a deep-water wave, and if the motion of the paddle is not suited to the water particle motion of the generated wave, then the water inertia pressure increases even more, and is greatest for piston motion. Therefore the wave forces acting on the paddle will reach a maximum when piston motion is used to generate short period waves.

When using Biésel's theory to calculate forces, it should be remembered that this theory is only of the first order of approximation and has not yet been fully proven.

4.3 Design Graphs

The behaviour of wave forces (F_n) and water inertia forces (F_i) is not readily apparent from Biésel's equations, so the author calculated values of F_n and F_i per unit flap width for various values of $\frac{L}{d}$ and plotted them in graph form. Because of the lengthy calculations involved, F_n and F_i curves were prepared only for the case of piston motion. This case is of prime interest as it yields the maximum forces to which a rigid paddle generator with double articulation could be subjected. For ease of design, the values of F_n and F_i were calculated in terms of e.d where e is the maximum displacement of the generator paddle in feet from its mean position (Fig. 20a) and d is the water depth in feet measured from the still-water level.

In preparing the graphs, the wave-amplitude coefficient K for piston motion was calculated for various values of $\frac{L}{d}$ using equation (27), the results being plotted in figure 21. A sample calculation for K for $\frac{L}{d} = 20$ is as follows:

$$K = \frac{2 \sinh^2 md}{\sinh md. \cosh md + md}$$

$$= \frac{2 \sinh^2 2\pi \frac{d}{L}}{\sinh 2\pi \frac{d}{L}. \cosh 2\pi \frac{d}{L} + 2\pi \frac{d}{L}}$$

$$= \frac{2 \sinh^2 \frac{2\pi}{20}}{\sinh \frac{2\pi}{20}. \cosh \frac{2\pi}{20} + \frac{2\pi}{20}}$$

$$= \frac{2 \sinh .314}{\sinh .314 \cosh .314 + .314}$$

$$= \frac{2 (.319)^2}{.319 \times 1.050 + .314}$$

$$= .315$$

The <u>maximum</u> wave force F_n per foot width of paddle for various values of $\frac{L}{d}$ was calculated for piston motion and <u>water on both sides</u> of the paddle using equation (32) and plotted in figure 22. A sample calculation for F_n for $\frac{L}{d} = 4$ is as follows:

$$F_n = 2\left(\frac{Ke\rho g}{m} \tanh md\right)$$

$$= 2\left(\frac{1.44 \text{ ed } 62.4 \times 32.2}{\frac{2\pi d}{L}} \tanh 2\pi \frac{d}{L}\right)$$

$$= 2\left(\frac{1.44 \times 62.4 \times 32.2 \text{ ed}}{\frac{2\pi}{4}} \tanh \frac{2\pi}{4}\right)$$

$$= 2\left(\frac{1.44 \times 62.4 \times 32.2 \times .917}{1.57} \text{ ed}\right)$$

$$= 105.2 \text{ ed lbs./ft. flap width.}$$

The <u>maximum</u> water inertial force F_i per foot width of paddle for various values of $\frac{L}{d}$ was calculated for piston motion and water on both sides of the paddle using equation (33) and plotted in figure 23. A sample calculation for F_i for $\frac{L}{d} = 4$ is as follows:

$$F_i = 2\rho g \left\{\left(\frac{c_1 d}{m_1 d} \cdot \tan m_1 d. \sin m_1 d\right) + \left(\frac{c_2 d}{m_2 d} \tan m_2 d. \sin m_2 d\right) + \left(\frac{c_3 d}{m_3 d} \cdot \tan m_3 d. \sin m_3 d\right)\right\}$$

from Fig. 27 $m_1 d = .85\pi = 2.67$ (radians)

$m_2 d = 1.93\pi = 6.06$ "

$m_3 d = 2.96\pi = 9.29$ "

from equation (32) $c_1 = \dfrac{2e \sin m_1 d}{\sin m_1 d.\ \cos m_1 d + m_1 d}$

$$= \frac{2e\ (.455)}{(.455)\ (-.891) + 2.67}$$

$$= .402\ e$$

similarly $c_2 = -\ .0761e$

and $c_3 = .0296e$

therefore $F_i = 2X62.4\{(\dfrac{.402\ ed}{2.67}\ X\ \dfrac{.455}{.891}\ X.455) + (\dfrac{-.0761ed}{6.06}\ X \dfrac{-.222}{.975}\ X(-.222))$

$$+\ (\frac{.0296ed}{9.29}\ X\ \frac{.1356}{-.991}\ X\ .1356)\}$$

$$= 2\ X\ 62.4\ (-.0349ed\ -.0006\ ed\ -.0001\ ed)$$

$$= -4.44\ ed.$$

The negative sign applies to the direction of action of force F_i relative to force F_n as illustrated in Fig. 24.

In studying Fig. 24 it becomes evident that there is considerable fluctuation in the load and hence in the required driving torque during one paddle oscillation. As the drive wheel rotates through quadrant 1 and 111 the load is relatively light. In quadrant 11 and 1V the load will be the same and reach a maximum value although in quadrant 11 the connecting rod is in tension while in quadrant lV the connecting rod is in compression.

Use of the graphs presented in Fig. 21, 22 and 23 greatly simplifies the design procedure when calculating wave heights and water forces, for a wave generator having a piston motion.

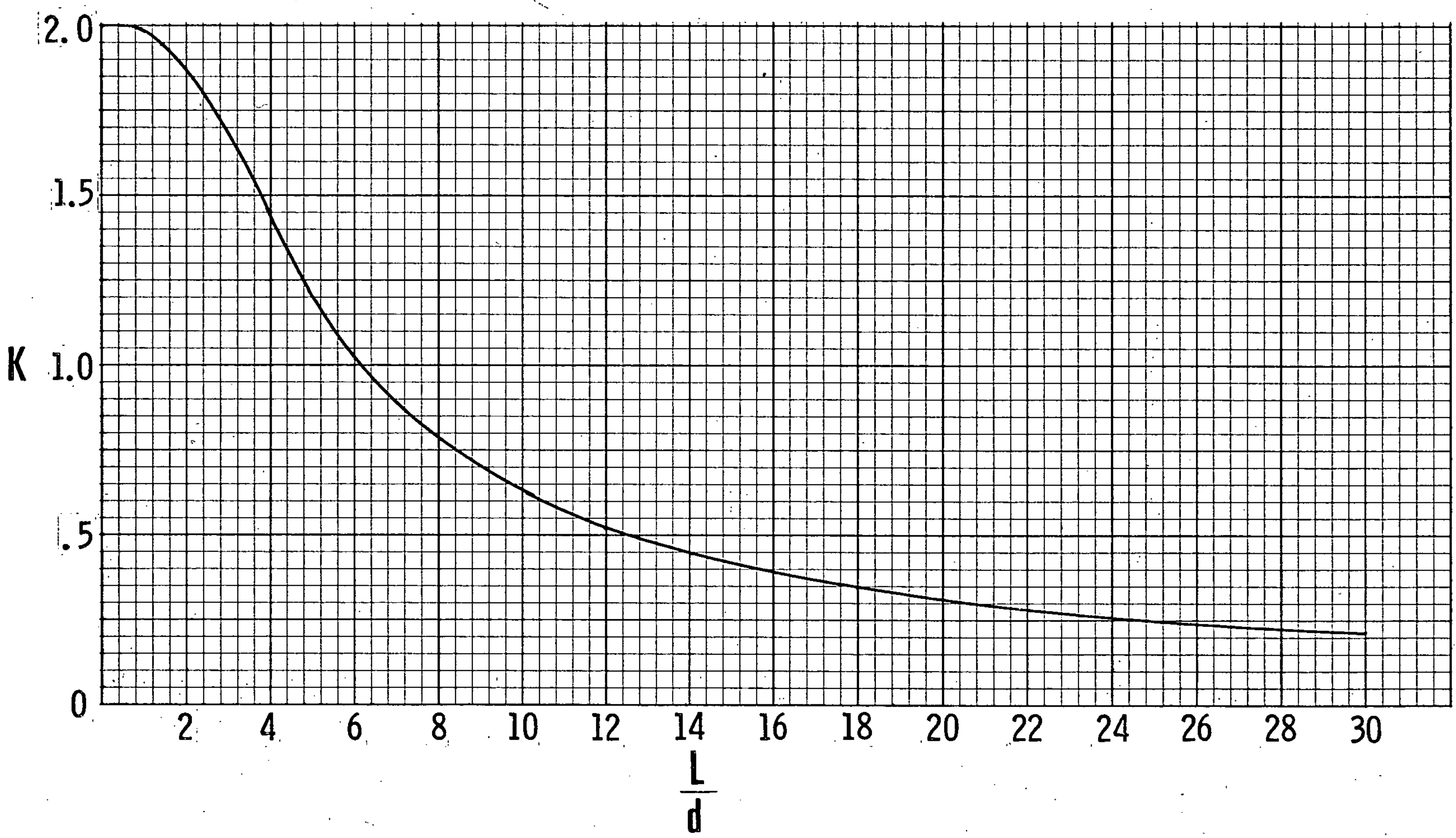


Fig. 21. Values of Wave Height Coefficient K (Piston Motion) for Various Values of $\frac{L}{d}$.

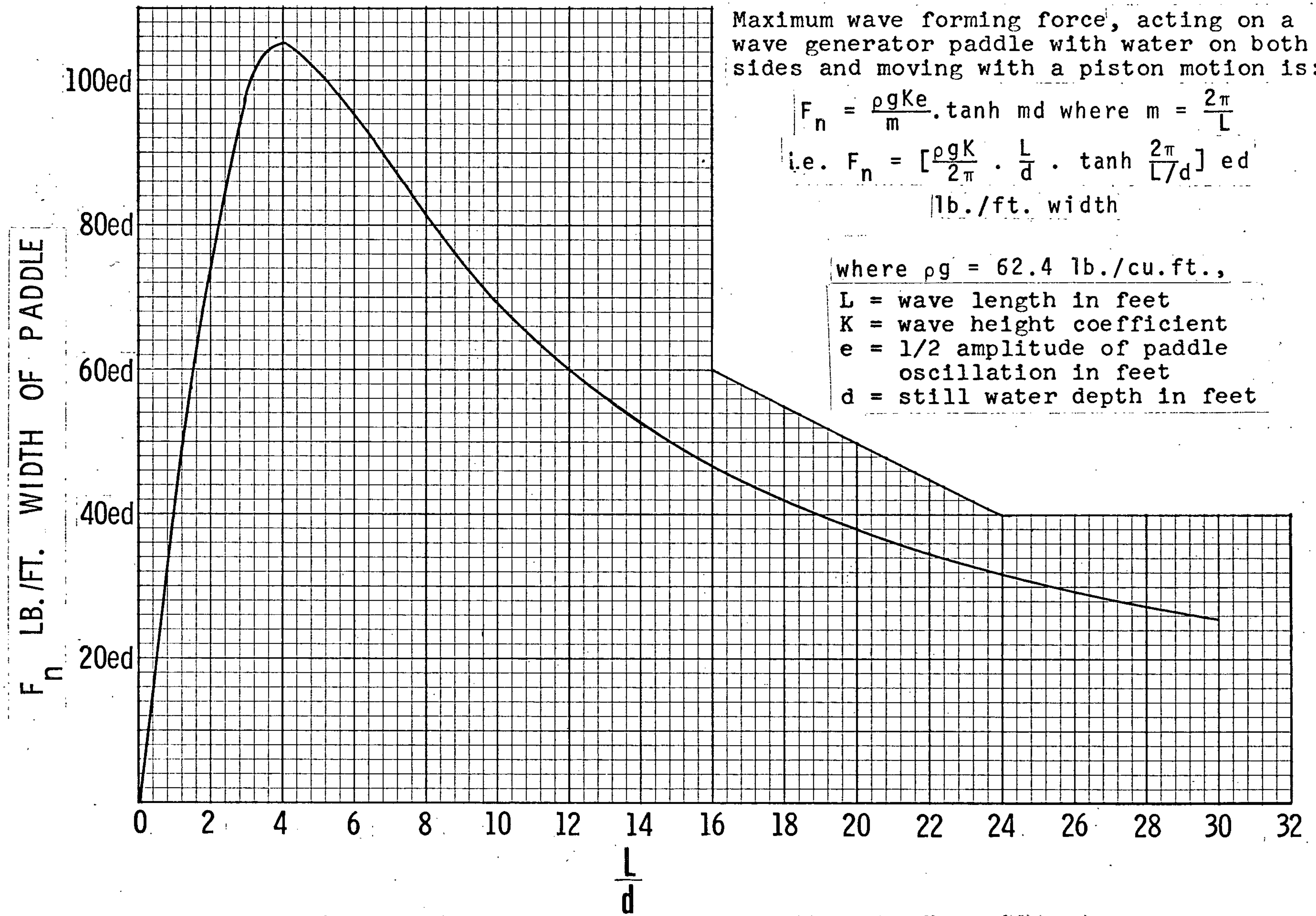


Fig. 22 - Maximum Values of Wave Force F_n per Foot of Paddle Width for Various Values of L/d.

Maximum water inertial force acting on a wave generator paddle moving with a piston motion with water on both sides of the paddle is:

$$F_i = 2\rho g\left[\left(c_1 \frac{d}{m_1 d}.\tan m_1 d.\sin m_1 d\right) + \left(c_2 \frac{d}{m_2 d}.\tan m_2 d.\sin m_2 d\right) + \left(c_3 \frac{d}{m_3 d}.\tan m_3 d.\sin m_3 d\right)\right] \text{ lb./ft. of paddle width.}$$

where

$$c_n = 2e \frac{\sin m_n d}{\sin m_n d.\cos m_n d + m_n d}$$

$m_n d$ = values read from Fig. 27

d = still water depth in feet

e = ½ total amplitude of paddle oscillation in feet.

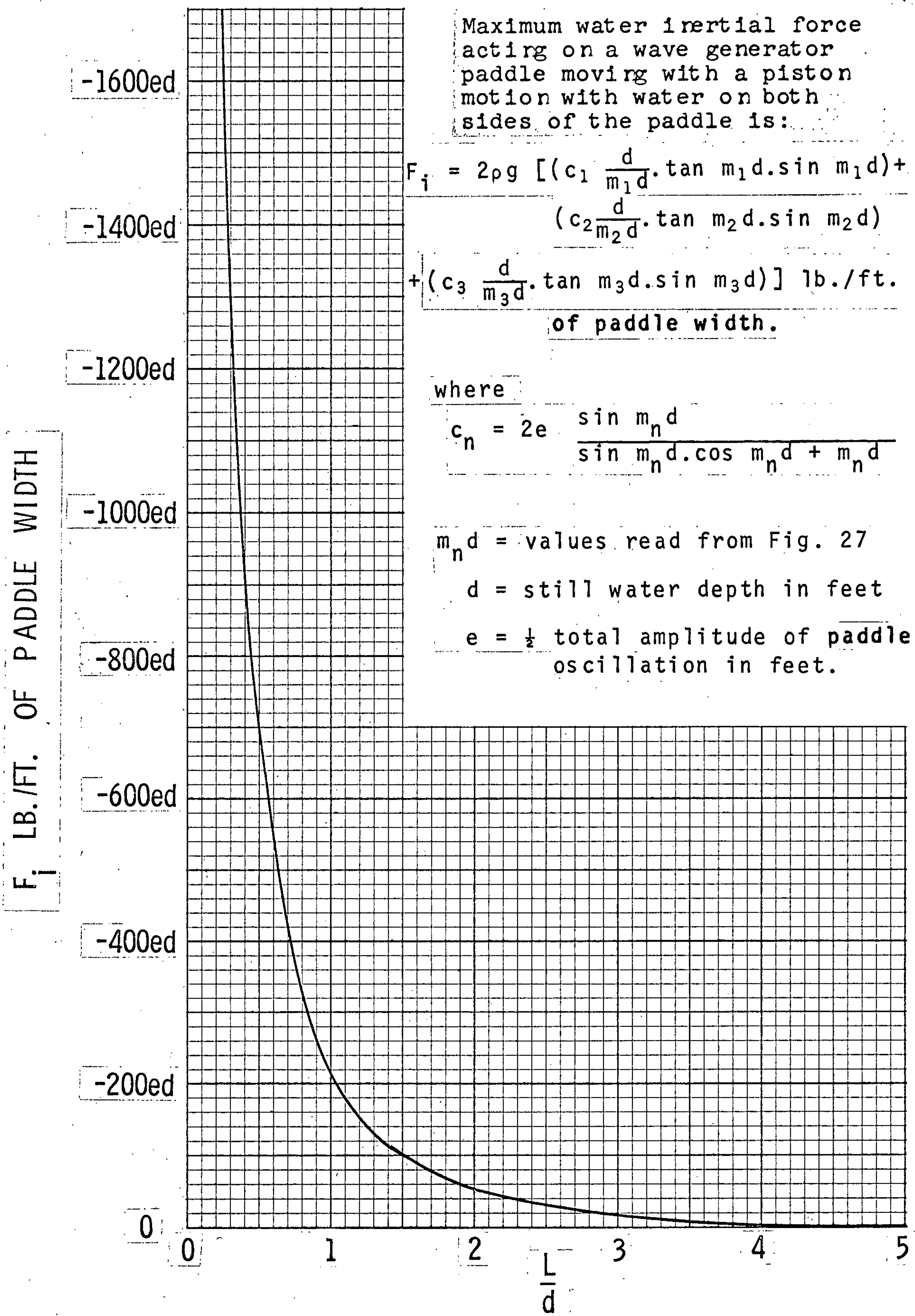


Fig. 23 - Maximum Values of Water Inertia F_i per Foot of Paddle Width for Various Values of L/d.

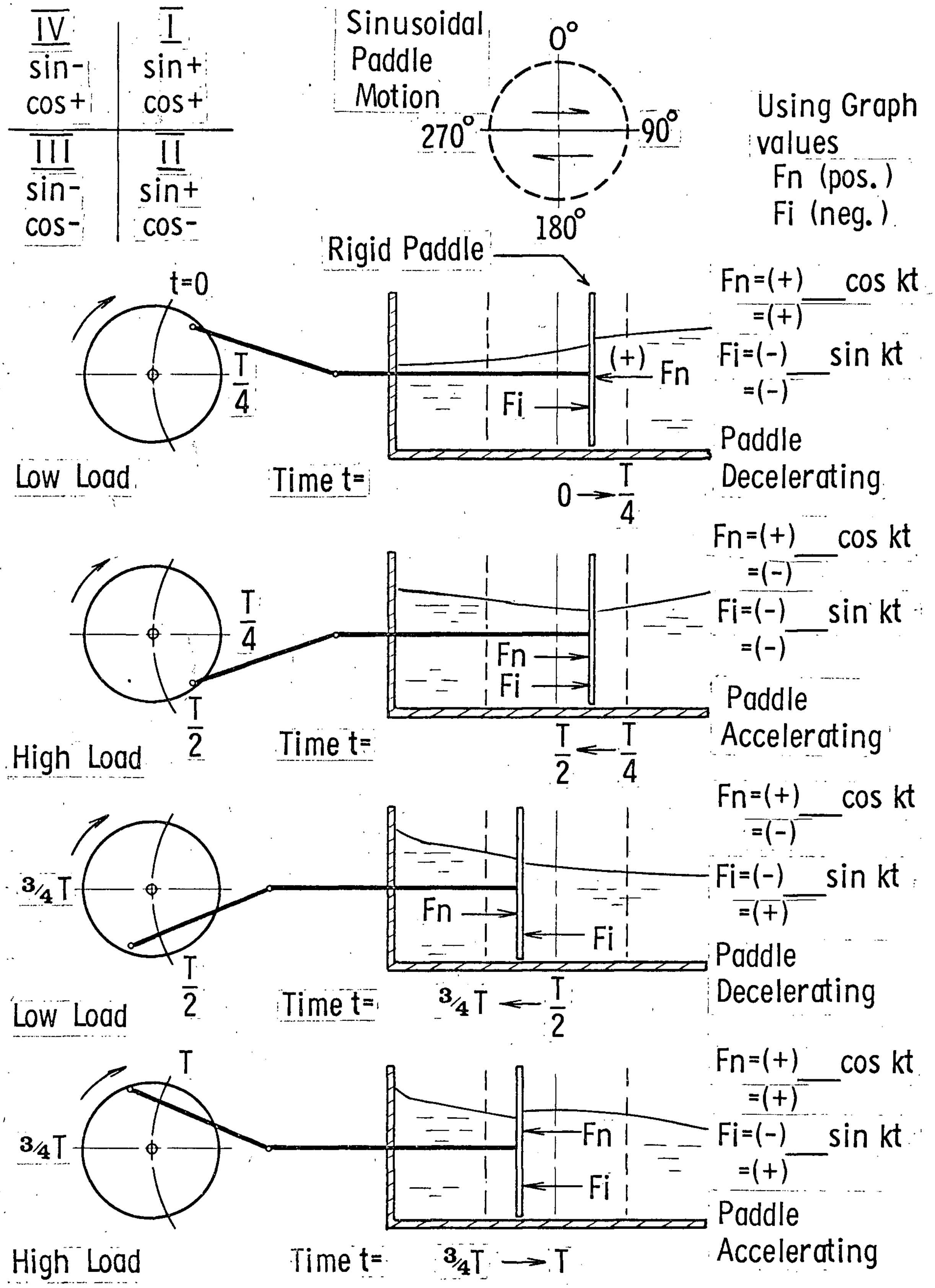


Fig. 24 - Direction of Forces Acting on Wave Generator Paddle During 1 Cycle of Oscillation.

5. RE-DESIGN OF A WAVE GENERATOR - isolating and rectifying the operating problems of the wave generator in the "small" flume.

5.1 Background

The wave generator installed in the "small", 21'-3 5/8" long, 8 3/8" wide and 10 5/8" deep, tilting, steel and glass flume (Fig. 2) in the University of British Columbia, Civil Engineering Department's Hydraulics Laboratory, is of the rigid paddle type with double articulation. The geometry of the paddle actuating mechanism was that developed by Coyer (Ref. 3 - Wave Generators). Fig. 25 shows the wave generator paddle mechanism adjusted to give a piston motion. Fig 27 shows the paddle mechanism adjusted to give a hinged-flap motion. Full details of this generator are given in a report by Pretious (Ref. 4 - Wave Generators).

The problem with this wave generator was that the drive system produced a very uneven paddle motion, resulting in irregularities in the wave form. Departures from the desired sinusoidal motion were extensive and visually evident.

Correction of the operating problems of this wave generator were undertaken before the design of the proposed new wave generator for the large flume, so that use could be made of the experience gained.

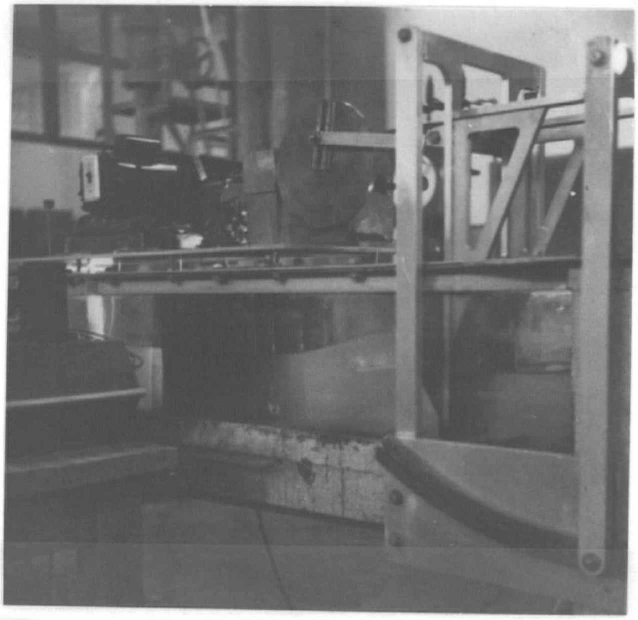


Fig. 25. The "small" Flume Wave Generator Adjusted for Piston Motion.

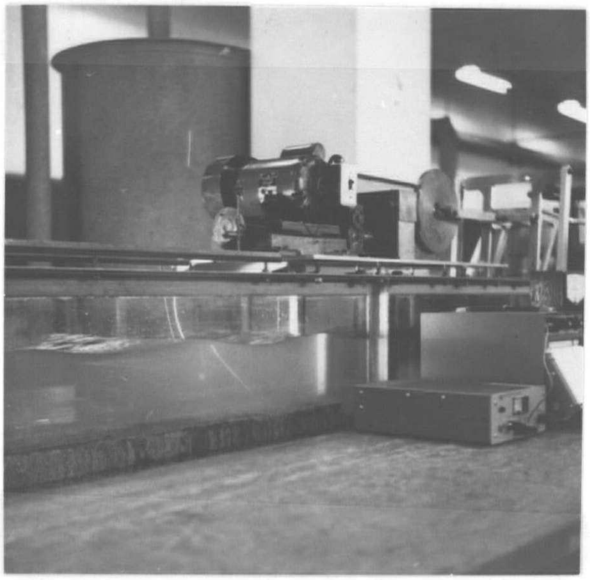


Fig. 26. Wave Surface Irregulatiries Due to Uneven Drive Motion.

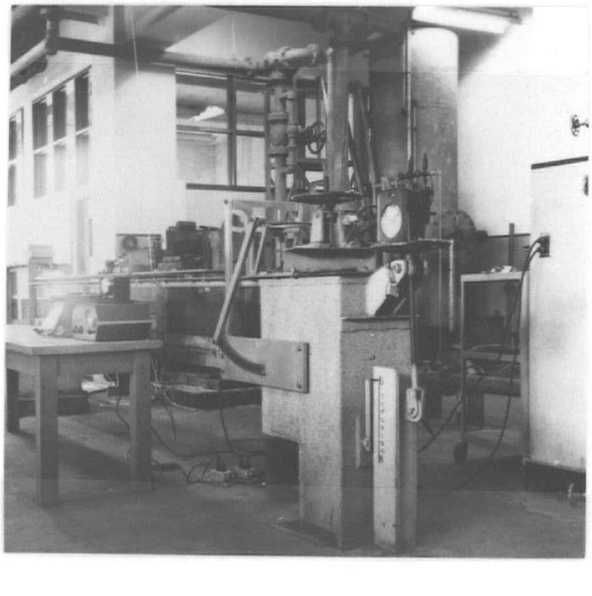


Fig. 27. The "Small" Flume Wave Generator Adjusted for Hinged-Flap Motion.

5.2 Isolating the Operating Problems

The predominant irregularities in motion consisted of two perceptible points of slowing down and two separate points of jerkiness in each rotation of the crank disc. The motion problem was previously assumed to be backlash only and a friction brake had been installed which rubbed on the crank disc circumference. Use of the brake reduced the jerkiness somewhat, but the slowing down still occurred. To isolate the causes, a methodical examination of the whole wave generator system was made, with the following results.

If vibration was a cause, paddle motion irregularities would increase for certain paddle oscillation frequencies, due to resonance, and would change significantly with changes in paddle mass. The results achieved by varying the paddle oscillation frequency and by changing the paddle mass about 30%, indicated that vibration was not responsible.

The crank disc was found to be slipping on its drive shaft due to an inadequate connection. This problem was rectified by machining a flange and welding it to the gear-box drive shaft. The crank disc was recessed slightly on the back to take the flange and the two units were bolted together (Fig. 28).

The drive system (Fig. 29) comprised three units. The prime mover was a 1/2-Hp., 1720 R.P.M., 110-volt, 60-cycle, single-phase, Brook Electric motor fitted with a vari-drive attachment. This attachment was a Boston, variable-speed drive (VDF6-1) consisting of an adjustable motor base and a spring-loaded, cone-belt sheave capable of a 3:1 speed reduction. The belt was connected to a Morton Engineering, worm gear, speed reducer, model T118, with a 20:1 speed reduction. This reducer was directly connected to a shop-manufactured, 2 speed, gear-box which allowed the selection of either a direct-drive slow setting, or a 1:3 speed increase, fast setting. The slow-setting output speed range was 90 to 30 R.P.M. while the fast setting range was 270 to 90 R.P.M.

The choice of a variable-speed drive unit was unfortunate, as the basic design of these units makes them unsuited for driving a wave generator. Variation in the diameter of the spring-loaded, cone-belt sheave on the motor drive shaft is achieved by moving the motor along its base, using the adjustment wheel shown in Fig.29(a) on the extreme right of the base. Moving the motor away from the pulley being driven, increases belt tension which forces the spring-loaded sides of the motor-mounted cone sheave apart, thus reducing the sheave diameter. This increases the ratio of speed reduction, which in effect reduces the output RPM of the system. When running, the diameter of the cone-sheave will be governed by initial belt tension and by tension due to load. For a constant load this unit gives good service, but for fluctuating loads the diameter of the cone sheave correspondingly fluctuates, resulting in variations in drive output speeds. For increasing load the output speed slows

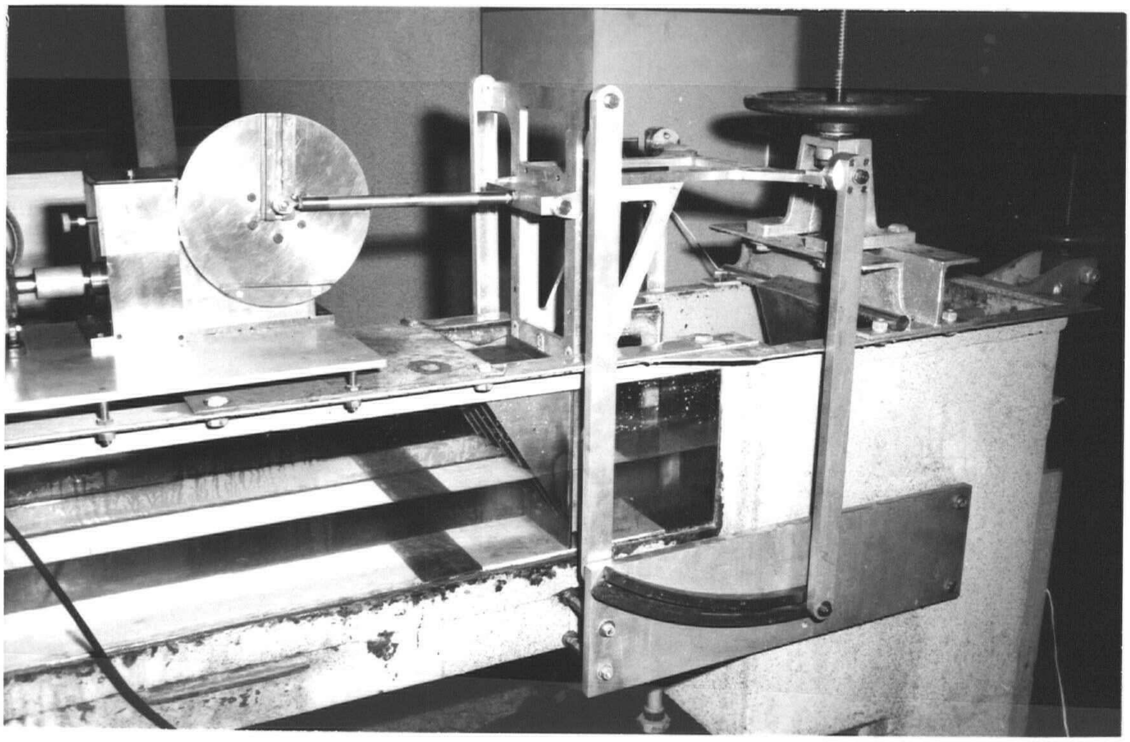


Fig. 28. The "Small" Flume Wave Generator Crank Disc

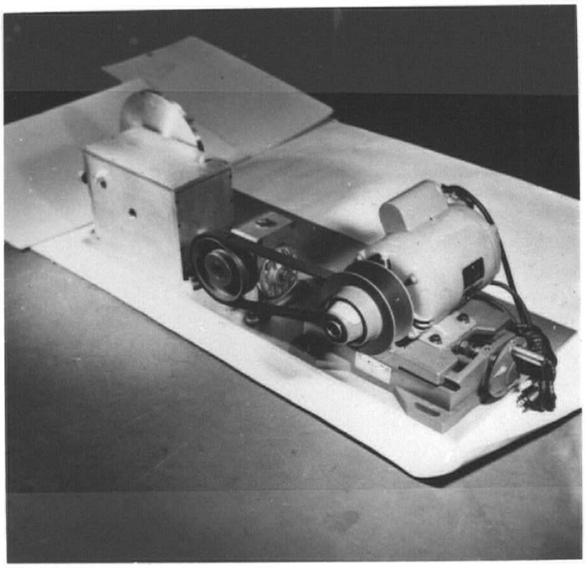
Bolted into Position on New Flange.

down and vice versa. As illustrated in Fig. 24, a wave generator load fluctuates. Obviously, it is physically impossible for this type of vari-drive to give the smooth, sinusoidal, paddle motion desired.

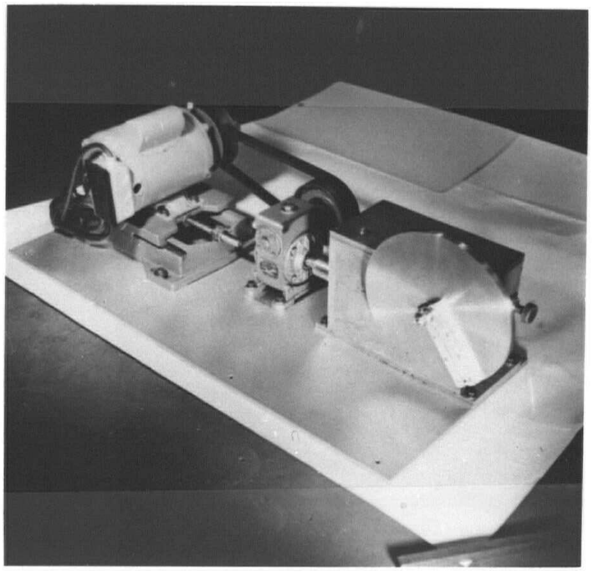
To check out this line of reasoning, the vari-drive cone-sheave was replaced by a fixed diameter pully shown in Fig. 31. The slowing down of the drive motion showed noticeable improvement but the jerkiness was still present.

This jerkiness was traced to the shop-manufactured gear box, the gear arrangement of which is shown in Fig. 30. A drive gear on the input shaft (the lower shaft barely visible in Fig. 30) is moved from one position to another along the shaft when a change is made from "low" to "high" speed. To allow the gear to slide over its key there has to be some clearance, and in this case the clearance was excessive. Because of the heavy #90 gear-oil bath this free-play was not evident unless firm pressure was applied. This was considered as the major source of the backlash and the resultant, jerky, drive motion.

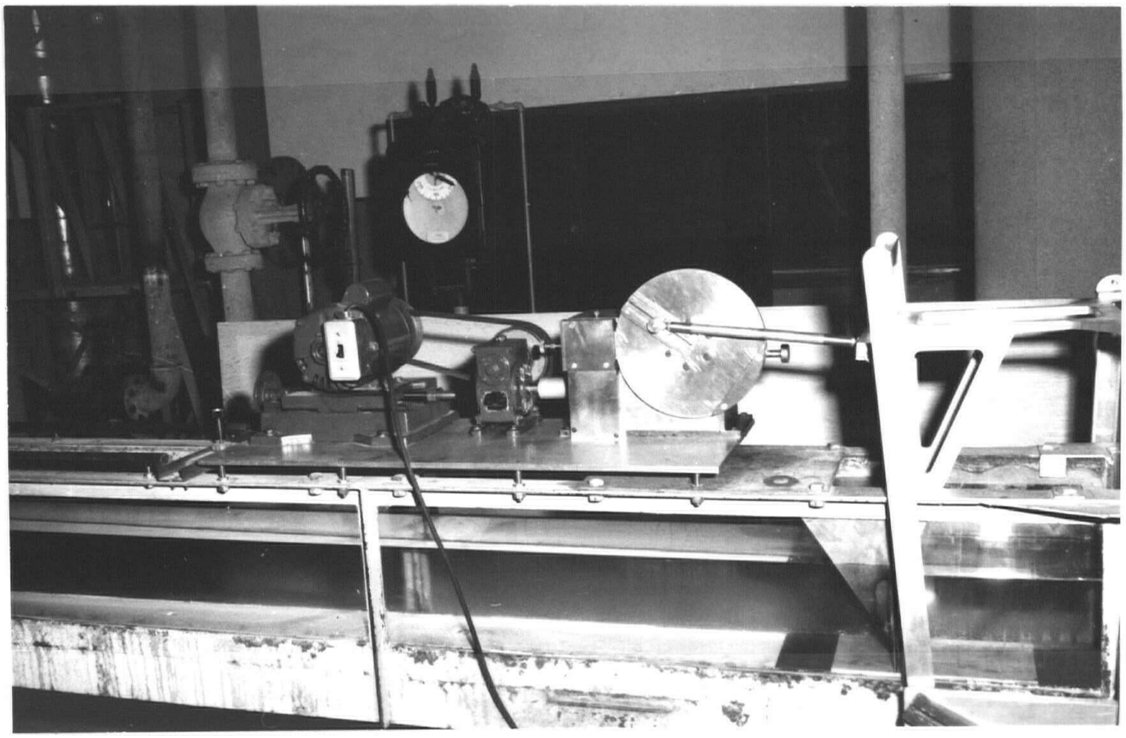
Further examination revealed a second cause of uneven motion. Bevelled wear was noticed on the ends of the teeth of the small gear on the right end of the intermediate shaft (Fig. 30) indicating that the teeth of this gear were not remaining parallel with the teeth of the gear driving it. The cause was assumed to be



(a)



(b)



(c)

Fig. 29. Three Views of the "Small" Flume Wave Generator Drive System Which Had an Output Speed Range of 270 to 30 R.P.M.

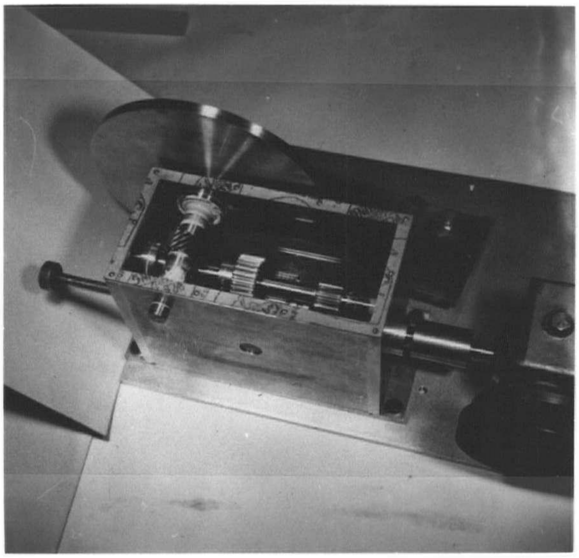


Fig. 30. Gear Arrangement in the Shop-Manufactured Gear Box, C.E.Dept., U.B.C.

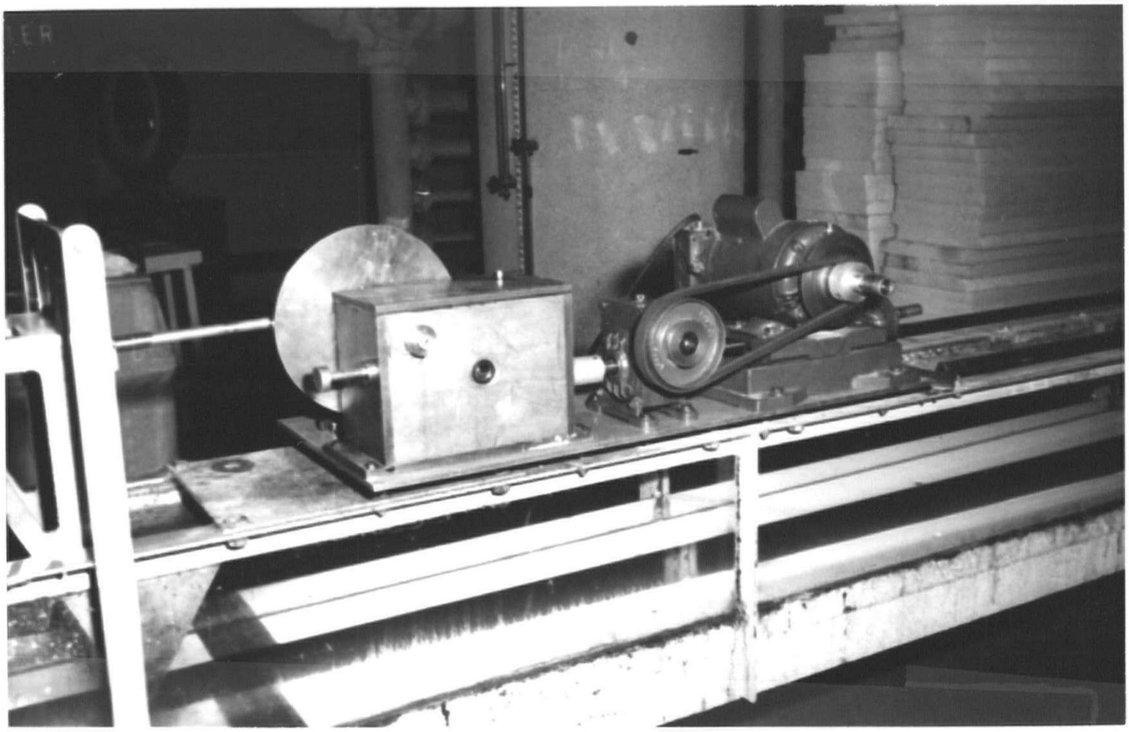


Fig. 31. "Small" Flume Wave Generator Motor Fitted with a Fixed Diameter Drive Pulley to Check Motion Improvements.

due to the helical gear, on the left end of the intermediate shaft, "walking" on the output shaft helical gear during movements of peak load, causing the intermediate shaft to bow out and then whip back as the load decreased. Dismantling of the intermediate shaft assembly, revealed an undersized inner shaft surrounded by spacing collars which gave it the appearance of having a larger diameter. Obviously, this shaft was not rigid enough for the job.

Since inertial forces produced by acceleration and deceleration of the water and paddle mass will produce forces acting on the connecting rod and crank disc, which reverse direction, it would appear necessary to minimize normal gear backlash in the drive system. This wave generator drive train employs 3 sets of mating gears. To reduce backlash it appears desirable to reduce this number to 1 gear set if possible.

The paddle of this wave generator has a brass front face which is free to move up and down (Fig. 28). The weight of this face causes it to ride on a friction-reducing, white, teflon sheet on the bottom of the flume (Fig.29), thereby stopping water leakage under the flap. The face formed a close fit in the channel and was found to be rubbing on parts of the glass walls of the flume evidenced by deposits of brass. On occasion, the friction with the glass appeared to induce a lateral vibration of the face causing additional momentary drag. Increasing the clearance between the face and flume sidewalls, cured this problem. However, the paddle face still makes a close fit in the flume and care should be taken to keep sand or grit away from the face during experiments.

The drive wheel (crank disc) was found to be slightly off from a precise parallel alignment with the connecting rod causing binding between the crank pin and the close-fitting sleeve bearing on the end of the connecting rod. Also, the aluminium base plate upon which the drive assembly was mounted (Fig. 29) flexed slightly with high load outputs, which would momentarily change the crank pin alignment. To cure this problem the drive assembly was re-aligned and the old connecting rod (Fig. 25) was replaced by a temporary connecting rod of the same length fitted with a self-aligning bearing which worked well. This practical aspect of maintaining precise alignment of the drive system, led to the use of a self-aligning bearing on the connecting rod of the proposed, new wave generator designed for use in the large flume.

The ratio of connecting rod length to maximum crank throw was 3.3:1. An improved sinusoidal motion would be obtained by increasing this ratio to about 6:1, or greater.

The two vertical arms supporting the rear of the paddle were each fitted at the top end with a large-diameter, sleeve-type bearing adjustable for friction (Fig. 28). The reason for these bearings was not clear, but it was assumed they were used to give more lateral stability to the wave generator and to provide limited friction as a counter to gear back lash. The adjustment nuts were slackened off to reduce the friction of these bearings, which was found to be considerable at the time of examination. Care will have to be taken in the future to see that these bearings are not tightly adjusted.

The lower ends of the vertical, rocker arms are positioned on a graduated arc mounted on plates located on each side of the flume (Fig. 28). One of these plates was tilted slightly out of vertical position. As the arm ends were fitted with sleeve type bearings this produced a slight binding. Re-alignment of this plate solved the binding problem. The practical solution of this problem suggested the use of self-aligning bearings on the lower ends of the rocker arms of wave generators of this double-articulating type.

Because this flume is a tilting type, the drive assembly must be supported on the flume making weight an item of consideration. The electric motor and its base (Fig. 29) weighed 86 lbs., the worm-drive gear box 20 lbs., and the shop-manufactured gear box 67 lbs., for a total of 173 lbs., not including the aluminium base plate upon which the whole drive system was mounted. This weight should be reduced.

Improved, wave-generator motion, will be obtained if the inertia of the moving parts is fairly low. This wave generator paddle was fabricated from solid aluminium plate, resulting in it being relatively heavy. Also the paddle face was made of brass. By drilling holes in the material, the weight of the paddle assembly was reduced from 22 to 18.5 lbs. and the brass face from 8.5 to 7.4 lbs. The total weight of 25.9 lbs. was still felt to be high, but no further weight reduction was done lest rigidity be sacrificed. The best solution would be to rebuild this paddle mechanism using thin aluminium tubing for rigidity and making the paddle face of aluminium plate instead of brass.

Besides the changes already made, it was concluded that a new drive system should be designed for this wave generator, including a longer connecting rod.

5.3 Re-Design of the "Small" Flume Wave Generator Drive System

5.3.1 Data on Existing Wave Generator and Channel

Maximum crank throw $e = 4.3''$

Useful channel depth $= 10\ 5/8''$

Operating water depth $d = 6.5''$

Channel width $w = 8\ 3/8''$

Channel length $= 21' - 3\ 5/8''$

5.3.2 Maximum Motor-Gear-Box Output RPM

To avoid significant surface tension effects (Fig. 5) choose the minimum wave length $L = 0.578 \times 12 = 6.94''$.

For still water depth d = 6.5",

$\frac{L}{d} = \frac{6.94}{6.50} = 1.068$ which is < 2, giving a deep-water wave.

From equation (3), the deep-water wave velocity is

$$C = \sqrt{\frac{g\ L}{2\ \pi}} = \sqrt{\frac{32.2 \times 6.94}{2\pi \times 12}} = 1.721 \text{ ft./sec.}$$

From equation (5) wave period $T = \frac{L}{C} = \frac{6.94}{12 \times 1.721} \equiv .336$ sec.

Therefore, maximum crank speed $= \frac{60}{.336} = \underline{178\ \text{RPM}}$.

<u>Note</u>: Galvin recommends a minimum wave period (T) of 0.5 sec. (120 RPM), with 0.75 secs. (80 RPM) preferred. The figure of 178 RPM was chosen to give a wider $\frac{L}{d}$ range for instructional purposes. Wave heights will be low for very short period waves. A lower RPM would reduce paddle stresses.

5.3.3 <u>Minimum Motor-Gear-Box Output RPM</u>

Galvin recommends at least two full wave lengths between paddle and spending beach, still-water line. Hence, maximum wave length $L = \frac{200}{2} = 100"$.

$\frac{L}{d} = \frac{100}{6.5} = 15.4$, which is < 20, giving a transition wave.

From equation (2) $C = \sqrt{\frac{gL}{2\pi} \tanh \frac{2\pi d}{L}}$

$$= \sqrt{\frac{32.2 \text{ X } 100}{2\pi \text{ X } 12} \tanh \frac{2\pi \text{ X } 6.5}{100}}$$

$$= 4.06 \text{ ft./sec.}$$

From equation (5) $T = \frac{100}{4.06 \text{ X } 12} = 2.05$ secs.

Hence, minimum crank speed = $\frac{60}{2.05}$ = 29 RPM

5.3.4 Estimated Maximum Wave Heights for Generator:

a) Piston motion yields maximum wave heights. From equation (21) semi-height (amplitude) of generated wave is $a = Ke$.

Hence wave height (H) = 2 K e.

For $\frac{L}{d} = 15.4$ and reading K from Fig. 21,

maximum H = 2 x .408 x .43 = 3.51"

Maximum wave heights for other values of $\frac{L}{d}$ were calculated and results plotted in Fig.32. (For intermediate motion the theoretical wave heights will be less.)

b) Maximum deep-water wave height is $H \simeq \frac{1}{7} L$ from equation (9).

For $\frac{L}{d} = 1.07$, L = 6.94" and $H \frac{6.97}{7} \simeq .96"$.

Height limits for other values of $\frac{L}{d}$ in the deep-water wave range were calculated and plotted in Fig.32.

c) Maximum shallow-water wave height from equation (10)

is $H = \frac{3}{4} d = \frac{3 \times 6.5}{4} = 4.87''$

d) Galvin's rule-of-thumb range, where waves lose stability due to depth, runs between

$$H = \frac{d}{1.5} = \frac{6.5}{1.5} = 4.3''$$

and $$H = \frac{d}{2} = \frac{6.5}{2} = 3.25''$$

5.3.5 <u>Maximum Paddle Forces Due to Water</u>

At 178 RPM $\frac{L}{d} = 1.07$.

Using Fig. 22

$$F_n = 42.2 \text{ edw} = 42.2 \times \frac{4.3}{12} \times \frac{6.5}{12} \times \frac{8.38}{12} = 5.73 \text{ lbs.}$$

Using Fig. 23

$$F_i = 190 \text{ edw} = 190 \times \frac{4.3}{12} \times \frac{6.5}{12} \times \frac{8.38}{12} = 25.8 \text{ lbs.}$$

Values of F_n and F_i were also calculated for crank RPMS of 151, 121, 88.5, 53 and 29. Results are recorded in Table I.

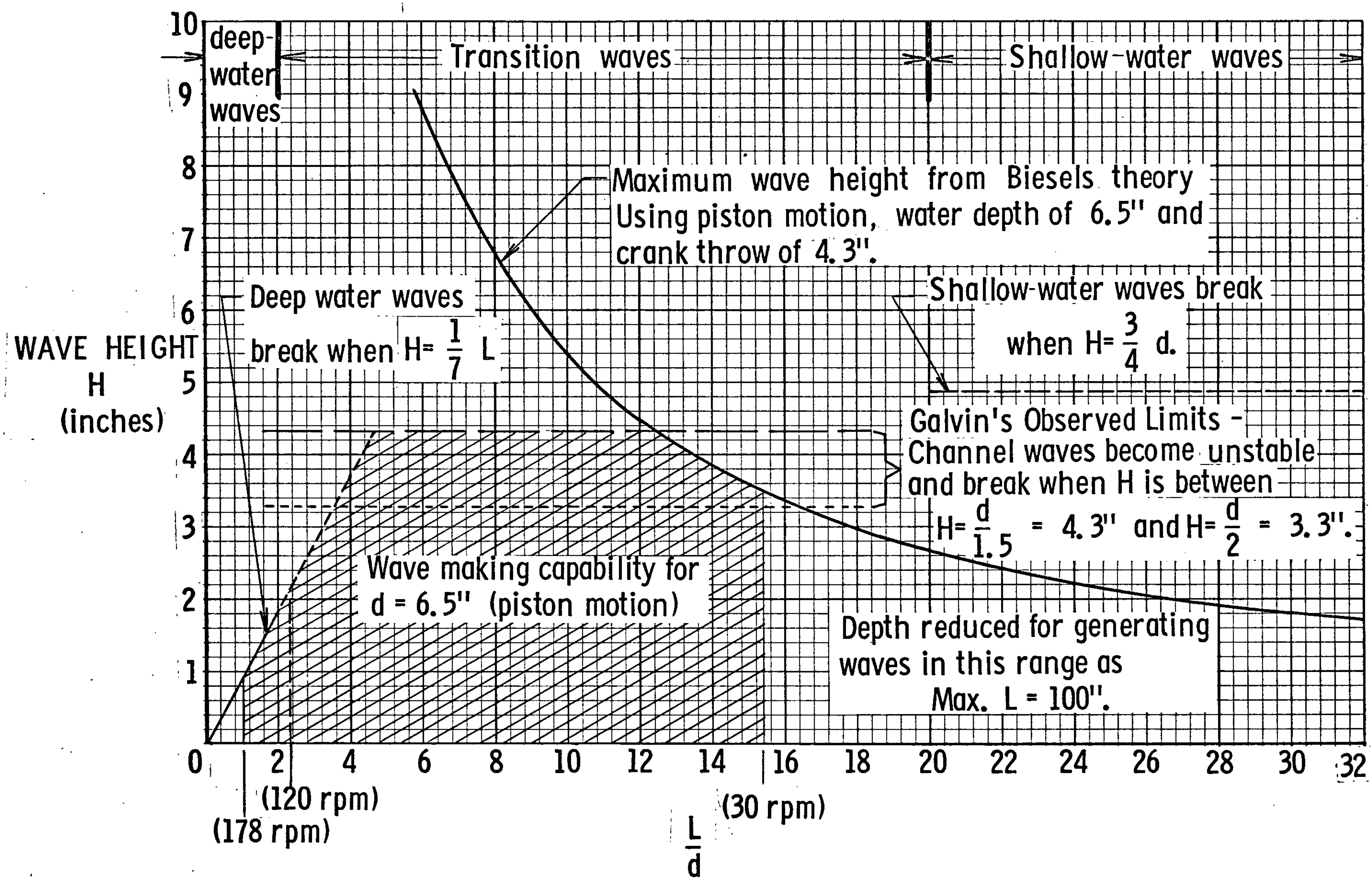


Fig. 32 - Estimated Wave Making Capability of "Small" Flume Wave Generator for a 6.5" Water Depth and Piston Paddle Motion.

5.3.6 Maximum Flap Force Due to Mechanism Inertia

Assume the motion of the paddle mechanism mass is that of a true piston. (The actual motion of the paddle mass is slightly curved and the light, paddle support arms move in arcs, but the assumption of a true piston motion for all parts is sufficiently accurate for design purposes and errs slightly on the safe side).

From Kent's Mechanical Engineer's Handbook (Ref. 4 - Mechanical Mechanisms) the inertia force of the paddle mechanism is

$$F_{im} = \frac{12 \quad WVc^2}{g \; R} \left(\cos \theta + \frac{R}{\ell} \cos 2\theta\right)$$

where W = total weight of reciprocating parts of paddle in lbs.

= 26 lb.

g = acceleration due to gravity in ft./sec^2.

Vc = velocity of crank pin in ft./sec.

ℓ = length of connecting rod in inches,

R = maximum crank throw radius in inches,

θ = angle in degrees defined on the diagram on Table II,

$\left(\cos \theta + \frac{R}{\ell} \cos 2\theta\right)$ values are given in Kent, Table - 2, page 7-38.

For a speed of 178 RPM

Vc = RPS X circumference of crankpin circle

$$Vc = \frac{178}{60} \times \pi \times \frac{2 \times 4.3}{12} = 6.68 \text{ ft./sec.}$$

$$\text{Maximum } F_{im} = \frac{12 \times 26 \times 6.68^2}{32.2 \times 4.3} (1.167) = 117.3 \text{ lbs.}$$

Values for F_{im} were also calculated for crank speeds of 151, 121, 88.5, 53 and 29 RPM. Results are recorded in Table I.

5.3.7 Calculating Motor-Gear-Box Output Torque

Calculations for the maximum motor-gear-box output torque at various crank RPMS are presented in Table II. Terms used relating to the mechanism are defined on a diagram included on the Table.

The ratio between connecting rod length and crank throw was set at $\frac{\ell}{R} = 6 : 1$. This ratio gives a connecting rod length of $\ell = 6 \times 4.3 = 25.8$ inches and represents a compromise between an improved sinusoidal motion, due to a longer connecting rod, and accessibility to the space in front of the paddle for installing wave filters.

The first two columns in Table II give the crank angle θ and the corresponding paddle angle kt from Biesel's theory. The paddle angle has been designated "nominal" as the geometry of the mechanism affects the relationship between crank angle θ and the paddle angle. Their relationship was approximated by considering the paddle as moving with true piston motion. (Actually the paddle moves along a slight "horizontal arc" as can be seen by studying Fig. 16 where the mechanism is shown adjusted for shallow-water waves (piston motion)). The approximated values of paddle angle (180° - kt) are designated "actual" and were found with the aid of Table I, page 7-04, Kent's Mechanical Engineer's Handbook (Ref. 4 - Mechanical Mechanisms) which gives piston(paddle) position for various crank angles. The values of cos (180°-kt) were then computed for use later on.

Values for the "tangential effort and velocity factor" $\sec \theta \sin (\theta + \phi)$ corresponding to chosen values of crank angle θ for $\frac{\ell}{R} = 6 : 1$, were obtained from Table I, P.7-36 in Kent and listed for use. Similarly, values for the "inertia and accelerat-ion factor" $\cos \theta + \frac{R}{\ell} \cos 2\theta$ were obtained from Table 2, P.7-38 in Kent and listed.

The first set of calculations was done for a crank speed of 178 RPM. The maximum value of F_n was read from Table I. Values of F_n cos (180-kt) were then computed for the listed crank angles. Similarly, values of F_i were calculated for the same crank angles. The maximum value of the mechanism inertia force F_{im} was read from Table I and the applicable value calculated for the required crank angles. It should be noted that the maximum value of

$\cos \theta + \frac{R}{\ell} \cos 2\theta$ for $\frac{\ell}{R} = 6 : 1$ is 1.167 and

therefore the maximum value of F_{im} must be divided by 1.167

CRANKWHEEL RPM	L (inches)	$\frac{L}{d}$	$C = \sqrt{\frac{gL}{2\pi}} \cdot \tanh \frac{2\pi d}{L}$ ft./sec.	$T = \frac{L}{C}$ (sec.)	F_n (lbs.)-This is max. value. (Fig. 31)	F_i (lbs.)-This is max. value. (Fig. 32)	edw = $\frac{4.3 \times 6.5 \times 8.375}{12 \times 12 \times 12}$ (Units in feet)	MAX. F_n (lbs.)	MAX. F_i (lbs.)	$\cos\theta + \frac{R}{\ell}\cos 2\theta$ Kent Handbook-Table 2 p.7-38 Max. Value for $R/\ell = 1/6$	CRANK VELOCITY V_c (ft./sec.)	$\frac{12W}{gR} = \frac{12 \times 26}{32.2 \times 4.3}$	MAX. $F_{im} = \frac{12WV_c^2(\cos\theta + \frac{R}{\ell}\cos 2\theta)}{g\ R}$ lbs.
178	6.94	1.07	1.721	.336	42.2 edw	190 edw	.1356	5.73	25.8	1.167	6.68	2.25	117.3
151	9.60	1.478	2.02	.396	57.6 edw	105.2 edw	"	7.81	14.28	"	5.66	"	84.2
121	15.08	2.32	2.53	.496	83.0 edw	36.0 edw	"	11.27	4.88	"	4.54	"	54.2
88.5	26.0	4.0	3.19	.678	105.2 edw	4.44 edw	"	14.28	.603	"	3.32	"	29.0
53	52.0	8.0	3.82	1.132	81.2 edw	≈0	"	11.00	0	"	1.989	"	10.40
29	100.0	15.4	4.06	2.05	48.4 edw	≈0	"	6.56	0	"	1.088	"	3.12

TABLE I - Calculation of Maximum Paddle Forces for Small Flume Wave Generator for Piston Motion, a Crank Throw of 4.3" and Water Depth of 6.5".

before multiplying by the next value of

$$\cos\,\theta + \frac{R}{\ell}\cos\,2\,\theta.$$

Values of F_n, F_i and F_{im} for a particular crank angle were summed to obtain P, the total force acting on the paddle. Then using the equation $F_t = P\,\sec\,\phi\,\sin\,(\theta \pm \phi)$ the effective tangential force, in pounds, acting on the crank pin was obtained. For a crank speed of 178 RPM the maximum value of F_t was -74.4 lb. The negative sign results from the relationship of forces as explained in Fig. 24 and can be ignored beyond this point. The maximum gear-box output torque required is 74.4 x 4.3 = 320 in.lb.

The maximum torque required at crank speeds of 151, 121, 88.5, 53 and 29 RPM was also computed and the calculations recorded in Table II. The torque requirements were plotted in Fig. 33 for various RPMS and values of $\frac{L}{d}$.

Maximum values of F_n, F_i, and F_{im} occuring at various speeds are plotted in Fig. 34 to show visually their relative significance for this wave generator, particularly that of mechanism inertia at higher speeds.

5.3.8 Protective High Speed Crank Throw Limitation

Because of the poor motion of the original drive system, the wave generator had not been operated at full crank throw at speeds much above 120 RPM. This was rather fortunate as some of the fastenings, and the axle supporting the ends of each of the four vertical-paddle-support arms, appear of inadequate strength for sustained high loadings.

The original ½ Hp. electric motor was of inadequate power rating for the load at high speed and full crank throw, but could easily have appeared adequate. Electric motors will operate at up to about 250% overload for short periods of time until a protective thermal switch cuts the power. Since the peak wave generator load is intermittent, occurring only briefly twice in each crank cycle (Fig.24), the motor would have driven the paddle for an appreciable time at higher RPMS and crank throws than those used, before motor overheating occurred.

Rather than rebuild parts of the paddle to guard against overstressing, it was felt that overstressing could be prevented by requesting the wave generator operator to limit the crank throw at high speed. This limitation would not limit the wave making capability of the wave generator.

The limitation was computed as follows:

At 178 R.P.M.

$$L = 6.94''$$

and $\frac{L}{d} = 1.07$ (Table I)

The maximum deep-water wave height is

$$H = \frac{1}{7} L = \frac{6.94}{7} = .99'' \qquad \text{(Fig. 32)}$$

For piston motion

$K \cong 2$ (Fig.21)

From equation (27)

$H = 2a = 2Ke$ where e is as defined in Fig. 20a.

The theoretical crank throw required to achieve maximum wave height for piston motion is

$$e = \frac{H}{2K} = \frac{.99}{2X2} = .25''$$

For hinged-flap operation

$K' \simeq 1.7$ (Fig.19)

and $e = \frac{H}{2K'} = \frac{.99}{2X1.7} = .29''$ where e is as defined in Fig. 20 B.

But e is located 6.5" above the channel bottom whereas the connecting rod connection to the flap is 20.25" above the channel bottom, (Fig. 16: Deep-Water Wave Setting). Therefore the required crank throw is $\frac{20.25}{6.5}$ X .29 = .91" .

As the calculated values of e are approached, the waves become unstable and break. For higher values of e, splash results (assuming 100% flap efficiency).

At 130 RPM where $\frac{L}{d}$ = 11.85, the maximum required crank throw

CRANK ANGLE θ (degrees)	(180°-kt) NOMINAL VALUE (degrees)	TABLE VALUE FOR $\frac{\ell}{R} = \frac{1}{6}$ *KENT-TABLE 1, p.7-04	.500-TABLE VALUE	$\text{SIN}(180\text{-kt}) = \frac{.5\text{-TABLE VALUE}}{.5}$	(180°-kt) ACTUAL VALUE FOR $\frac{\ell}{R} = \frac{1}{6}$ (Degrees)	COS (180-kt)	FOR $\ell/R=1/6$: SEC $\emptyset$ SIN $(\theta + \emptyset)$ *KENT-TABLE 1, p.7-36	FOR $\ell/R=1/6$: $\text{COS}\,\theta + \frac{R}{\ell}\text{COS}\,2\theta$ *KENT-TABLE 2, p.7-38
0	(90)	0	.500	1.000	90	0	0	1.167
10	(80)	.009	.491	.982	79.1	-.189	.202	1.142
20	(70)	.035	.465	.930	68.5	-.367	.396	1.067
30	(60)	.077	.432	.846	57.8	-.533	.572	.949
40	(50)	.134	.366	.732	47.1	-.682	.725	.795
50	(40)	.203	.297	.594	36.4	-.805	.849	.614
60	(30)	.281	.219	.438	26.0	-.899	.939	.417
70	(20)	.366	.134	.268	15.6	-.963	.994	.214
80	(10)	.454	.046	.092	5.28	-.996	1.014	.015
90	(0)	.542	-.042	-.084	-4.82	-.996	1.000	-.167

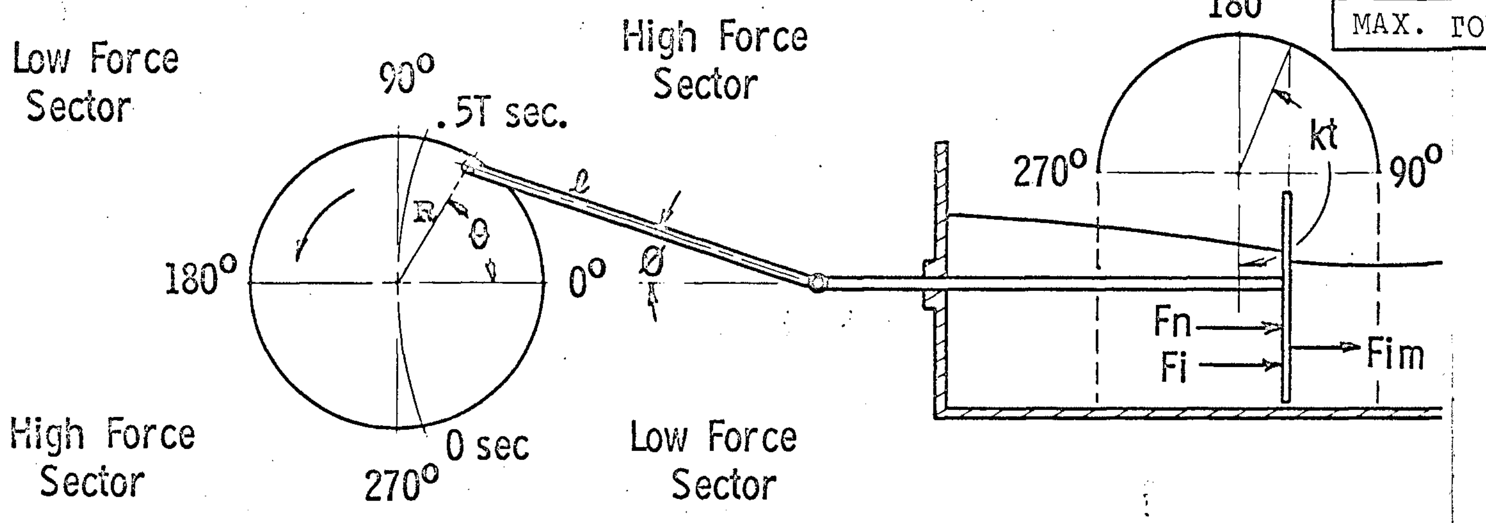
CRANK ANGLE θ (degrees)	178 RPM: F_n COS (180-kt) (MAX. F_n = 5.73 LB.)	178 RPM: F_i SIN (180-kt) (MAX. F_i =-25.8 LB.)	178 RPM: $F_{im} = \frac{12W\,V_c^2}{gR}(\text{COS}\theta+\frac{R}{\ell}\text{COS}\,2\theta)$ (MAX. Fim = -117.3 LB.)	178 RPM: $P = F_n + F_i + F_{im}$	178 RPM: $F_t = P$ SEC $\emptyset$ SIN$(\theta+\emptyset)$ LB.
0	0	-25.8	-117.3	-143.1	0
10	-1.083	-25.3	-114.8	-141.2	-28.5
20	-2.10	-24.0	-107.2	-133.3	-52.8
30	-3.05	-21.8	-95.4	-120.3	-68.8
40	-3.90	-18.88	-79.8	-102.6	-74.4
50	-4.61	-15.33	-61.7	-81.6	-69.3
60	-5.15	-11.30	-41.9	-58.4	-54.8
70	-5.51	-6.91	-21.5	-33.9	-33.7
80	-5.70	-2.38	-1.508	-9.59	-9.74
90	-5.70	+2.16	+16.75	+13.21	+13.21
	MAX.	TORQUE=74.4X4.3=320in/lb			

CRANK ANGLE θ (degrees)	151 RPM: F_n COS (180-kt) (MAX. F_n = 7.81 LB.)	151 RPM: F_i SIN (180-kt) (MAX. F_i =-14.28 LB.)	151 RPM: $F_{im} = \frac{12W\,V_c^2}{gR}(\text{COS}\theta+\frac{R}{\ell}\text{COS}\,2\theta)$ (MAX. Fim = -84.2 LB.)	151 RPM: $P = F_n + F_i + F_{im}$	151 RPM: $F_t = P$ SEC $\emptyset$ SIN$(\theta+\emptyset)$ LB.	121 RPM: F_n COS (180-kt) (MAX. F_n = 11.27 LB.)	121 RPM: F_i SIN (180-kt) (MAX. F_i =-4.88 LB.)	121 RPM: $F_{im} = \frac{12W\,V_c^2}{gR}(\text{COS}\theta+\frac{R}{\ell}\text{COS}\,2\theta)$ (MAX. Fim = -54.2 LB.)	121 RPM: $P = F_n + F_i + F_{im}$	121 RPM: $F_t = P$ SEC $\emptyset$ SIN $(\theta+\emptyset)$ LB.
0										
10										
20						-4.13	-4.54	-49.6	-58.3	-23.1
30	-4.16	-12.08	-68.5	-84.7	-48.4	-6.00	-4.13	-44.1	-54.2	-31.0
40	-5.33	-10.45	-57.4	-73.2	-53.0	-7.68	-3.57	-36.9	-48.2	-34.9
50	-6.29	-8.48	-44.3	-59.1	-50.2	-9.06	-2.90	-28.5	-40.5	-34.4
60	-7.02	-6.26	-30.1	-43.4	-40.7	-10.12	-2.14	-19.39	-31.7	-29.7
70	-7.51	-3.83	-15.45	-26.8	-26.6					
80						MAX.TORQUE = 34.9 X 4.3=151 inlb				
90										
	MAX. TORQUE=53X4.3 = 228 in/lb									

CRANK ANGLE θ (degrees)	88.5 RPM: F_n COS (180-kt) (MAX. F_n = 14.28 LB.)	88.5 RPM: F_i SIN (180-kt) (MAX. F_i =-.603 LB.)	88.5 RPM: $F_{im} = \frac{12W\,V_c^2}{gR}(\text{COS}\theta+\frac{R}{\ell}\text{COS}\,2\theta)$ (MAX. Fim = 29.0 LB.)	88.5 RPM: $P = F_n + F_i + F_{im}$	88.5 RPM: $F_t = P$ SEC $\emptyset$ SIN$(\theta+\emptyset)$ LB.	53 RPM: F_n COS (180-kt) (MAX. F_n = 11.00 LB.)	53 RPM: F_i SIN (180-kt) (MAX. $F_i \approx$ 0 LB.)	53 RPM: $F_{im} = \frac{12W\,V_c^2}{gR}(\text{COS}\theta+\frac{R}{\ell}\text{COS}\,2\theta)$ (MAX. Fim = 10.40 LB.)	53 RPM: $P = F_n + F_i + F_{im}$	53 RPM: $F_t = P$ SEC $\emptyset$ SIN$(\theta+\emptyset)$ LB.	29 RPM: F_n COS (180-kt) (MAX. F_n = 6.56 LB.)	29 RPM: F_i SIN (180-kt) (MAX. $F_i \approx$ 0 LB.)	29 RPM: $F_{im} = \frac{12W\,V_c^2}{gR}(\text{COS}\theta+\frac{R}{\ell}\text{COS}\,2\theta)$ (MAX. Fim = -3.12 LB.)	29 RPM: $P = F_n + F_i + F_{im}$	29 RPM: $F_t = P$ SEC $\emptyset$ SIN $(\theta+\emptyset)$ LB.
0															
10															
20	-5.24	-.561	-26.5	-32.3	-12.79										
30	-7.61	-.510	-23.6	-31.72	-18.10	-5.86	0	-8.46	-14.32	-8.2	-3.50	0	-2.54	-6.04	-3.46
40	-9.72	-.441	-19.80	-30.0	-21.8	-7.50	0	-7.09	-14.59	-10.57	-4.47	0	-2.13	-6.60	-4.78
50	-11.50	-.358	-15.30	-27.2	-23.1	-8.85	0	-5.48	-14.33	-12.18	-5.28	0	-1.643	-6.92	-5.88
60	-12.83	-.264	-10.38	-23.5	-22.1	-9.89	0	-3.72	-13.61	-12.79	-5.90	0	-1.118	-7.02	-6.58
70						-10.59	0	-1.908	-12.50	-12.42	-6.31	0	-.574	-6.88	-6.84
80	MAX.TORQUE =23.1X4.3=99 in.lb.					MAX.TORQUE=12.79X4.3=55 in.lb.					-6.54	0	-.0402	-6.58	-6.68
90											MAX. TORQUE=6.84X4.3=29 in.lb.				

* Carmichail, Colin - KENT'S MECHANICAL ENGINEERS' HANDBOOK, Design and Production Volume, 12th Edition, John Wiley and Sons, New York, 1950

TABLE II Calculations for Maximum Gear-box Output Torque Required at Various Speeds by the 21' - 3 5/8" X 8 3/8" x 10 5/8" Flume Wave Generator for Piston Motion, Maximum Crank Throw of 4.3" and Water Depth of 6.5".



is 0.5" for piston motion and 1.8" for hinged-flap motion. It is evident that at crank speeds above 130 RPM the crank throw required to achieve maximum theoretical wave height is less than ½ maximum crank throw, i.e. ½ x 4.3 = 2.15".

The torque demand for a crank throw setting of ½ maximum, or 2.15", was computed for crank speeds above 130 RPM and plotted in Fig.33.

In view of the drive unit finally selected it was concluded that the crank throw should be limited to ½ maximum (i.e.2.15") for crank speeds above 130 RPM, irrespective of the paddle motion involved. This limitation should restrict the maximum applied torque to about 170 in.lb. without, as shown, restricting the wave making capability.

5.3.9 Crank Wheel Mounting Height for Optimum Sinusoidal Paddle Motion.

The geometry of the paddle mechanism affects the desired sinusoidal motion of the paddle. Since transitional and shallow-water waves were felt to be of greater interest for model tests than deep-water waves, the best location of the crank wheel was determined for the paddle adjusted for piston-motion. This work was done graphically and the final calculation is shown in Fig. 35. It was decided that the crank wheel should be mounted 0.34" above the horizontal line passing through the centre of the joint where the connecting rod is attached to the paddle.

5.3.10 New Drive Unit

After investigating a number of drive systems, it was decided to use a 1 Hp. Winspeed SCR drive unit, with an RPM range from 180 to 9, to replace the original components. The torque curve is shown in Fig. 33.

The torque available from the new unit substantially exceeds the calculated value of torque required by the wave generator crank. This situation was considered desirable for three reasons:

a) Biesel's theory, used to compute the wave and water inertia forces, is only first order theory and is not yet fully proven;

b) bearing friction losses and friction losses from the paddle face dragging over the teflon pad (Fig. 28) and along the flume's glass sides are unknown; and

c) better speed regulations will be obtained if a unit having surplus power is used.

The 1 Hp. motor and speed reducer are flanged coupled providing a "tidy" installation. Their combined weight of 96 lb. is less than 60% of the weight of the previous drive system.

The Winspeed MCTR (2) worm-gear speed reducer, was checked for overhung, output-shaft, load and torque rating and found satisfactory.

5.3.11 Re-design of Connecting Rod

The length of the new connecting rod was previously specified at 25.8" (Section 5.3.7). Calculation of the maximum load was as follows:

Maximum load acting along the centre line of the connecting rod is

$$F_c = \frac{P}{\sqrt{1 - \left(\frac{R \sin \theta}{\ell}\right)^2}}$$ from

Kent's Mechanical Engineer's Handbook P. 7-37. Using values of P and θ from Table II for a crank speed of 178 RPM., the maximum value of F_c occurred for P=143.1 lb. and $\theta = 0^o$, that is

F_c = 143 lb. in compression.

A new connecting rod was designed of aluminium alloy 6061-T6 to carry this load. The crank end of the connecting rod was fitted with a self-aligning, ball bearing. A new crankpin was fabricated to accommodate this bearing and to fit the existing crank disc.

5.3.12 Summary of New Operating Specifications

design wave period range	= 0.34 to 2.1 secs.
design water depth (d)	= 6.5"
minimum wave length	= 6.9"
maximum wave length (d = 6.5")	= 100"
$\frac{L}{d}$ range for a depth of 6.5"	= 1.07 to 15.4

maximum crank throw $= 4.3''$

(estimated) maximum wave height $\simeq 4''$ when $5 < \frac{L}{d} < 13$

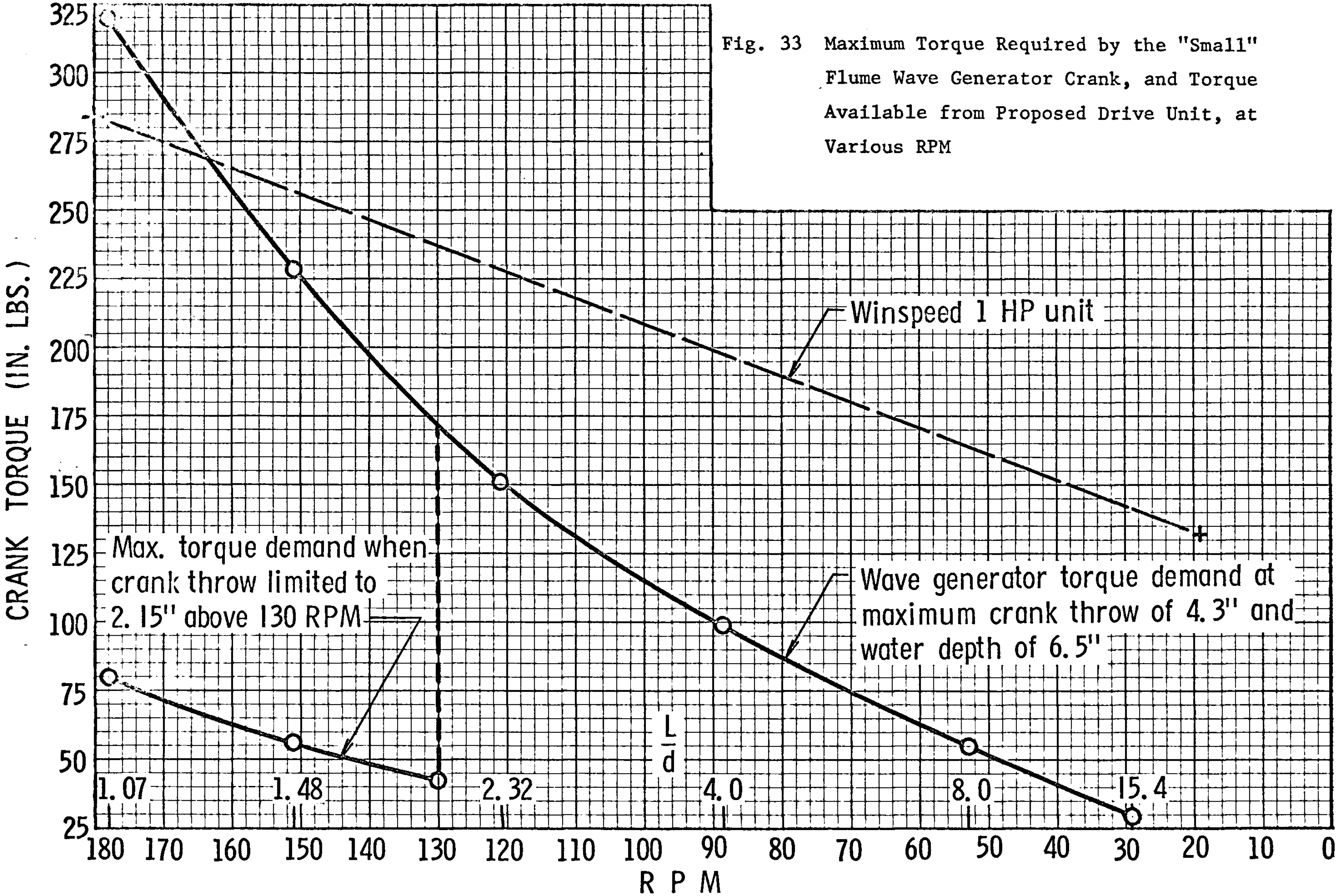


Fig. 33 Maximum Torque Required by the "Small" Flume Wave Generator Crank, and Torque Available from Proposed Drive Unit, at Various RPM

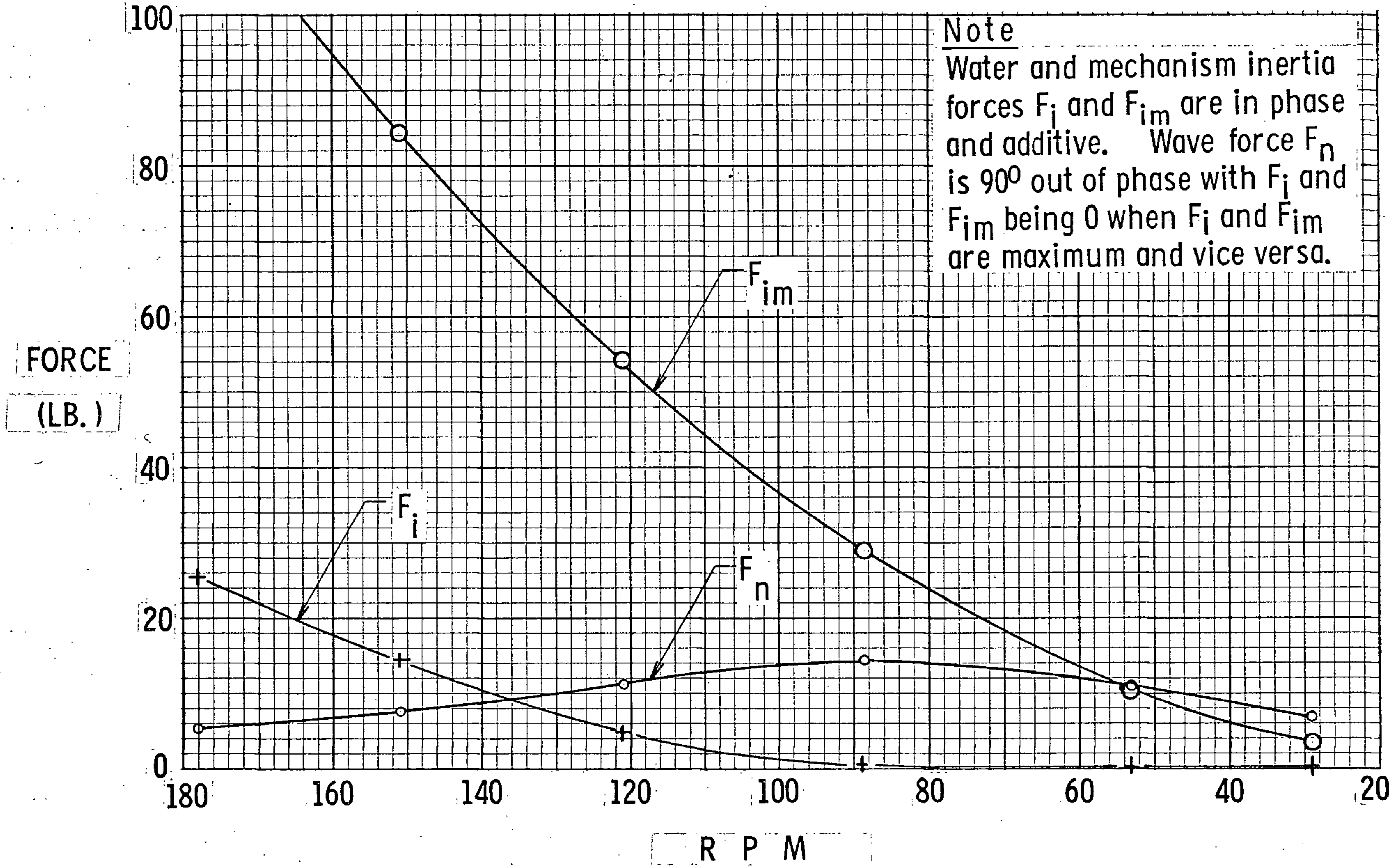


Fig. 34. A Plot of Maximum Values of Wave Forces F_n and Inertia Forces F_i and F_{im} to Show Relative Magnitudes.

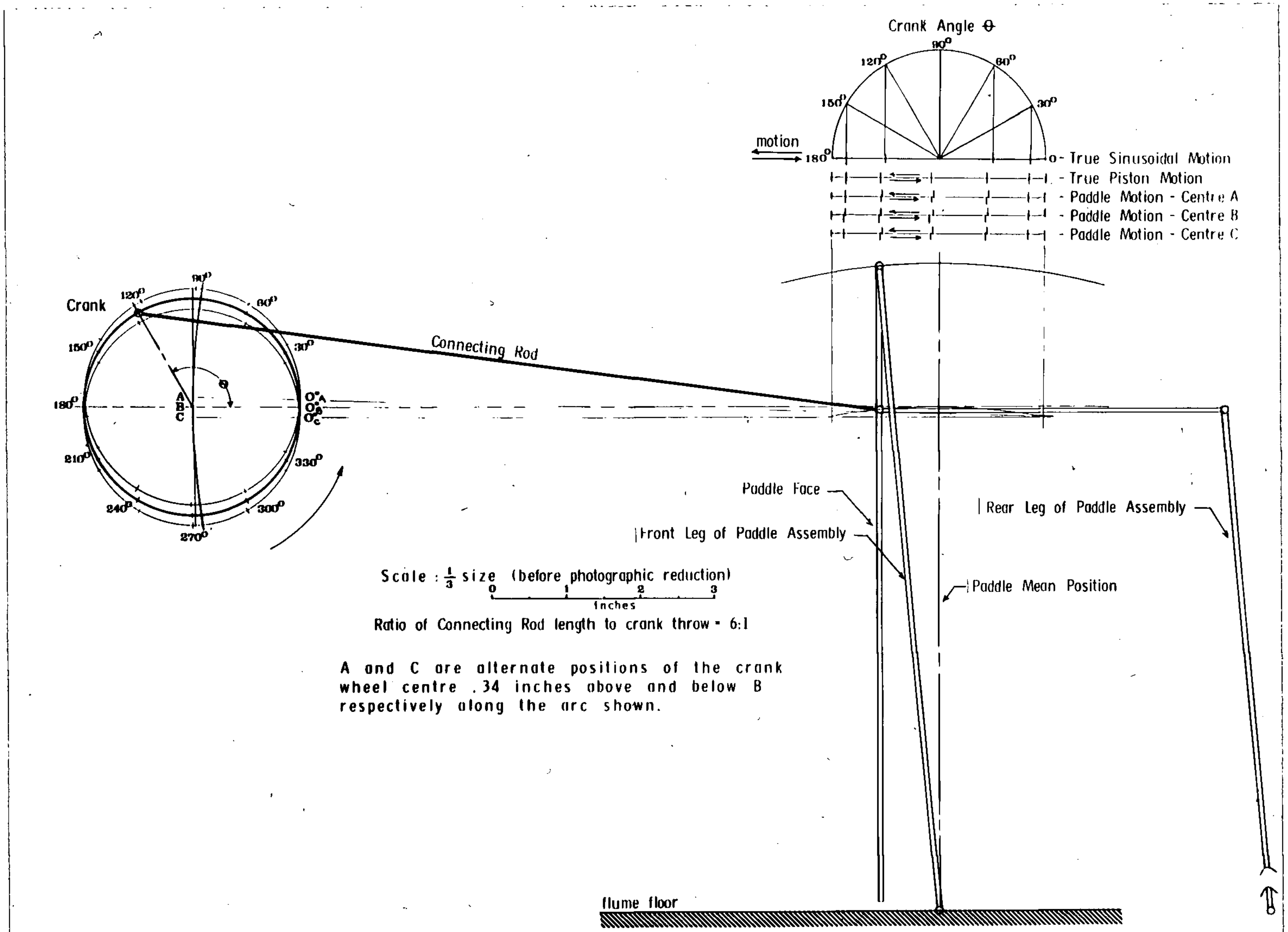


Fig. 35 Determining the Crank Wheel Location for Optimum Sinusoidal Displacement of the Paddle at the Piston Motion Setting.

6. DESIGN OF A WAVE GENERATOR - designing the proposed new wave generator for the "large" flume.

6.1 Scope

The large, 39'-4¼" long, 30" wide and 36" deep, fixed,steel and glass flume in the U.B.C. Hydraulics Laboratory is, in wave-channel practice, relatively short in length, and as a result requires a wave generator which establishes the correct wave form and water particle motion right at the paddle. It was therefore decided to design a wave generator for this flume of the rigid paddle, double articulation type (Section 3.6), using the geometry arrangement of Coyer (Ref. 3 - Wave Generators). The design of this wave generator is presented in this section.

6.2 Flume Data

The 39'-4¼" long, 30" wide and 36" deep flume is fabricated of steel, rigidly attached to the laboratory floor, and has glass panels on both sides along the middle third of its length. Views of the large flume are presented in Fig. 1. Interior detail, in the region chosen for installation of the proposed new wave generator, is shown in Fig. 36, while corresponding exterior detail is shown in Fig. 37. Basic flume layout and dimensions are given in Fig.38.

For design purposes:

usable flume length	=	39'-4¼"
total flume depth	=	36 3/4"

usable flume depth	=	33"
interior flume width	=	30"
and extreme exterior width	=	38 1/8"

6.3 Fundamental Design Decisions

The first decision concerned the paddle motion upon which force calculations should be based. The wave forces acting on the paddle of a double-articulation type wave generator will be greatest should piston motion be used to generate shallow-water waves (Section 4.2.5). Also, mechanism inertia forces (F_{im}) are highest for piston motion due to the manner in which the material masses move. Since the operator is free to adjust the wave generator mechanism for either piston, intermediate or hinged-flap motion (Fig. 20), it was decided that the proposed new wave generator should be designed on the basis of piston motion being used throughout the operating range.

Due to obstructions at the inlet end of the flume it was decided to install the wave generator at the tail-gate end, as indicated in Fig. 38. The distance from this end of the flume to the end of the last glass panel is only 25'-4". For purposes of observation, most tests would be conducted in the region of the last set of glass panels. To allow a maximum reach between the wave generator paddle and the test area, for the installation of wave filters and for the water-particle motion to stabilize, it was necessary to minimize the length of flume behind the paddle for the installation of wave absorbers. The volume of this space behind the paddle then

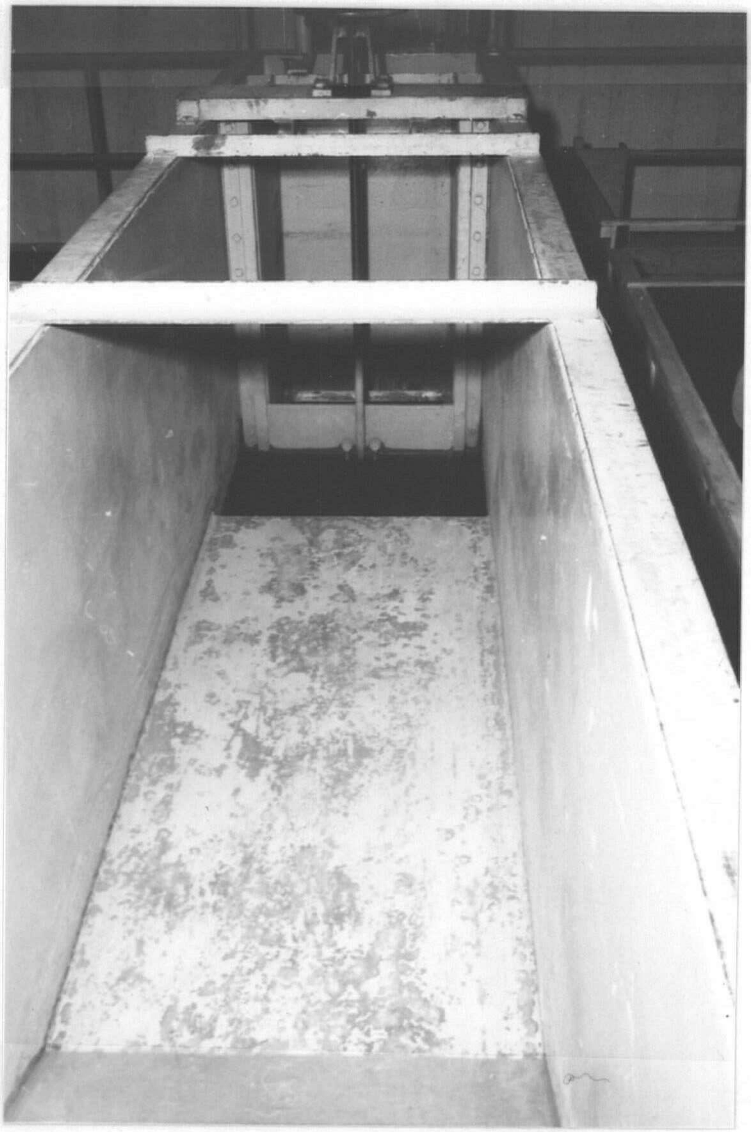


Fig. 36. Interior Detail of the 39'-4¼" Long, 30" Wide and 36" Deep Flume in the Region Chosen for Installation of the Proposed New Wave Generator.

Fig. 37. Exterior Detail of the 39'-4¼" Long, 30" Wide and 36" Deep Flume in the Region Chosen for Installation of the Proposed New Wave Generator.

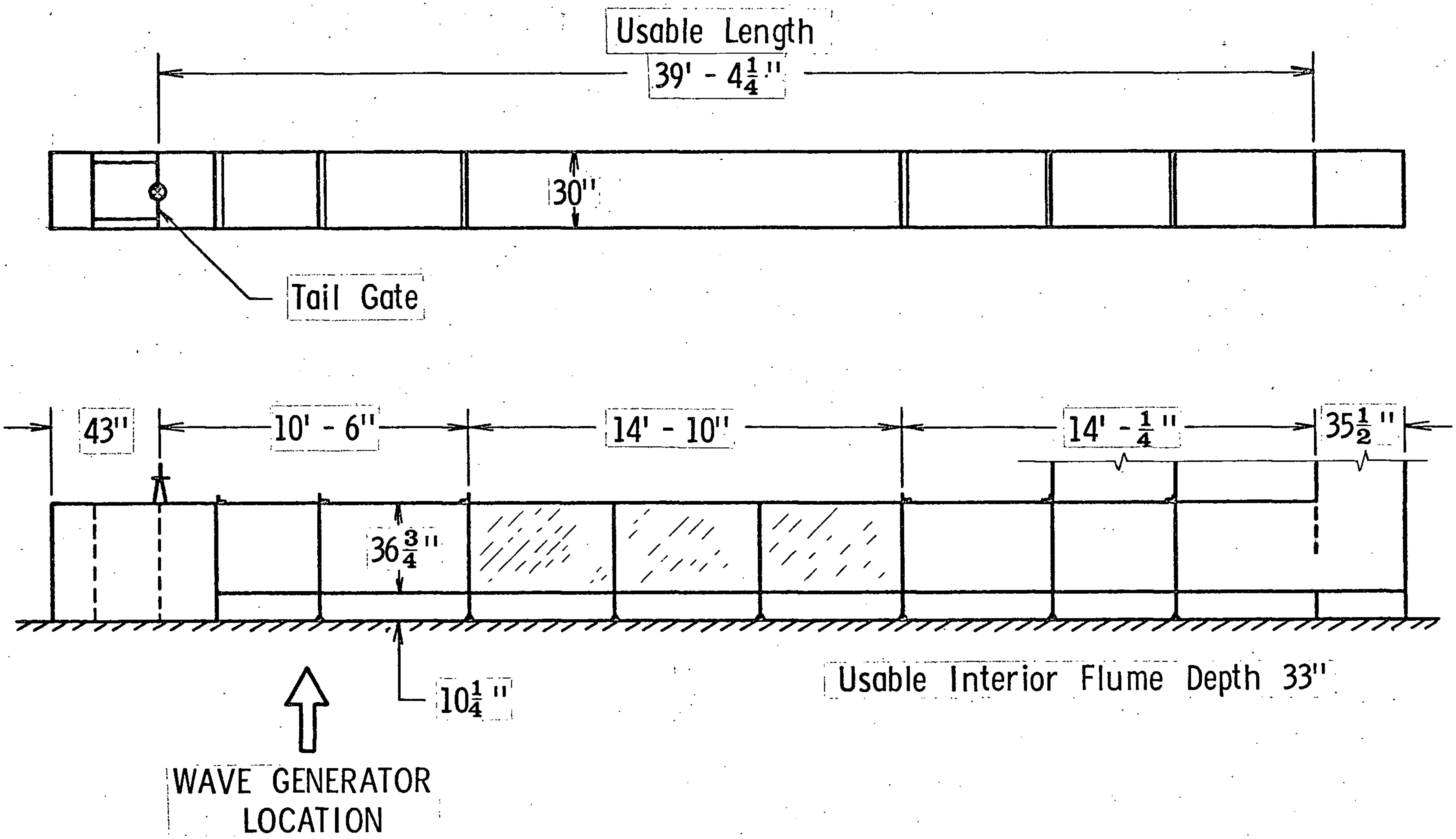


Fig. 38 Basic Layout of the "Large" Steel and Glass Flume.

becomes relatively small related to the volume of water displaced by the paddle, resulting in an appreciable fluctuation of the mean water level in it as the paddle oscillates. Because of this fluctuation, the decision on maximum crank throw, design water depth and generator paddle location became interdependent, necessitating some compromises.

The crank throw chosen for the drive mechanism will govern the wave heights obtainable. A maximum wave height in the range of 10" to 12" was desired. An initial estimate of wave heights for piston motion and different crank throw lengths was obtained from equation (27), rearranged to give wave height $H = 2a = 2\ Ke$. Similarly, wave heights for hinged-flap motion were estimated, using the equation $H = 2a = 2\ K'e$. Values of K and K′ were read from Fig.19. For hinged-flap operation allowance was made for the value of flap displacement at the water level being only about half the flap displacement at the connecting-rod connection. Normally, the wave generator paddle will be adjusted to operate with intermediate motion, suited to the water particle motion of the wave being developed. This will give a wave height lying between that of piston motion and hinged-flap motion.

When deciding the crank throw to be used, another point was kept in mind. Large paddle displacements rapidly increase the drive unit power requirements and costs, especially in the case of piston motion. Yet, when operating in the hinged-flap mode, a large crank throw is desirable to compensate for the reduced flap displacement at the still-water level, due to mechanism geometry.

Water depth affects the maximum wave height which can be obtained. Galvin's observation that waves become unstable in wave channels when the wave height H is in the region of $\frac{d}{1.5}$ to $\frac{d}{2}$, means that large paddle displacements will only result in breaking waves, unless adequate water depth is available. Since water depth sets a limit on the maximum wave height, and since deep-water wave heights are also limited by wave length ($H = \frac{1}{7} L$), the use of large paddle displacements, beyond a certain limit, serves only to increase the shallow=water wave heights. In setting the design water depth, wave height and channel freeboard were taken into account.

Due to the short reach between the paddle and the test area, it is desirable that the paddle motion should be as close to sinusoidal as feasible. Therefore it was decided to use a connecting rod length to crank-throw ratio of $\frac{\ell}{R} = \frac{7}{1}$. For a 10.5" crank throw, this means use of a 73.5" long connecting rod, which is about as long as the available space will permit without having the motor mount extending beyond the end of the flume.

Trial calculations were made to investigate the interlocking requirements of crank throw, water depth and paddle location in the flume. It was decided to use the following design limits:

a) maximum crank throw = 10.5",

b) maximum water depth = 25",

c) the paddle face positioned 5' from the interior tail-gate end of the flume; and

d) a connecting rod length to crank-throw ratio of

$\frac{\ell}{R} = \frac{7}{1}$.

6.4 Effect of Limited Flume Length Behind Paddle

The effect of the flume length behind the paddle being limited to 5 ft. was investigated. In its rearward stroke the paddle displaces $\frac{30}{12} \times \frac{25}{12} \times \frac{10.5}{12} = 4.56$ cu.ft. of water causing a rise in mean water level behind the flap of 5.3". For a 12" wave, the wave crests behind the flap will be $25 + 5.3 + \frac{12}{2} = 36.3''$ above the flume floor. With a total flume depth of 36.75", it is apparent that splash boards will be required along with an efficient wave absorber and possibly a cover plate, to prevent water spillage. This situation indicates that full crank throw will be excessive when used with the piston-type motion at the design depth of 25 in. However, the extra crank throw is desirable for intermediate or hinged-flap operations, where the amount of water displaced by the paddle is less than that for a piston-type operation and the effective paddle displacement at the water level is less than the crank throw due to mechanism geometry.

The fluctuating water level behind the paddle may serve to improve the paddle motion at higher crank speeds by reducing water and mechanism inertia effects. With the aid of Fig. 24 it can be seen that as the paddle moves forward, the water level behind the paddle drops, producing an unbalanced hydrostatic pressure on the front of the paddle which works against, or absorbs, the inertia energy of the water and paddle mechanism. As the paddle starts to move backwards, the difference in water level will assist in accelerating the paddle. In the third diagram the pressure difference due to the higher water level behind the paddle absorbs some of the inertia energy. In the last diagram the pressure difference will aid in accelerating the paddle.

6.5 Design of the Proposed, New Wave Generator Drive System

6.5.1 Summary of Basic Design Data

Usable flume length	=	39'-4¼"
Total flume depth	=	36 3/4"
Usable flume depth	=	33"
Interior flume width	=	30"
Maximum crank throw	=	10.5"
Maximum water depth	=	25"
Mean position of paddle face from interior tail-gate end of flume	=	5'
Connecting rod length to crank throw ration $\frac{\ell}{R} = \frac{1}{7}$		
Connecting rod length	=	73.5"

6.5.2 Maximum Crank RPM

To avoid surface tension effects, Fig. 5, choose a minimum wave length of L = 12". Then at the design depth of d = 25"

$\frac{L}{d}$ = .48 which is less than 2, indicating deep-water wave.

From equation (3) the deep-water wave velocity is

$$C = \sqrt{\frac{gL}{2\pi}}$$

$$= \sqrt{\frac{32.2}{2\pi} \times \frac{12}{12}}$$

$$= 2.26 \text{ "/sec.}$$

From equation (5)

$$T = \frac{L}{C} = \frac{12}{12 \times 2.26} = .443 \text{ sec.}$$

requiring a maximum crank speed = $\frac{60}{.443}$ = 135.5 RPM.

Galvin recommends a minimum wave period of 0.5 sec.(120 RPM) with 0.75 sec (80 RPM) preferred. These slower crank speeds reduce inertial forces considerably, with only a slight loss in deep-water wave making capability. Therefore two other values of crank speed, wave length (L) and wave length to water depth ratios($\frac{L}{d}$) were computed, for use in selecting the maximum speed of the power unit:

Assume minimum L = 20"

then $\frac{L}{d}$ = 0.80

C = 2.92 ft./sec.

T = .571 sec.

requiring a crank speed $= 105$ RPM.

Assume a crank speed $= 88$ RPM

then $T = .683$ sec.

$C = 5.12\ T = 3.50$ ft./sec.

$L = C\ T = 28.7''$

and $\frac{L}{d} = 1.15$

These calculations were summarized in Table III.

Final selection of maximum crank speed will be done later, when the drive system is selected, but the speed will lie somewhere between 80 and 135.5 RPM.

6.5.3 Minimum Crank RPM

Galvin recommends having at least two full wave lengths between the mean paddle position and the beach, still-water line. Therefore, the maximum wave length is

$$L = \tfrac{1}{2}\ (34' - 4\tfrac{1}{4}'') = 17' - 2\ 1/8'' = 206''$$

For $d = 25''$

$$\frac{L}{d} = \frac{206}{25} = 8.24 \text{ which is a transition wave giving}$$

$$C = \sqrt{\frac{gL}{2\pi} \tanh \frac{2\pi d}{L}} = \sqrt{\frac{32.2 \times 206}{2\pi \times 12} \tanh \frac{2\pi}{8.24}} = 7.52 \text{ ft./sec}$$

$$T = \frac{206}{7.52 \times 12} = 2.28 \text{ sec.}$$

and a minimum crank speed $= \frac{60}{2.28} = 26.3$ RPM

Assuming a minimum, usable water depth of $d = 6''$ gives

$\frac{L}{d} = \frac{206}{6} = 34.8$ which is greater than 20 and is a shallow-water wave.

From equation (4)

$$C = \sqrt{g\ d} = \sqrt{32.2 \times \frac{6}{12}} = 4.01 \text{ ft./sec.}$$

From equation (5)

$$T = \frac{206}{4.01 \times 12} = 4.28 \text{ sec.}$$

and minimum crank speed $= \frac{60}{4.28} = 14$ RPM

6.5.4 Maximum Water Forces Acting on Paddle

Maximum values of the wave forces (F_n) and the water inertia forces (F_i) acting on the paddle during piston motion at speeds of 135.5, 105 and 88 RPM, were calculated with the aid of Fig. 22 and 23 and listed in Table III. Since the maximum wave forces for different $\frac{L}{d}$ values occur when $\frac{L}{d} = 4$, Table III was extended to include values of $\frac{L}{d}$ equal to 2, 3 and 4. Values of F_n and F_i for $\frac{L}{d}$ greater than 4 were not considered, as these forces both decrease beyond this point for any fixed water depth and crank throw.

6.5.5 Maximum Paddle Force Due to Mechanism Inertia

To calculate forces due to the inertia of the paddle mechanism, it was assumed that the piston-type motion of the paddle mechanism was that of a true piston. This assumption is sufficiently accurate for design purposes and errs on the safe side. The calculations for mechanism inertia at various speeds were carried out as previously explained in Section 5.3.6 and the results listed in Table III.

For these calculations the weight of the paddle mechanism was estimated at 160 lbs.

A sample calculation for paddle mechanism inertia at 88 RPM is as follows:

$$F_{im} = \frac{12 \ WVc^2}{g \ R} \left(\cos \theta + \frac{R}{\ell} \cos 2 \theta\right)$$

where $V_c = \frac{88}{60} \times \pi \times \frac{21}{12} = 8.06$ ft./sec.

and maximum $F_{im} = \frac{12 \times 160 \times (8.06)^2}{32.2 \times 10.5} \quad (1.143)$

$= 422$ lb.

6.5.6 Calculating Motor-Gear-Box Output Torque

Calculations for the maximum motor-gear-box output torque required to drive the crank at various RPMS are presented in Table IV and were carried out as previously described in section 5.3.7. These values of torque were computed for full crank throw of 10.5" and a 25" depth of water. The results are plotted in Fig. 39.

6.5.7 Discussion of Torque Requirements

Fig. 39 shows a rapidly increasing torque requirement at high crank speeds. When generating deep-water waves, wave steepness $\frac{H}{L}$ limits the wave heights and hence the useful crank throw begins decreasing with decreasing wave period. Therefore, at high crank speeds the power required to actually generate waves of maximum height is considerably less than that needed to run the wave generator at maximum crank throw which will only give splash and breaking waves.

The peak torque demand at which waves are produced, occurs when the waves being produced have a length to depth ratio of $\frac{L}{d} = 4$ which is obtained in a 25" depth of water at a crank speed of 45.1 RPM. It is obvious, then, that the power requirements can be considerably reduced, without reducing the wave making capability, if the crank throw at high RPM is limited.

Hinged-flap wave generators most often do not require a crank throw limitation unless the maximum speeds are very high. This is due to the lower water inertia forces and lower mechanism inertia. When the paddle moves with a piston motion and deep-water waves are to be developed, economics dictate a limitation of the torque demand at high speeds.

There are a number of ways of limiting the high speed torque demand. The operator of the piston-type wave generator at the University of Washington is required to not exceed a certain crank throw at higher speeds. Torque limiters or motor-current limiters can be used, but at added expense. NRC describes its approach to this problem, for a hinged-flap generator, in their report on the

CRANK RPM	L (inches)	$\frac{L}{d}$ for d = 25"	$C = \sqrt{\frac{gL}{2\pi} \cdot \tanh \frac{2\pi d}{L}}$ ft./sec.	$T = \frac{L}{C}$ (sec.)	F_n (lb.) : This is max. value read from Fig. 31.	F_i (lb.) : This is max. value read from Fig. 32	edw = $\frac{10.5}{12} \times \frac{25}{12} \times \frac{30}{12}$ (Units in feet)	Max. F_n (lb.)	Max. F_i (lb.)	$\cos\theta + \frac{R}{\ell}\cos 2\theta$ Kent Handbook -Table 2 p.7-38 Max. value for $R/\ell = 1/7$	CRANK VELOCITY v_c (ft./sec.)	Estimated **Paddle Wt. W (lb.)**	Max. Fim $= \frac{12WV_c^2}{g\,R}(\cos\theta+\frac{R}{\ell}\cos 2\theta)$ lb.
135.5	12	.48	2.26	.443	19.0 edw	720 edw	4.56	87	3280	1.143	12.41	160	1000
105	20	.80	2.92	.571	31.6 edw	332 edw	"	144	1514	"	9.63	"	602
88	28.7	1.15	3.50	.683	44.5 edw	168 edw	"	203	766	"	8.06	"	422
66.5	50	2	4.62	.902	73.4 edw	52.8 edw	"	334	241	"	6.09	"	240
53.4	75	3	5.57	1.122	97.2 edw	6.0 edw	"	443	27.4	"	4.89	"	155
45.1	100	4	6.26	1.332	105.2 edw	4.4 edw	"	480	20.2	"	4.13	"	111

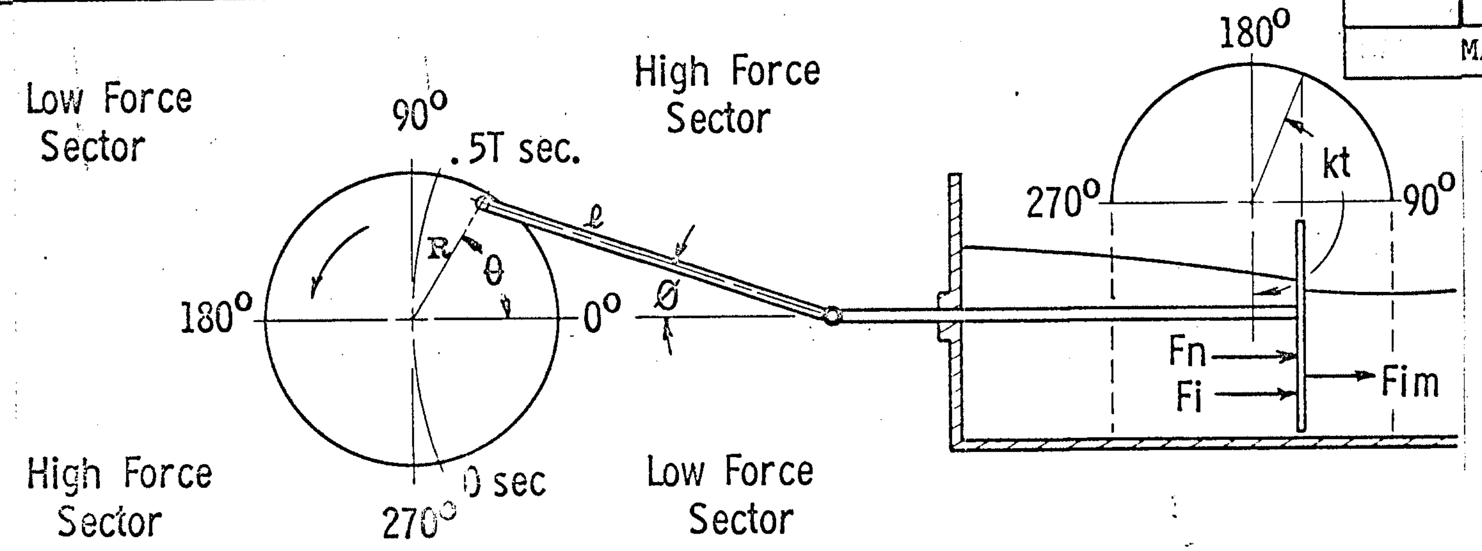
TABLE III. Calculation of Maximum Paddle Forces at Various Crank RPM for the Large Flume Wave Generator for Piston Motion, Crank Throw of 10.5" and Water Depth of 25".

CRANK ANGLE θ (degrees)	$(180^o - Kt^o)$ NOMINAL VALUE (degrees)	TABLE VALUE FOR $\frac{\ell}{R} = \frac{1}{7}$ * KENT: TABLE 1, P.7-04	.500 - TABLE VALUE	SIN(180-Kt)= $\frac{.5 - \text{TABLE VALUE}}{.5}$	$(180^o - Kt^o)$ ACTUAL VALUE FOR $\frac{\ell}{R} = \frac{1}{7}$ (degrees)	COS (180 - Kt)	$\frac{\ell}{R} = \frac{1}{7}$: SEC φ SIN (θ + φ) * KENT: TABLE 1, P.7-36	$\frac{\ell}{R} = \frac{1}{7}$: COS θ + $\frac{R}{\ell}$ COS 2θ * KENT: TABLE 2, P.7-38
0	(90)	0	.500	1.000	90	0	0	1.143
10	(80)	.009	.491	.982	79.1	-.189	.198	1.119
20	(70)	.034	.466	.932	68.8	-.362	.388	1.049
30	(60)	.076	.424	.848	58.0	-.530	.562	.938
40	(50)	.132	.368	.736	47.4	-.677	.713	.791
50	(40)	.200	.300	.600	36.9	-.800	.837	.618
60	(30)	.277	.223	.446	26.5	-.895	.928	.429
70	(20)	.361	.139	.278	16.1	-.961	.986	.233
80	(10)	.448	.052	.104	5.90	-.995	1.009	.039
90	(0)	.536	-.036	-.072	-4.10	-.998	1.000	-.143

CRANK ANGLE θ (degrees)	F_n COS (180 - Kt) (MAX. F_n = 480 lb.)	45.1 RPM: F_1 SIN (180-Kt) (MAX. F_1 = -20.2 lb.)	45.1 RPM: $F_{1m} = \frac{12WV_c^2}{gR}(\text{COS } \theta + \frac{R}{\ell}\text{COS } 2\theta)$ (MAX. F_{1m} = -111 lb.)	45.1 RPM: $P = F_n + F_1 + F_{1m}$	45.1 RPM: F_t = P SEC φ SIN (θ + φ) lb.	53.4 RPM: F_n COS (180-Kt) (MAX. F_n = 443 lb.)	53.4 RPM: F_1 SIN (180-Kt) (MAX. F_1 = -27.4 lb.)	53.4 RPM: $F_{1m} = \frac{12WV_c^2}{gR}(\text{COS } \theta + \frac{R}{\ell}\text{COS } 2\theta)$ (MAX. F_{1m} = -155 lb.)	53.4 RPM: $P = F_n + F_1 + F_{1m}$	53.4 RPM: F_t = P SEC φ SIN (θ + φ) lb.
0	0	-20	-111	-131	0					
10	-91	-20	-109	-220	-44					
20	-174	-19	-102	-295	-115					
30	-254	-17	-91	-362	-203	-235	-23	-127	-385	-216
40	-323	-15	-77	-415	-296	-300	-20	-107	-427	-305
50	-380	-12	-60	-452	-378	-354	-16	-84	-454	-380
60	-429	- 9	-42	-480	-445	-396	-12	-58	-466	-433
70	-462	- 6	-23	-491	-484	-425	- 8	-32	-465	-458
80	-478	- 2	- 4	-484	-488	-440	- 3	- 5	-448	-453
90	-479	+1.4	+16	-462	-462					
	MAX. TORQUE=488X10.5=5120 in.lb					MAX. TORQUE = 4820 in. lb.				

CRANK ANGLE θ (degrees)	66.5 RPM: F_n COS (180 - Kt) (MAX. F_n = 334 lb.)	66.5 RPM: F_1 SIN (180 - Kt) (MAX. F_1 = -241 lb.)	66.5 RPM: $F_{1m} = \frac{12WV_c^2}{gR}(\text{COS } \theta + \frac{R}{\ell}\text{COS } 2\theta)$ (MAX. F_{1m} = -240 lb.)	66.5 RPM: $P = F_n + F_1 + F_{1m}$	66.5 RPM: F_t = P SEC φ SIN (θ + φ) lb	88 RPM: F_n COS (180 - Kt) (MAX. F_n = 203 lb.)	88 RPM: F_1 SIN (180 - Kt) (MAX. F_1 = -766 lb.)	88 RPM: $F_{1m} = \frac{12WV_c^2}{gR}(\text{COS } \theta + \frac{R}{\ell}\text{COS } 2\theta)$ (MAX. F_{1m} = -422 lb.)	88 RPM: $P = F_n + F_1 + F_{1m}$	88 RPM: F_t = P SEC φ SIN (θ + φ) lb
0						0	-766	-422	-1180	0
10						-39	-753	-412	-1204	-239
20	-121	-224	-220	-565	-219	-74	-714	-388	-1176	-456
30	-177	-204	-197	-578	-325	-108	-650	-346	-1104	-620
40	-226	-177	-166	-569	-405	-138	-564	-292	-994	-707
50	-367	-145	-130	-542	-454	-162	-460	-228	-850	-711
60	-298	-107	-90	-495	-459	-182	-342	-158	-682	-633
70	-321	-67	-49	-437	-430	-195	-213	-86	-494	-487
80	MAX. TORQUE = 4830 in. lb.					MAX. TORQUE = 7460 in. lb.				

CRANK ANGLE θ (degrees)	105 RPM: F_n COS (180 - Kt) (MAX. F_n = 144 lb.)	105 RPM: F_1 SIN (180 - Kt) (MAX. F_1 = -1514 lb.)	105 RPM: $F_{1m} = \frac{12WV_c^2}{gR}(\text{COS } \theta + \frac{R}{\ell}\text{COS } 2\theta)$ (MAX. F_{1m} = -602 lb.)	105 RPM: $P = F_n + F_1 + F_{1m}$	105 RPM: F_t = P SEC φ SIN (θ + φ) lb	135.5 RPM: F_n COS (180 - Kt) (MAX. F_n = 87 lb.)	135.5 RPM: F_1 SIN (180 - Kt) (MAX. F_1 = -3280 lb.)	135.5 RPM: $F_{1m} = \frac{12WV_c^2}{gR}(\text{COS } \theta + \frac{R}{\ell}\text{COS } 2\theta)$ (MAX. F_{1m} = - 1000 lb.)	135.5 RPM: $P = F_n + F_1 + F_{1m}$	135.5 RPM: F_t = P SEC φ SIN (θ + φ) lb
0										
10										
20	-52	-1410	-552	-2014	-780	-32	-3060	-918	-4010	-1556
30	-76	-1284	-493	-1853	-1042	-46	-2780	-820	-3646	-2050
40	-98	-1113	-416	-1627	-1158	-59	-2410	-694	-3163	-2250
50	-115	-908	-326	-1349	-1128	-70	-1968	-541	-2597	-2180
60	-129	-675	-226	-1030	-955	-78	-1462	-375	-1915	-1775
70	-138	-422	-123	-683	-673					
80	MAX. TORQUE = 12,180 in. lb.									
90						MAX. TORQUE = 23,600 inlb				



* Carmichail, Colin - KENT'S MECHANICAL ENGINEERS' HANDBOOK, Design and Production Volume, 12TH Edition, John Wiley and Sons, New York, 1950.

TABLE IV. Calculations for Maximum Motor-Gear-box Output Torque Required by the Large Flume Wave Generator for Piston Motion, Maximum Crank Throw of 10.5" and Water Depth of 25".

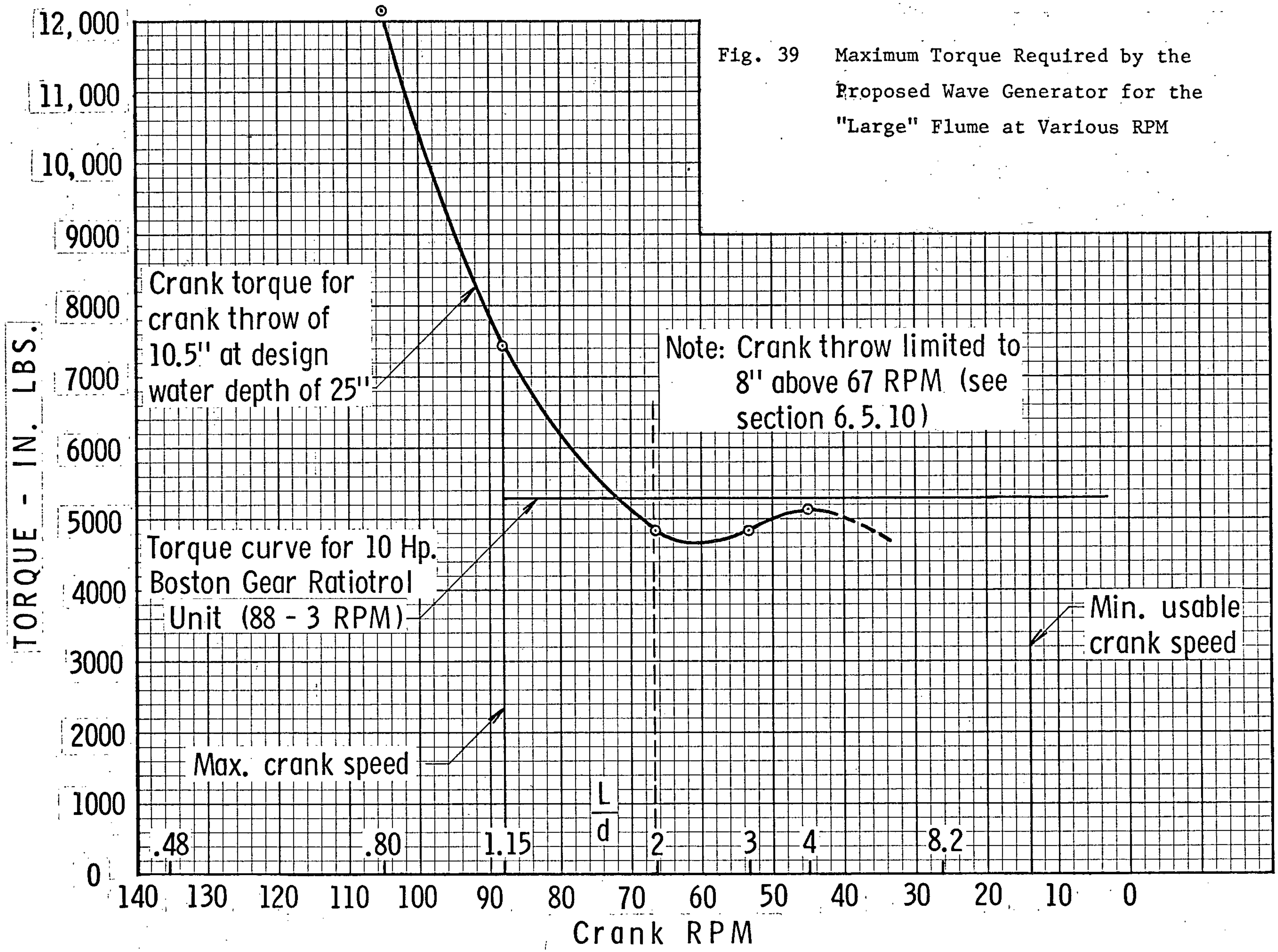


Fig. 39 Maximum Torque Required by the Proposed Wave Generator for the "Large" Flume at Various RPM

"Design of a Wave Generator for the Hydraulics Laboratory" (Ref. 7 - Wave Generators).

It was decided that the drive unit must provide at least 5120 in.lb. of torque at 45 RPM and that the crank throw would be limited for high speed operation.

The crank throw and water depth used in the calculations presented in Tables III and IV, were only decided upon after numerous other trial calculations had been made. The results of these other calculations are summarized in Table V, which lists the maximum torque requirements for generating a wave having $\frac{L}{d} = 4$, using different crank throws and water depths.

6.5.8 Selection of a Drive System

The selection of a variable-speed drive system was narrowed down to a choice between either a variable-speed, DC motor system or a variable-speed, mechanical system. After considering speed regulation, costs and compactness of the final installation, it was decided to use a variable-speed, DC motor, "Ratiotrol" system, produced by the Boston Gear Co.

The Ratiotrol system chosen is a 10-Hp unit delivering a constant torque of 5290 in.lb. over a speed range of 88 - 3 RPM. The available torque exceeds the maximum required torque of 5120 in.lb. Addition of the optional tachometer feedback circuit will give a 1% speed regulation and reduce the effects of changing line

PISTON MOTION			
Crank Throw (inches)	Still Water Depth d (inches)	Max. Crank Torque (in.lbs.)	Crank RPM for Wave Having L/d = 4
9.5	22	3750	48
	23	3960	-
	24	4120	-
	25	4200	45.1
10	22	4140	48
	25	4650	45.1
10.5	22	4540	48
	25	5120	45.1
11	22	5040	48
	25	5640	45.1

Note; For piston motion the paddle displacement (e) equals the crank throw.

TABLE V - Summary of Maximum Torque Requirements for the 39'-4½" x 30" x 36" Flume Wave Generator for Generating a Wave Having L/d = 4 Using Piston Motion and Different Limits of Crank Throw and Water Depth.

voltage.

The maximum speed of 88 RPM will give a true, deep-water wave in 25" of water with a length of depth ratio of $\frac{L}{d} = 1.15$. This wave has a period of 0.68 sec. which is still shorter than Galvin's preferred limit of 0.75 sec. The minimum speed of 3 RPM exceeds the minimum usable speed of 14 RPM.

Specifications for ordering the recommended drive unit are given in Appendix A.

6.5.9 Estimated Wave Height Capability

The maximum wave height capability of the wave generator will be obtained using piston motion for waves of all periods and a maximum water depth of 25". Theoretical values of these wave heights were computed using equation (27), rearranged to give $H = 2a = 2Ke$, and plotted in Fig. 40. Similarly, wave heights for hinged-flap motion were calculated using the equation $H = 2a = 2K'2$. Values of K and K′ were read from Fig. 19. For hinged-flap motion, the value of flap displacement (e) at the still water level for full crank throw, was taken as $e = \frac{25}{48} \times 10.5 = 5.47"$, where 48" is the height of the connecting rod attachment to the paddle, measured above the flume floor.

For normal paddle operation (i.e. intermediate motion) the wave heights obtained will lie between those for piston motion and those for hinged-flap motion.

Galvin's observed wave height limitations due to depth, were plotted along with the height limitation for deep-water waves due to steepness ($\frac{H}{L} = \frac{1}{7}$). Although Biésel's theories are mostly unproven, some research work has been done by Galvin concerning the wave height factor K´ (Ref. 3 and 4 - Wave Generator Theory). Using the 635-ft. long, 15 ft. wide, 20 ft. deep, concrete wave channel at the U.S. Army, Coastal Engineering Research Centre in Washington, D.C., he found that wave heights obtained for hinged-flap motion were within 90% of the theoretical height. This lends validity to Biésel's work and to his statement that wave heights for smaller sized wave generators will be about 70% of the theoretical height, depending on the size of the paddle, the wave period, and the amount of water leakage past the paddle. Values of wave heights for 70% efficiency were computed and plotted in Fig. 40.

6.5.10 <u>High Speed, Crank Throw, Limitation</u>

From Fig. 39 it is evident that the crank torque at speeds above 72 RPM must be limited to protect the electric motor against overload. It was decided to accomplish this by limiting the crank throw. In Fig. 40 it is apparent that, even for hinged-flap motion, the crank throw required to generate a wave of maximum height having $\frac{L}{d} = 2$, is small. The actual value can be obtained from the equation

$$H = 2K'e$$

where $e = \frac{25}{48} \times$ crank throw. Solving gives a crank throw

$$= \frac{H}{2K'} \times \frac{48}{25}$$

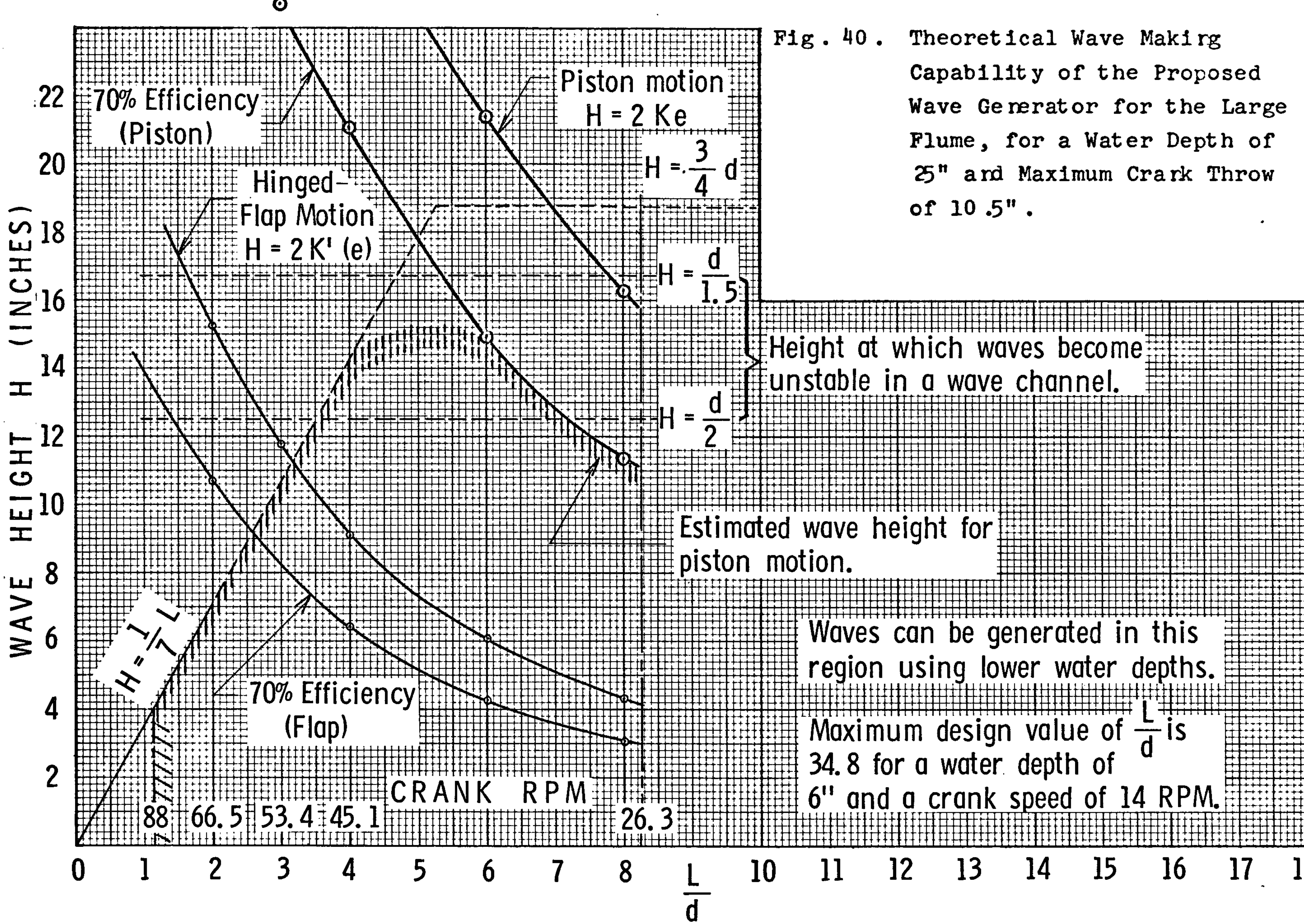


Fig. 40. Theoretical Wave Making Capability of the Proposed Wave Generator for the Large Flume, for a Water Depth of 25" and Maximum Crank Throw of 10.5".

$$= \frac{7.14}{2x1.4} \times \frac{48}{25}$$

$$= 4.9 \text{ in.}$$

A wave having $\frac{L}{d} = 2$ is produced in 25" of water at a crank speed of 66.5 RPM (Fig. 39). At full crank throw of 10.5" the required crank torque is 4830 in.lbs. which is less than the 5290 in. lbs. capacity of the drive system. Since, as just shown, a crank throw of 4.9 in. is all that is required to get maximum wave height, even with hinged-flap operation, this speed appeared as a good choice to start a crank throw restriction. If the crank throw is limited to a maximum of 8", at a crank speed above 67 RPM, the crank torque at the maximum speed of 88 RPM rises to only 4350 in.lbs. giving full protection against overload. It is evident that the wave-making capability of the wave generator is in no way restricted.

If the operator forgets to use this restriction, the drive system fuzing should protect the electric motor and control unit against overload damage. In any case, the maximum possible overload is $\frac{7460}{5290} \times 100 = 141\%$ at 88 RPM which is normally not serious for an electrical system.

When the wave generator is built, both the speed control unit and the crank should have the following warning attached:

"Maximum crank throw limited to 8" at speeds above 67 RPM "

6.6 Design of Paddle Mechanism Geometry

6.6.1 Basic Mechanism Geometry

With reference to Fig. 41 length of the arms AB and CD was arbitrarily set at 51.000". Increasing the length of these arms reduces the amount of change in the opening between the end of the paddle and the flume bottom as the paddle oscillates, but has the disadvantage of increasing the paddle mass and inertia.

Considering the size of the crank disc and the method of attaching the connecting rod to the paddle, the attachment position G was set 3.00" from A.

The length of EC was set at 22.00". Increasing this length increases the separation between arm AB and CD making the paddle motion less sensitive to adjustments of point D along the arc BD. The length of EC is limited by the necessity of having to clear the flume top edge when point C is at its extreme position C´.

When the paddle is in its mean position, point C must lie on the arc of point A and therefore AE may be computed using the formula shown in Fig. 41, which gives $EC^2 = AE\ (2AB - AE)$

i.e. $$AE^2 = 2AE.AB + EC^2 = 0$$

$$AE^2 = 102.000AE + (22.000)^2 = 0$$

and $$AE = \frac{+\ 102.000 \pm \sqrt{102.000 - 4\ (22.000)}^{\,2}}{2}$$

The required solution is AE = 4.989"

The length of AF is limited by the space available when the paddle is in the extreme position for piston motion. Using the same procedure as for finding AE, the length of AF must be shorter than AB by the amount

$$h^2 - 102.000\,h + 10.50^2 = 0$$

$$h = 1.09"$$

To prevent binding when sand is present in the water, allow an operating clearance of 0.11". Therefore,

$$AF = 51.00 - 1.09 - 0.11 = 49.80"$$

6.2.2 Paddle End-Clearance Plate

The paddle will clear the bottom of the flume by 1.09 + 0.11 = 1.20" when it is in its mean position. This space will permit considerable water leakage past the paddle with a loss in wave height. To reduce this leakage, use of an end-clearance plate is recommended, the profile of which is shown in Fig. 41.

The profile of this plate can be safely approximated by two arcs, representing the extremes of position of the end of the paddle F. throughout its complete range of possible motion. To maintain operating clearance, the length of AF is taken as 49.91" rather than 49.80" when drawing this profile. The first arc is F'F" with a radius of 1.09", centre B and running from the mean paddle position F' to the extreme position F" for hinged-flap motion. The second arc is F"F with a radius of 49.91 - 3.00 =46.91",

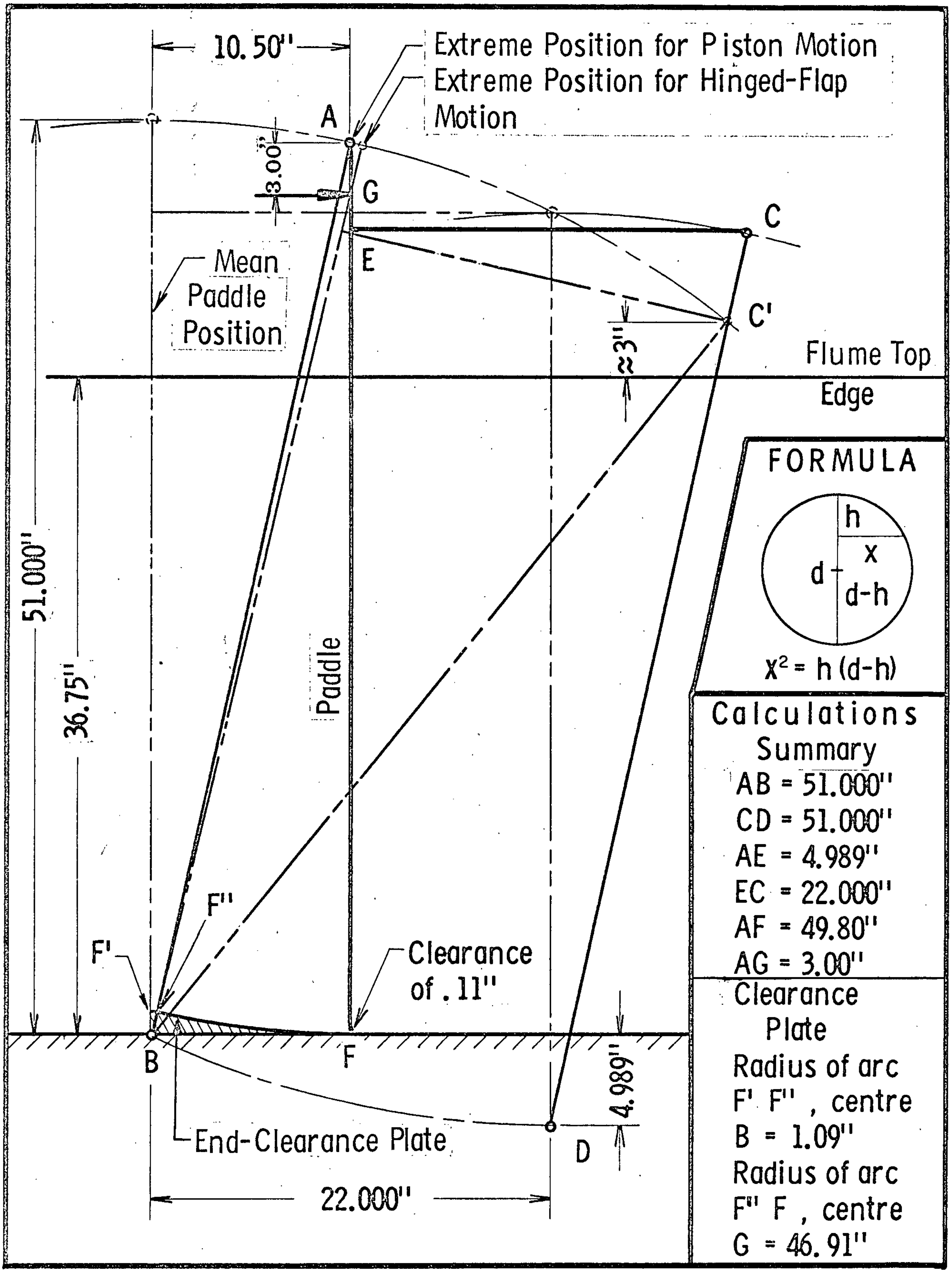


Fig. 41. Developing the Geometry of the Paddle Mechanism

centre G and running from F", the extreme of hinged-flap motion to F, the extreme of piston motion. This profile is mirrored in the second half of the paddle oscillation.

Although an end-clearance plate with the curved profile shown, would be more efficient, the curves may present manufacturing difficulties, in which case a wedge shape could be used. In this case the side of the wedge would be a straight line from F' tangent to the arc F"F.

The end-clearance plate should be made of mild steel and attached to the flume floor with an epoxy resin glue, to avoid distortion of the flume floor from the heat of welding.

6.6.3 Paddle Structural Geometry

A side view of the paddle structural geometry is shown in Fig. 42 along with basic dimensions.

6.7 Wave Generator Structural Design Loads

6.7.1 General

The design loads for various parts of the wave generator mechanism are presented in this section. The loads, as given, do not include load impact factor, due to the reciprocating motion, or any safety factor. The actual selection of members to carry these loads is not detailed in this thesis, but the selected members are specified in the accompanying drawings (Appendix D).

To keep the inertia of the oscillation members low, the connecting rod, paddle assembly and paddle support arms, were designed for construction using aluminium alloy 6061 - T6 (Alcan No.65S-T6). Other parts were designed for construction using mild steel.

In selecting structural components, rigidity was the guiding factor, and therefore, some of the chosen components are more than adequate in strength in order to ensure a minimum deflection.

6.7.2 Crank and Crankpin Design Loads

The overhung crank was designed in accordance with procedures outlined in Kent's Mechanical Engineer's Handbook, page 7-34. The maximum bending moment when crank and connecting rod are at right angles, was taken as 150% of the output torque of the drive system, which is 5290 x 1.50 = 7940 in.lbs. The greatest bending moment on the crank, due to overhand of the crankpin and occurring when the crank is on dead centre, results from a load of 1188 lbs. at 88 RPM.

Maximum bending moment on the overhung crankpin is 1.10 x 1204 = 1325 in.lbs. and occurs at 88 RPM. Crankpin proportions were detailed as given in Kent, P.7-35.

6.7.3 Connecting Rod Design Load

The maximum load in pounds, acting along the centre line of the connecting rod, was obtained from the equation

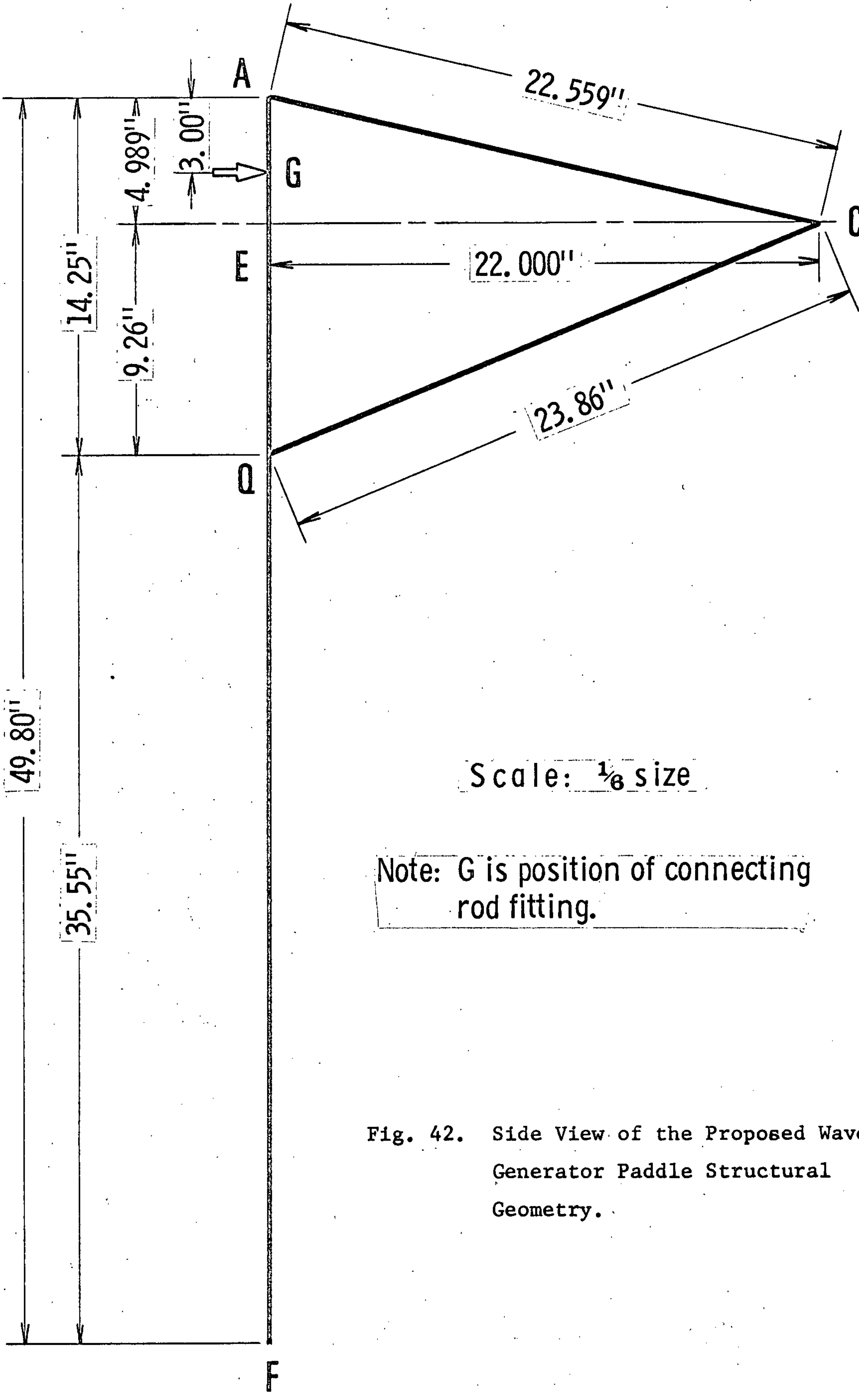


Fig. 42. Side View of the Proposed Wave Generator Paddle Structural Geometry.

$$F_c = \frac{P}{\sqrt{1 - \left(\frac{R}{\ell} \sin \theta\right)^2}}$$

given in Kent's Mechanical Engineer's Handbook P.7-37. Using values of P and θ from Table IV, the maximum sustained value of F_c when generating waves, was found to occur for P = 491 lbs. and $\theta = 70^\circ$, at a crank speed of 45.1 RPM. The value obtained was F_c = 495 lbs. acting alternately as a tensile and then compressive load.

Should the crank throw limitation be ignored and the wave generator run at 88 RPM using full crank throw and a water depth of 25", the peak motor overload would be $\frac{7460}{5290} \times 100 = 141\%$. Mostly splash and breaking waves would be generated and it is assumed that the operator would realize his error and shut down. Should he not shut down, the motor would probably run for some time before overheating caused the fuzing to shut it off, since the 141% overload occurs during only part of each paddle cycle. Under such conditions, the connecting rod load F_c = 1204 lbs. for P = 1204 lbs. and $\theta = 10^\circ$ at a crank speed of 88 RPM.

It was decided to design the connecting rod for an axial load in compression of 1204 lbs.

A short 6.62"-long rod, used to transmit the load from the end of the connecting rod to the frame of the flap, was designed in aluminium to transmit a load of 1204 lbs. The large diameter was based on the need to limit shaft deflection to a maximum of .010" per foot, between bearings. Although the specified bearings carrying the rod are self-aligning, in that they align with the shaft, they are not of a type designed to handle a continually varying misalighment.

6.7.4 Paddle Design Loads

Using the logic employed in determining the connecting rod design load (Section 6.7.3.), the paddle was designed to operate at 88 RPM at full crank throw in water 25" deep.

The water forces acting on the paddle are the wave forces (F_n) and the water inertia forces (F_i). The point of application of F_n on the paddle, measured from the flume bottom, was determined by finding the height of the centre of gravity of the area of the wave pressure (P_n) diagram using equation (29) as follows:

$$\bar{y}_n \div {}_0\int^d \rho \text{ gke } \frac{\cosh my}{\cosh md} \cos kt \,(dy) = {}_0\int^d \rho \text{ gke } \frac{\cosh my}{\cosh md} \cos kt.y(dy)$$

$$\bar{y}_n = \frac{{}_0\int^d y \cosh my.\, dy}{{}_0\int^d \cosh my.\, dy}$$

$$= \frac{\left(\frac{y}{m} \sinh my\right)_0^d - {}_0\int^d \frac{1}{m} \sinh my.\, dy}{\frac{1}{m} (\sinh my)_0^d}$$

$$= \frac{d \sinh md - \frac{1}{m} (\cosh md - 1)}{\sinh md}$$

$$= d - \frac{1}{m} \left(\frac{\cosh md - 1}{\sinh md} \right)$$

$$= (d - \frac{1}{m} \tanh \frac{md}{2})$$

and for d = 25", $m = \frac{2\pi}{L}$, L = 28.7" and $\frac{L}{d} = 1.15$ (Table III)

$$\bar{y}_n = (25 - \frac{28.7}{2} \tanh \frac{2\ \pi}{2 \times 1.15})$$

$$= 25 - 4.54$$

$$= 20.46''$$

Similarly the point of application of F_i, measured from the flume bottom, was found using equation (30) as follows:

$$P_i = \rho g \ \Sigma_{n=1}^{\infty} C_n \tan m_n d . \cos m_n y \ . \sin kt$$

$$\bar{y}_i \ {}_o\int^d p_i \ dy = {}_o\int^d p_i \ y.(dy)$$

$$\bar{y}_i = \frac{{}_o\int^d p_i \ y.(dy)}{{}_o\int^d p_i . \ dy}$$

Now $_o\int^d p_i \ y. \ (dy) = \rho g \ {}_o\int^d y \ (\Sigma_{n=1}^{\infty} C_n \tan m_n d \sin kt \cos m_n y).dy$

$$= \rho g \sum_{n=1}^{\infty} C_n \tan m_n d \sin kt \int_o^d y \cos m_n y \,.\, dy$$

$$= \rho g \sum_{n=1}^{\infty} C_n \tan m_n d \sin kt \left\{ \left(\frac{y}{m_n} \sin m_n y \right)_o^d - \int_o^d \frac{\sin m_n y \,.\, dy}{m_n} \right\}$$

$$= \rho g \sum_{n=1}^{\infty} C_n \tan m_n d \sin kt \left\{ \frac{d \sin m_n d}{m_n} + \frac{1}{m_n^2} (\cos m_n d - 1) \right\}$$

and $\int_o^d p_i . dy = \rho g \sum_{n=1}^{\infty} C_n \tan m_n d \sin kt \dfrac{\sin m_n d}{m_n}$

yielding $\bar{y}_i = \dfrac{\sum_{n=1}^{\infty} C_n \dfrac{1}{m_n^2} \tan m_n d \;\; (m_n d \sin m_n d + \cos m_n d - 1)}{\sum_{n=1}^{\infty} C_n \tan m_n d \;\; \dfrac{\sin m_n d}{m_n}}$

where $C_n = \dfrac{2 e \sin m_n d}{\sin m_n d \cos m_n d + m_n d}$ from equation (31).

The series was evaluated for the first three terms using

d = 25" = 2.08',

e = 10.5" = .875', and converting $\frac{1}{m_n}$ into the form $\frac{d}{m_n d}$ where necessary:

From Fig. 18 for $\frac{L}{d} = 1.15$

$m_1 d = .64\pi = 2.01$ radians,

$m_2 d = 1.76\pi = 5.52$ radians,

$m_3 d = 2.82\pi = 8.85$ radians,

then $$C_1 = \frac{2e\ (.904)}{(.904)\ (-.427) + 2.01} = 1.116e = .976$$

$$C_2 = \frac{2e\ (-.689)}{(-.689)\ (.825) + 5.52} = -.279e = .244$$

$$C_3 = \frac{2e\ (.548)}{(.548)(-.837) + 8.85} = .1306e = .1142$$

and $$\bar{y}_i = \frac{.976\ (\frac{2.08}{2.01})^2\ (-2.12)\ \{2.01\ (.904) + (-.472) - 1\}}{.976\ (-2.12)\ \frac{2.08}{2.01}\ (.904)}$$

$$\frac{+ (-.244)\ (\frac{2.08}{5.52})^2\ (-.951)\ \{5.52\ (-.689) + .825 - 1\}}{+ (-.244)\ (-.951)\ \frac{2.08}{5.52}\ (-.689)}$$

$$\frac{+ .1142\ (\frac{2.08}{8.85})^2\ (-.655)\ \{8.85\ (.548) + (-.837) - 1\}}{+ .1142\ (-.655)\ \frac{2.08}{5.52}\ (.548)}$$

$$= \frac{-.764 - .1311 - .0124}{-1.932 - .0603 - .016}$$

$$= .452' = 5.4''$$

Forces due to mechanism inertia were considered to act 30" above the flume floor.

The situation for maximum paddle structural loading due to external forces, is shown in Fig. 43.

Maximum loading at 88 RPM occurs at 10^{o} of crank motion (Table IV). The values of external load are :

$$P = 1204 \text{ lbs.}$$

$$F_n = 39 \text{ lbs.}$$

$$F_i = 753 \text{ lbs.}$$

and

$$F_{i_m} = 412 \text{ lbs.}$$

At a crank angle of 10^{o}, the angle ϕ (Table IV) that the connecting rod makes with the horizontal, is very small and P (Fig.43) can be considered a horizontal force. Also in the first 10^{o} of crank rotation the paddle has moved only 0.091" and therefore the paddle was considered as being at the extreme position of crank throw, so that available measurements (Fig.41 and 42) could be used. The resulting errors are negligible. The paddle weight was considered to act through the front of the paddle, giving a small error on the safe side since it increased the load on the support arm set AB, which gave the design load used for all four support arms.

The loads acting on the paddle structure, for conditions shown in Fig.43, were computed as follows:

a) Load on DC.

Taking moments about point A, the load acting along DC was found to be 1740 lbs. Divided between the DC set of two support arms, the load per arm is 870 lbs. in tension.

b) Load on AB,

Taking moments about point C, the load acting along AB was found to be 1860 lbs. Divided between the AB set of two support arms, the load per arm is 930 lbs. in compression. All four support arms were designed to carry this load.

c) Load on QC.

Taking moments about point A, the load acting along QC was found to be 2990 lbs. Divided among 4 members, this would be 813 lbs. per member, in compression.

d) Load on AC.

The sum of the vertical forces at point C is zero. Therefore, the load acting along AC is 2460 lbs. Divided among 4 members, this would be 615 lbs. per member, in tension.

e) Bending moment in AF.

Using bending moment and shear diagrams, the maximum bending moment in AF was found to occur at point Q and to have a value of 25,700 in.lbs.

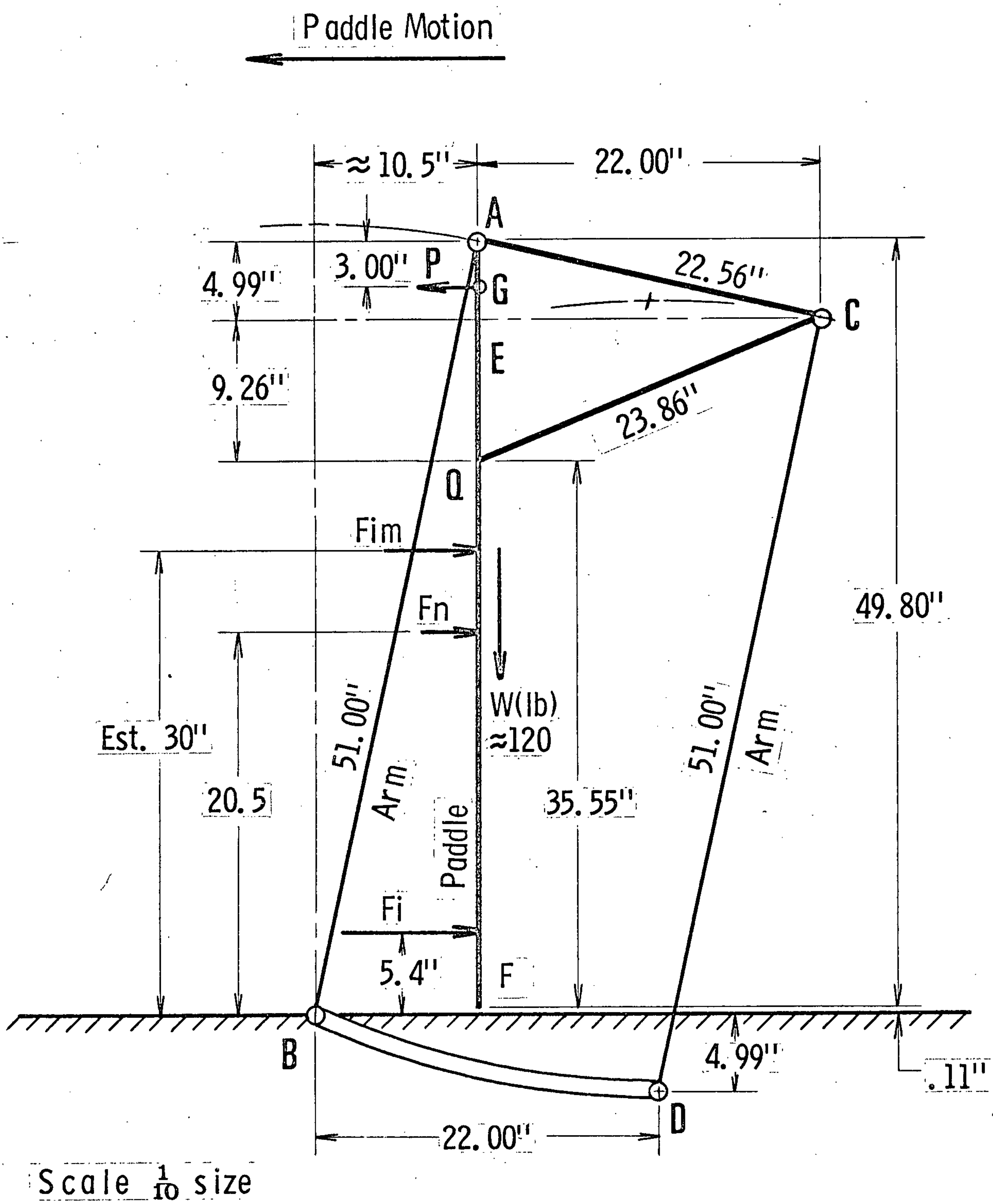


Fig. 43. Diagram of the Case for Maximum Loading by External Forces Acting on the Paddle of the Proposed New, Wave Generator.

6.7.5 Base Plate Reaction Loads

The two paddle support arms on each side of the wave generator are connected to base plates, one on each side of the flume, into which a slot BD (Fig.43) is cut. The horizontal separation between the arm ends B and D on the base plates is 22". Assuming a minimum base plate length of 30" between floor attachments, spaced equal distances on either side of B and D, the vertical reaction at the B end of the base plate was estimated to be 678 lbs. Therefore, each end of a base plate will have to be anchored to the floor with fasteners capable of resisting an uplifting force in the order of 680 lbs.

6.7.6 Other Component Design Loads

Loads in lateral members of the paddle assembly were computed as required when selecting components using the load data given.

6.7.7 Bearing Loads

Ball bearing units were designed for a minimum life of 2500 hrs., but some units exceed this, since use of large shaft diameters, to limit shaft deflection, required use of bearings of larger load capacities.

Ball bearing assemblies at either end of the connecting rod were designed for a radial load of 1204 lb.

The ball bearing assemblies at either end of the four support arms, were designed for radial loads of 930 lbs. each. The upper arm bearings are double-row ball bearings, selected to give the paddle mechanism lateral rigidity. The lower arm bearings are screw-on, self-aligning, hangar bearings. At 14 threads per inch, one half a turn of these bearing units will allow the support arms to be adjusted 0.036" in length, if assembly adjustments are required. The self-aligning bearings will avoid bending moments being applied to the support arms by any minor, base-plate misalignments.

6.8 Corrosion and Installation Problems

Information concerning corrosion protection is presented in Appendix B. Problems pertaining to the installation of this wave generator in the large flume are presented in Appendix C.

6.9 Design Drawings

Design drawings for the proposed, new wave generator for the 39'-4¼" long, 30" wide and 36" deep flume are included in Appendix D.

6.10 Summary of Operating Specifications

The summarized operating specifications for the proposed new wave generator are as follows:

maximum crank throw	= 10.5"
minimum crank throw	= 0"

usable wave period range	= 0.68 to 4.3 secs.
design water depth	= 25"
usable wave period range	
at design water depth	= 0.68 to 2.3 secs.
(equivalent crank speed range	= 88 to 26.3 RPM)
maximum usable wave period	
at 6" water depth	= 4.3 secs.
(equivalent crank speed	= 14 RPM)
minimum wave length	= 28.7"
maximum usable wave length	= 206"
$\frac{L}{d}$ range for a depth of 25"	= 1.15 to 8.24
$\frac{L}{d}$ maximum for a depth of 6"	= 34.8
estimated maximum wave height	
at design water depth	≃ 14"

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Appendix A

Variable-Speed Drive Unit for the Proposed New Wave Generator for the Large Flume

RPM range; 88 - 3

Recommended unit: 10 Hp. Boston Gear PE Series Ratiotrol System.

Reference: Boston Gear Catalogue No. 59, page 623.

Torque output: 5290 in. lb.

Motor: 10 Hp. foot mounted DC motor, catalogue No. 241001

Reductor: catalogue No.U 152 F - 20, page 286

Ratiotrol unit: PE series, catalogue No. PE 1000 equipped with "Tachometer Feedback", item 24, page 658

Weight of motor and reductor = 470 + 200 = 670 lbs.

Manufacturer: Boston Gear Works, Quincy 71, Mass., U.S.A.

Vancouver Distributor: Robert Morse Corp. Ltd.

Notes:

a) Tachometer feedback on the motor gives an improved speed regulation of 1% and minimizes the effects of changing line voltage.

b) The worm gear service factor for moderate shock up to 10 hrs./day is 1.2. The maximum equivalent torque requirement is 5290 x 1.2 = 6350 in.lb. in the speed range of 40 to 50 RPM which is within the reductors capacity.

Appendix B

Corrosion Protection for the Proposed New Wave Generator for the Large Flume

The aluminium alloy 6061 (65S-T6) has excellent resistance to atmospheric and fresh water corrosion, and fair resistance to sea water. It is not considered necessary to paint the aluminium parts of the paddle mechanism.

All fittings in direct contact with the aluminium should be either zinc-plated or cadmium-plated, or made of stainless steel. Cadmium-plated fasteners are readily available and are recommended.

The bearing assemblies unfortunately necessitate contact between aluminum and steel. To avoid crevice corrosion, it is recommended that the joints between the bearing units and the housing, and between bearing units and shafts, be sealed against the ingress of moisture using a polysulphide or butyl rubber sealant.

Where feasible, the steel components should be undercoated and painted to prevent rusting.

Appendix C

Problems Relative to the Installation of the Proposed New Wave Generator in the Large Flume

The installation of the wave generator in the 39'-4¼" long, 30" wide and 36" deep flume, will require attention to the following problems:

a) The flume interior in the region of the paddle installation has a level floor and true sides up as high as the horizontal angle running along the flume sides about 14" up from the flume floor. (Fig.37). Above this height, the sidewalls bulge slightly. It is recommended that these bulges be filled with a quality auto-filler and sanded true.

b) A cross member joins the two sides of the flume at the point of paddle installation. This must be removed and replaced by two cross members - one in front of and one behind the paddle.

c) The glass sides of the flume are not laterally braced at the top. Large waves create a fluctuating lateral pressure as they run down the flume. To avoid fluctuating changes in flume cross section, two clamp-on cross members should be made up and fitted. These members should be removable, so that they do not interfere with instrumentation.

d) The threaded gate rod and wheel which raise and lower the tail gate, must be moved out of line with the wave generator, connecting rod and crank.

e) Pipes running along one side of the flume must be removed to permit installation of the base plate supports.

f) The base plate supports require a firm foundation. The asphalt covering the floor must be removed at the points of support and the supports bedded on poured concrete pads with the anchor bolts firmly bedded in the concrete floor. Because of the reciprocating load, the reaction load of 680 lbs. at each support (Section 6.7.6) should be multiplied by an impact factor of 1.5 and each support made capable of resisting a vertical pull of 680 x 1.5 = 1020 lbs.

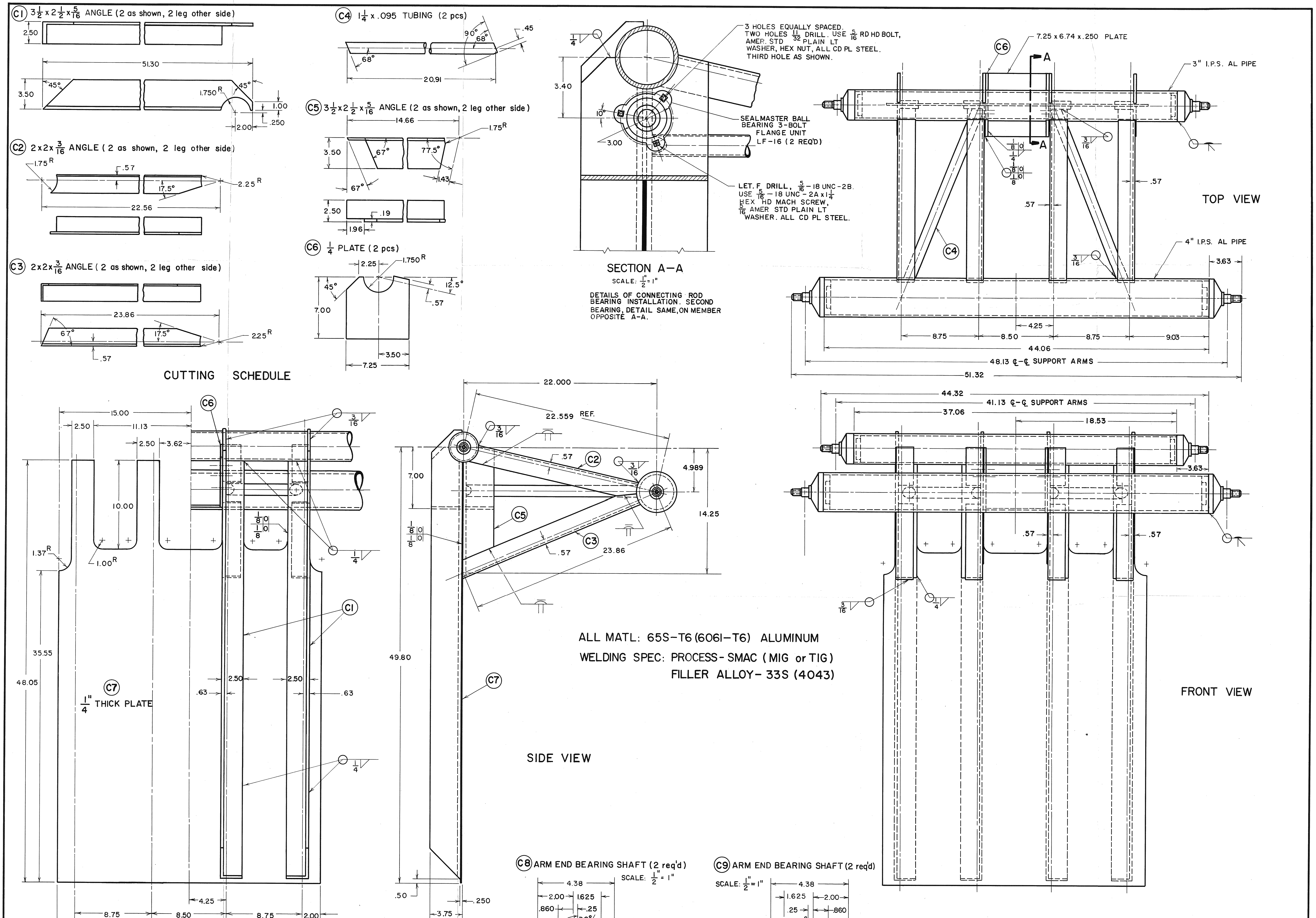
g) A mount will be required for the drive unit. It must be designed for an overturning force, allowing for impact, of 1204 x 1.5 = 1800 lbs. This force acts through the crank-wheel drive shaft, alternating fore and aft along a horizontal line parallel to the flume longitudinal axis. This drive-unit mount should also be bedded into the concrete floor slab.

h) The gate slots between the glass panels must be filled if a smooth wave profile is to be obtained.

Appendix D

Design Drawings for the Proposed New Wave Generator for the Large Flume

SHEET 1	Paddle
SHEET 2	Connecting Rod, Support Arms and Base Assembly
SHEET 3	Adjustable Crank
SHEET 4	Wave Generator Assembly and Installation Data



CIVIL ENGINEERING DEPARTMENT
UNIVERSITY OF BRITISH COLUMBIA

PADDLE
FOR 39' FLUME WAVE GENERATOR

DESIGNED BY: E. R. CHAPPELL	DATE: JULY 89
DRAWN BY: E. R. CHAPPELL	SCALE: 1/4" = 1"
REF: THESIS "THEORY AND DESIGN OF A WAVE GENERATOR FOR A SHORT FLUME"	SHEET NO. 1 OF 4

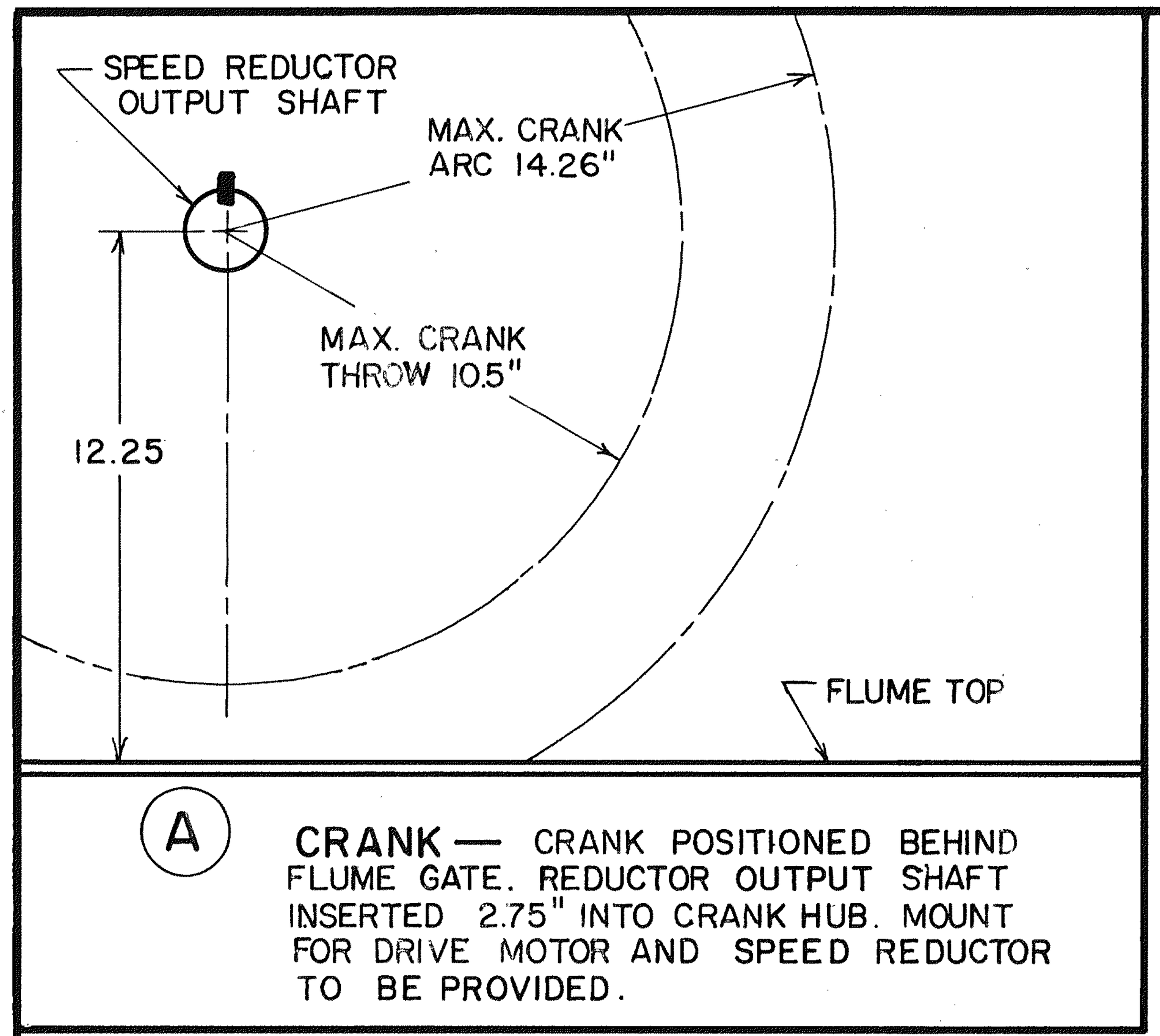

CRANK SIDE
$\frac{3}{4}$ -14 N.P.S.M. x .94
DRILL AND REAM FOR NO. 0 TP (STN STL) x 1" WITH CROSS-PIECE IN POSITION.
SEALMASTER HANGER BEARING SEHB-22, SHAFT DIAM. $1\frac{3}{8}$.
$1\frac{1}{2}$ I.P.S. PIPE
23° CHAMFER ALL AROUND
CONNECTING ROD
CROSS-PIECE
PRESS INTO END OF CONNECTING ROD.
SCALE: $\frac{1}{2}$" = 1"
MATL: 65S-T6 (6061-T6) ALUMINUM
WELDING: PROCESS SMAC (MIG or TIG)
FILLER ALLOY 33S (4043)
FACES SHAFT'S SUPPORTED SIDE
$\frac{3}{4}$ -14 NPSM x .94
SEALMASTER HANGER BEARING SEHB-16 SHAFT DIAM. 1"
2 x .083 ROUND TUBING
DRIVE FIT SKF DOUBLE ROW ANGULAR CONTACT BALL BEARING UNIT NO. 3205
SUPPORT ARM (4 req'd)
(E1) SUPPORT ARM LOWER AXLE (4 req'd)
SCALE: $\frac{1}{2}$" = 1"
45° CHAMFER TO CLEAR ANGLE FILLET
BASE PLATE (2 req'd, 2nd reversed)
SCALE: $\frac{1}{4}$" = 1"
LEVEL AND SET AT SAME ELEV. AS FLUME FLOOR SURFACE WHEN INSTALLED.
1.0000 / 1.0008 DIA.
SCALE
1" DRILL, 2 HOLES
1" MED PLAIN WASHER
ADJUSTABLE AXLE (2 req'd)
SCALE: $\frac{1}{2}$" = 1"
POSITION AXLE ON OTHER SIDE OF SECOND UNIT
1" - 12 UNF - 2A x $3\frac{3}{4}$ FIN HEX BOLT WITH FIN HEX NUT. CUT BOLT TO SHORTEN LENGTH TO 2.90. CHECK DIA. WHEN SELECTING. 2 REQ'D PER AXLE UNIT.
1.0000 / 1.0012, 2 HOLES
L 5 x $3\frac{1}{2}$ x $\frac{3}{8}$
DRILL HOLES FOR TIE-DOWN BOLTS (USE 4 / PLATE)
$\frac{5}{16}$ PLATE
36.40 INSTALLED
FLUME FLOOR
$\frac{3}{4}$ - 16 UNF - 2A x $1\frac{3}{4}$ FIN HEX BOLT TO HOLD BASE PLATE, $\frac{3}{4}$ MED PLAIN WASHERS UNDER HD AND NUT, HEX NUT (6 sets req'd)
SCALE: $\frac{1}{4}$" = 1"
BASE PLATE SUPPORTS (2 sets req'd, 2nd set reversed)
BASE ASSEMBLY
MATL: STEEL (1020–1025)
CIVIL ENGINEERING DEPT. UNIVERSITY OF BRITISH COLUMBIA
CONNECTING ROD, SUPPORT ARMS, and BASE ASSEMBLY
FOR 39' FLUME WAVE GENERATOR
DESIGNED BY: E. R. CHAPPELL
DATE: JULY 69
DRAWN BY: E. R. CHAPPELL
SCALE: AS SHOWN
REF: THESIS "THEORY AND DESIGN OF A WAVE GENERATOR FOR A SHORT FLUME"
SHEET NO. 2 OF 4

CIVIL ENGINEERING DEPARTMENT
UNIVERSITY OF BRITISH COLUMBIA

ADJUSTABLE CRANK

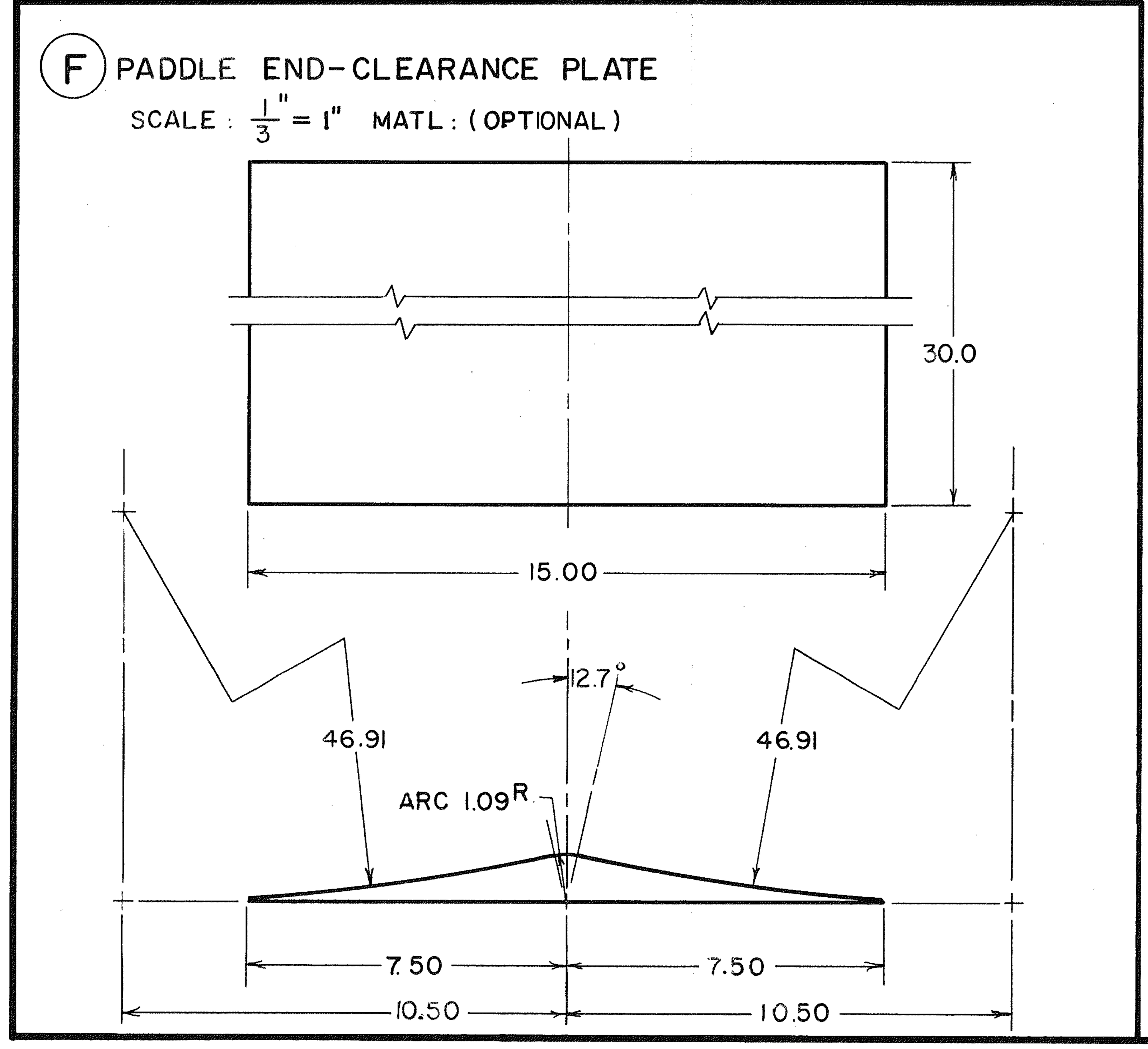
39' FLUME WAVE GENERATOR

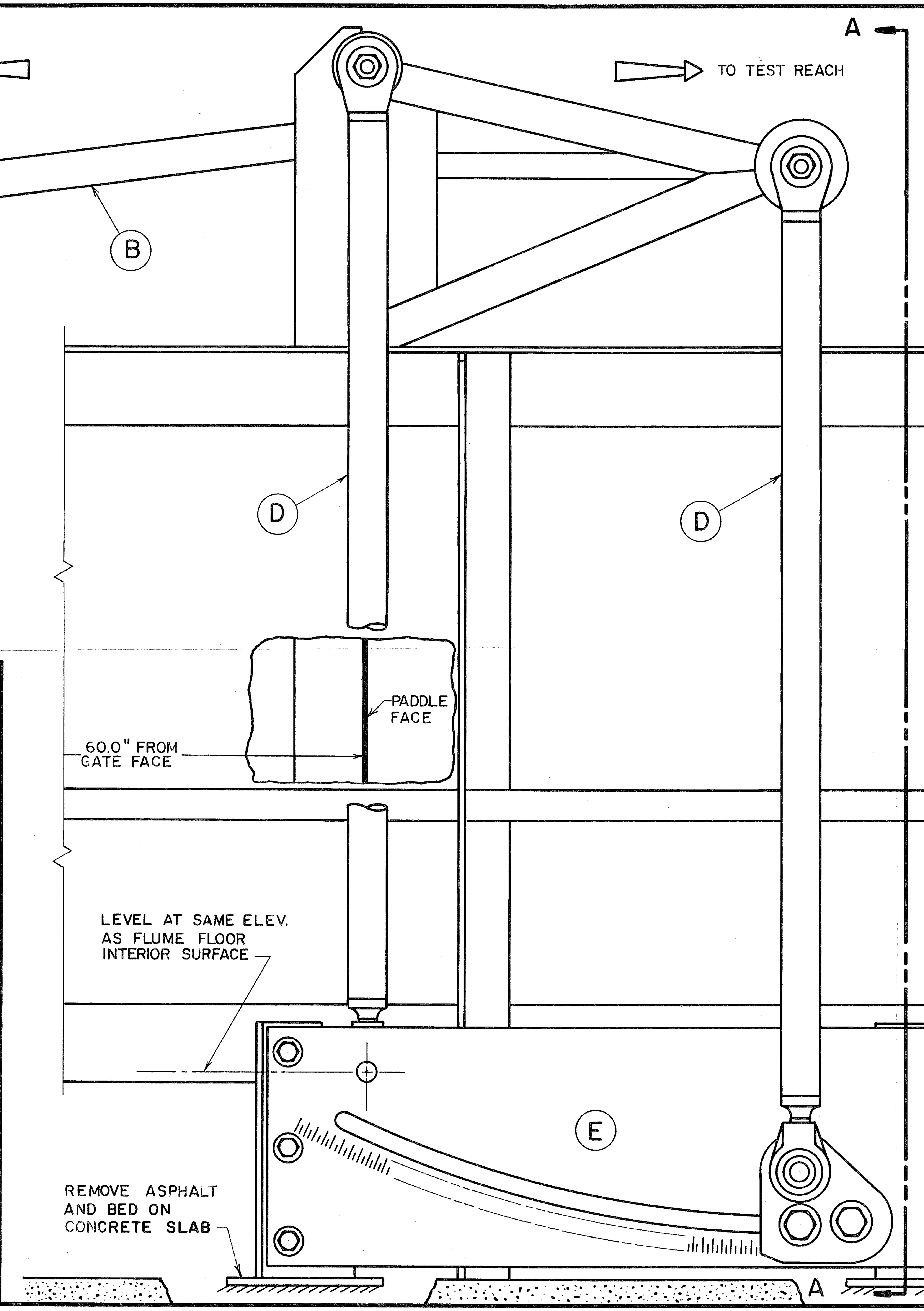
DESIGNED BY: E.R. CHAPPELL	DATE: JULY 69
DRAWN BY: E.R. CHAPPELL	SCALE: 1"=1", 1/2"=1"
REF: THESIS "THEORY AND DESIGN OF A WAVE GENERATOR FOR A SHORT FLUME"	SHEET NO. 3 OF 4

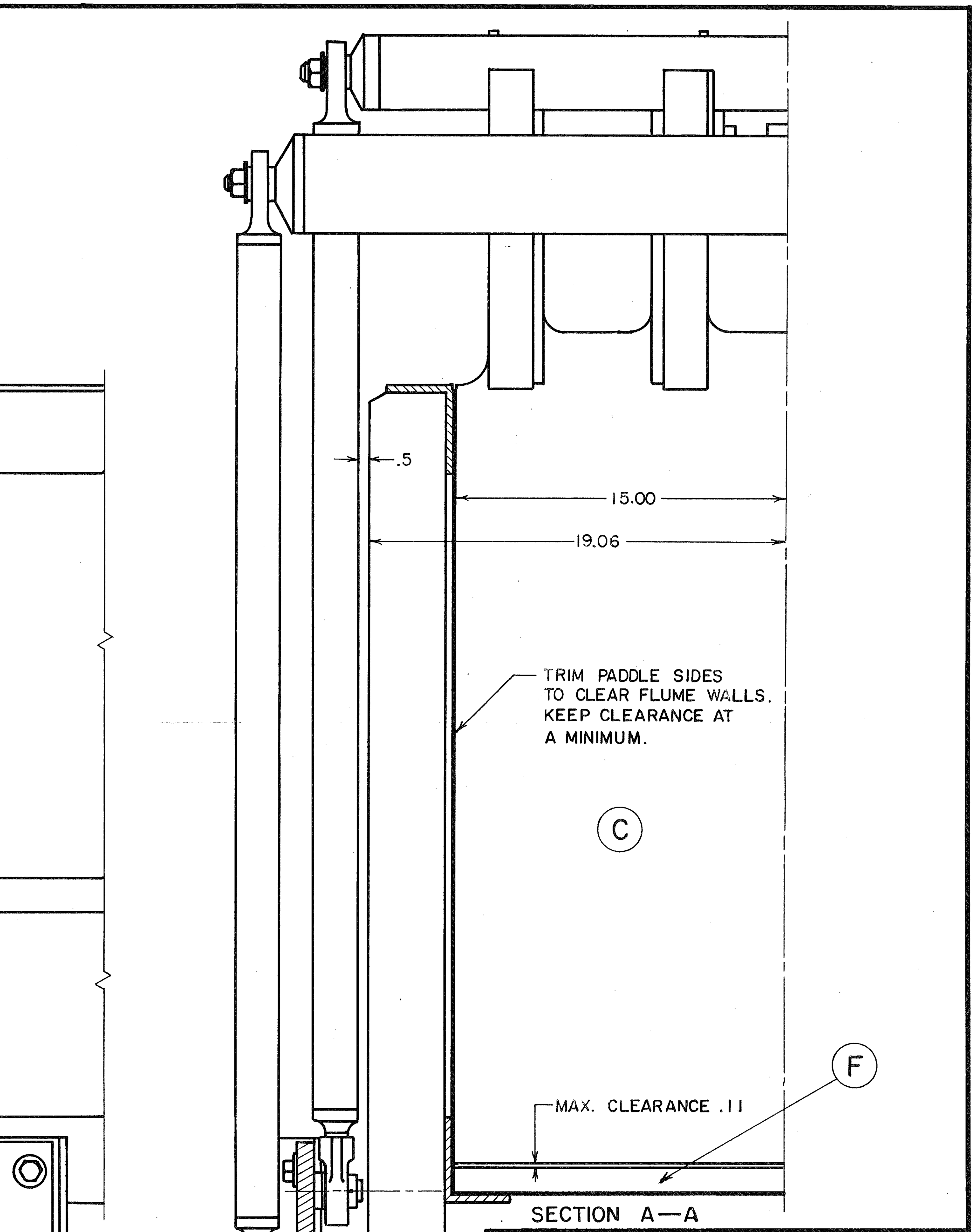


MAJOR COMPONENT LIST			
PC	NAME	MATL	QUAN
A	CRANK	STL	1
B	CONNECTING ROD	AL	1
C	PADDLE	AL	1
D	SUPPORT ARMS	AL	4
E	BASE ASSEMBLY	STL	2
F	END-CLEARANCE PLATE	OPT	1

SET SCREW TIGHTENING	
SCREW DIAM.	MIN. TORQUE INCH LBS
5/16	126
3/8	228
7/16	348
1/2	504







CIVIL ENGINEERING DEPARTMENT
UNIVERSITY OF BRITISH COLUMBIA

WAVE GENERATOR ASSEMBLY AND INSTALLATION DATA
FOR 39' FLUME WAVE GENERATOR

DESIGNED BY— E.R. CHAPPELL	DATE — JULY '69
DRAWN BY— E.R. CHAPPELL	SCALE: 1/4" = 1"
REF: THESIS "THEORY AND DESIGN OF A WAVE GENERATOR FOR A SHORT FLUME"	SHEET NO. 4 OF 4