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An Experimental Study and Engineering Evaluation of the Salter Cam Wave Energy Converter

MIT Sea Grant
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Massachusetts
Institute of Technology
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RELATED SEA GRANT REPORTS

Mei, Chiang C. NUMERICAL METHODS IN WATER-WAVE DIFFRACTION AND RADIATION. MITSG 78-17J. Cambridge: Massachusetts Institute of Technology, 1978. Journal reprint: Annual Review of Fluid Mechanics 1978, 10-393-416. $1.00.

Chen, H.S., Chiang C. Mei, Dick K.P. Vue. A HYBRID ELEMENT METHOD FOR CALCULATING THREE-DIMENSIONAL WATER WAVE SCATTERING. MITSG 76-10. NTIS: PB-262 040/AS. Cambridge: Massachusetts Institute of Technology, 1976. 222 pp. $5.00.

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ABSTRACT

An experimental study and an engineering evaluation of the Salter cam wave energy device has been conducted. The experiments were carried out with a model cam placed in a wave channel and in a towing tank. The tests in the towing tank were with a floating model to simulate ocean conditions. The experimental results indicated that the Salter cam was capable of extracting much of the energy in regular waves and confirmed Salter's findings.

From the engineering evaluation it was concluded that there are major unsolved problems associated mainly with the mooring system. The predicted cost of electricity brought to the shore was much higher than costs predicted for more conventional methods of electrical power production during the 1990's.

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1. INTRODUCTION

In 1973-1974 Salter invented a novel wave energy absorber which has been called the Salter Cam or Duck. The device was the result of considerable development effort by him on various types of wave energy absorbers. The cam shaped device in Salter's experimental facility was supported on a fixed axis with the lobe of the cam facing the waves, Figure 1.1. In regular waves it was observed that the cam rolled back-and-forth at the frequency of the waves. Furthermore, the experiments indicated that the device, when coupled to a suitable power absorber, was capable of extracting most of the power of the waves (greater than 90%) for a small range of frequencies. Hydrodynamic studies have indicated that the cam with a single degree of freedom (roll) can be modeled as a forced torsional vibrational system with damping. For small wave heights the system is linear, and Mynett et al. (1979) have provided solutions for Salter's geometry and for the geometry investigated experimentally for this report. There was good agreement between the theoretical predictions and the experimental results.

A Salter device with a fixed axis about which the cam rotates, is an idealized situation and very difficult to achieve when there are tides and when the water is deep. In order to examine the practicality of a Salter cam wave energy converter it is necessary to consider arrangements of the device which may operate satisfactorily in the ocean. Furthermore, the cost of providing electrical power from the waves is of importance for comparison with the various alternatives. These problems were investigated in the study reported here.

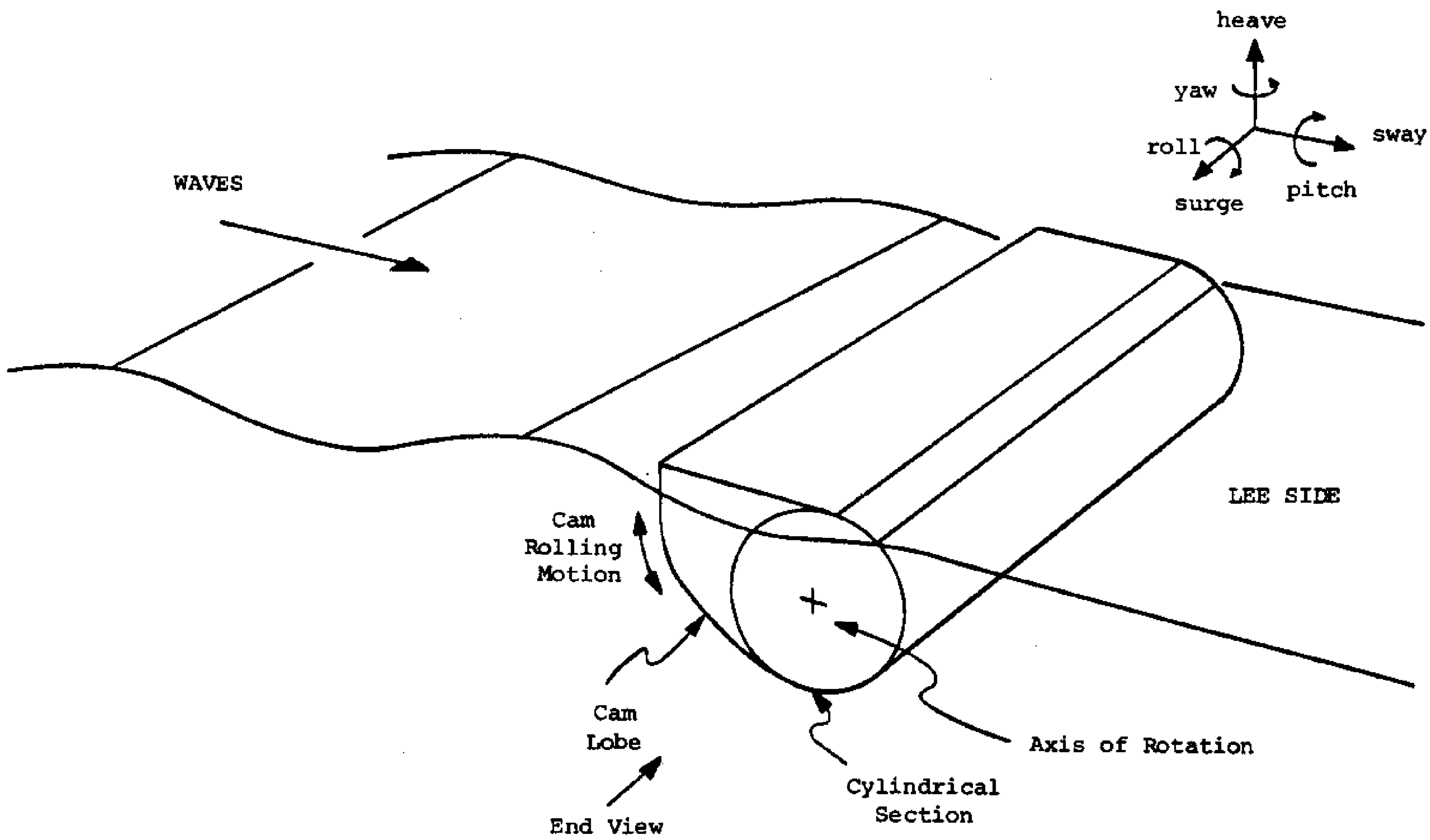


FIGURE 1.1 Salter Cam Wave Energy Extractor

The purpose of the present study was twofold:

1. to confirm the high level of efficiency of the Salter cam
2. to provide an engineering evaluation of the wave energy conversion system.

In order to carry out these tasks it was necessary to build a small model of the Salter cam and to test it first in a two dimensional channel and then in a much larger towing tank. The results from the experimental study were then utilized to provide an engineering evaluation, including the cost, of the system.

2. WAVE ENERGY

In the study of wave energy conversion many of the laboratory experiments are conducted in wave channels with regular waves of constant wave height and wave period. The theoretical models of such regular waves are well established, at least for waves having small wave heights.

At real sites where wave energy conversion devices may be used the waves are continuously varying in wave height, direction, and wave period. For such applications the simple theoretical models must be augmented with statistical information regarding the distributions of wave heights and wave periods. This statistical information is incomplete because of the difficulty and cost of collecting and processing the data, although some information is available.

REGULAR WAVES

From linear wave theory (Wiegel, 1964) the energy flux (or power) per unit of wave crest length for waves of height H and period T may be expressed as

$$P = (\rho g H^2/8) C_g \tag{2.1}$$

where C_g is the group velocity

$$C_g = \frac{gT}{4\pi} \tanh\,(2\pi d/L) \left(1 + \frac{4\pi d/L}{\sinh\,4\pi d/L}\right) \tag{2.2}$$

T is the wave period
L is the length
d is water depth

The wave period is related to the depth and wave length by the expression

$$T = \left[\frac{2\pi L}{g \tanh 2\pi d/L}\right]^{1/2} \tag{2.3}$$

The equation for wave power can be rewritten as

$$P = \frac{\rho g^2 H^2 T}{16\pi} \times P_* \tag{2.4}$$

where

$$P_* = 1/2 \tanh (2\pi d/L) \left[1 + \frac{4\pi d/L}{\sinh 4\pi d/L}\right] \tag{2.5}$$

for linear waves. In deep water $P_* = 0.5$ and wave power is approximately given by $P \simeq H^2T$ in kw/m when the wave height is measured in meters and the period in seconds. For larger waves the expression for P_* would take account of wave steepness (Nath and Williams, 1976).

Most of the experimental results contained in this report were obtained in a wave channel and a towing tank using regular waves of small height. It was assumed that linear wave theory would apply to these results and equations 2.3, 2.4 and 2.5 were used.

OCEAN WAVE STATISTICS

At a particular site the ocean waves would vary in wave height and wave period. The power available from the waves would depend on the average wind speed, wind direction, fetch, sheltering, and dissipation. In order to predict the wave power at a site it is necessary to determine the average value of the square of wave height, $\overline{H^2}$ and average period $\overline{T}$. The data presented in the literature are usually in the form of monthly or seasonal tabulations of numbers of waves within specified ranges of significant wave height, H_s, and wave period, T. Significant wave height is the average value of heights of the largest one third of the waves. The wave period is sometimes presented as the period associated with maximum energy as determined by energy spectra (Thompson, 1977) or as the average period of the waves having the largest one third wave heights.

It is customary to assume that the wave heights at a particular site have a Rayleigh distribution (Nath and Williams, 1976).

$$P_H(h) = P(H < h) = 1 - e^{-(h/z)^2} \tag{2.6}$$

where

$$z^2 = \overline{H^2}$$

With this distribution function the significant wave height $\underline{H_s}$ is related to the average of the square of the wave heights, $\overline{H^2}$ by the expression

$$H_s^{\,2} = \overline{2H^2} \tag{2.7}$$

The power, from linear wave theory, for waves having significant H_s and period T from equation 2.4 is

$$P = \frac{\rho g^2 H_s^{\,2} T}{32\pi} \times P_* \tag{2.8}$$

Nath and Williams (1977) provide a correction to the term P_* to account for wave steepness. This correction was small for the mean values of the wave data utilized in this study and was neglected.

The published distributions of significant wave heights and wave periods were used to calculate the seasonal and annual average wave power available at various sites using equation (2.8) in the form

$$\overline{P} = \left(\frac{\rho g^2}{32\pi}\right) \overline{H_s^{\,2}\ T\ P_*} \tag{2.9}$$

At deep water sites $P_* = 1/2$ and equation (2.9) becomes

$$\overline{P} = 1/2 \left(\frac{\rho g^2}{32\pi}\right) \overline{H_s^{\,2}\ T} \tag{2.10}$$

Equation (2.9) and (2.10) were utilized in conjunction with the data from Thompson (1977) for coastal sites and Hogben and Lumb (1967) for ocean regions to estimate the average available power from ocean waves for six regions of the U.S. coasts (Bisceglia, 1978):

1. North Atlantic, Latitude 40 - 50°
2. Mid Atlantic, Latitude 30 - 40°
3. South Atlantic, Latitude 20 - 30°
4. North Pacific, Latitude 40 - 50°
5. Mid Pacific, Latitude 35 - 40°
6. South Pacific, Latitude 30 - 35°

The seasonal and annual average power levels, $\overline{P}$, mean significant wave heights $\overline{H}_S$, and mean periods $\overline{T}$ are presented in Table 2.1 (if the significant wave heights have a Rayleigh distribution then $\overline{H}_S = .886 \sqrt{\overline{H_S{}^2}}$).

The data presented in this table provides the potential for wave power production around the U.S. coasts while the realization of a proportion of this potential depends on the performance of wave energy conversion devices such as the Salter Cam studied for this report.

SIMULATED RANDOM SEAS

The wave maker at the MIT towing tank has a control mechanism which can provide simulated two dimensional random seas based on the Pierson-Moskowitz sea spectrum. This spectrum is an empirical equation for the spectral density $s(\omega)$ in terms of the wind speed, V_k.

$$s(\omega) = (135/\omega^5) \quad e^{-9.7 \times 10^4/V_k{}^4 \ \omega^4} \tag{2.11}$$

where V_k is the wind velocity measured at 64 ft above the ocean surface, kts.

ω is the circular frequency, radians/s.

s is the spectral density, $ft^2 s$.

The significant wave height, H_s, for this formulation is

$$H_s{}^2 = 3.5 \ V_k{}^4/10^4 \tag{2.12}$$

The circular frequency at the peak of the spectrum is

$$\omega_p = 16.69/V_k \tag{2.13}$$

In the towing tank the random seas are of course scaled down from the Pierson Moskowity spectrum in wave height and wave period to be suitable for model testing. The scale factor for wave height is 96 and is $\sqrt{96}$ for wave period.

TABLE 2.1

WAVE CLIMATOLOGY

ALL SEASONS

	COASTAL			DEEP OCEAN		
	$\bar{H}_s$ m	$\bar{T}$ s	$\bar{P}$ kw/m	$\bar{H}_s$ m	$\bar{T}$ s	$\bar{P}$ kw/m
North Atlantic	1.0	8.5	5.2	3.2	6.3	37.1
Mid Atlantic	.8	7.9	3.1	2.7	5.9	25.6
South Atlantic	.7	6.7	2.5	2.4	6.0	22.1
North Pacific	---	---	---	3.4	11.0	81
Mid Pacific	1.0	10.4	5.7	2.6	10.3	52
South Pacific	.9	13.2	4.9	2.1	13.2	25

DECEMBER TO FEBRUARY

	COASTAL			DEEP OCEAN		
	$\bar{H}_s$ m	$\bar{T}$ s	$\bar{P}$ kw/m	$\bar{H}_s$ m	$\bar{T}$ s	$\bar{P}$ kw/m
North Atlantic	1.2	8.5	8.7	3.6	6.7	54.6
Mid Atlantic	1.0	8.3	5.2	2.9	6.2	31.1
South Atlantic	.9	7.7	4.0	2.5	6.3	28.0
North Pacific	---	---	---	5.2	10.0	150
Mid Pacific	1.1	10.8	6.1	3.3	10.9	90
South Pacific	.9	11.9	6.2	2.4	11.9	33

MARCH TO MAY

	COASTAL			DEEP OCEAN		
	$\bar{H}_s$ m	$\bar{T}$ s	$\bar{P}$ kw/m	$\bar{H}_s$ m	$\bar{T}$ s	$\bar{P}$ kw/m
North Atlantic	1.1	8.2	5.6	3.1	6.3	38.8
Mid Atlantic	.9	8.1	3.3	2.6	6.0	24.9
South Atlantic	.8	7.6	2.6	2.3	6.0	21.5
North Pacific	---	---	---	2.7	9.8	45
Mid Pacific	1.0	10.4	6.9	2.4	9.7	27
South Pacific	.9	13.3	6.6	2.1	11.8	22

JUNE TO AUGUST

	COASTAL			DEEP OCEAN		
	$\bar{H}_s$ m	$\bar{T}$ s	$\bar{P}$ kw/m	$\bar{H}_s$ m	$\bar{T}$ s	$\bar{P}$ kw/m
North Atlantic	.8	7.4	2.9	2.4	5.7	21.2
Mid Atlantic	.6	8.0	1.9	2.2	5.6	16.9
South Atlantic	.5	6.7	1.5	2.0	5.6	14.6
North Pacific	---	---	---	1.7	11.2	17
Mid Pacific	.9	11.6	6.7	1.5	10.6	13
South Pacific	.8	14.0	6.2	1.5	13.5	12

SEPTEMBER TO NOVEMBER

	COASTAL			DEEP OCEAN		
	$\bar{H}_s$ m	$\bar{T}$ s	$\bar{P}$ kw/m	$\bar{H}_s$ m	$\bar{T}$ s	$\bar{P}$ kw/m
North Atlantic	1.1	8.7	5.4	3.0	6.4	37.1
Mid Atlantic	.9	8.5	3.7	2.6	6.0	25.0
South Atlantic	.8	7.0	2.6	2.3	6.1	21.6
North Pacific	---	---	---	3.9	11.9	110
Mid Pacific	.9	11.8	7.2	3.3	10.4	80
South Pacific	.7	14.4	5.4	2.2	14.3	35

3. THE EXPERIMENTAL MODEL OF THE SALTER CAM

The experiments on the model conversion device were planned for the wave channel as a two-dimensional device and it was also proposed to place the model in the towing tank in a simulated open-ocean situation. The model was therefore designed to fit neatly into the wave channel (Davis, 1978). It would then be of adequate size to operate in the towing tank.

The cam was designed with a cylinder of nominal diameter 6 inches (16.8 cm) in polyvinyl chloride (PVC) as the backbone and a foamed plastic addition to form the cam, strapped to it. The foamed plastic was coated with fiberglass and three 2.5 cm (1 in.) holes were drilled parallel to the cam axis, so that metal rods could be inserted to adjust the trim and inertia of the cam. The dimensions of the cam are shown in Figure 3.1. A fixed shaft was placed on the center line of the cylinder to act as an axle.

In order to take energy out of the wave a form of energy absorber or damper is required. Originally, a hydraulic damping system was built and tested. This device did not provide large enough variation in load, nor precise control. A second generation of hydraulic device was designed and built and although this provided better characteristics it was unsatisfactory. Finally, a D.C. torque motor was selected as the damper.

The torque motor used in the experiments had an integral tachometer to sense angular speed. The tachometer output provided the electrical signal necessary to produce linear damping (i.e. proportional to angular velocity). The circuits to do this are described by Davis (1978). The motor - tachometer

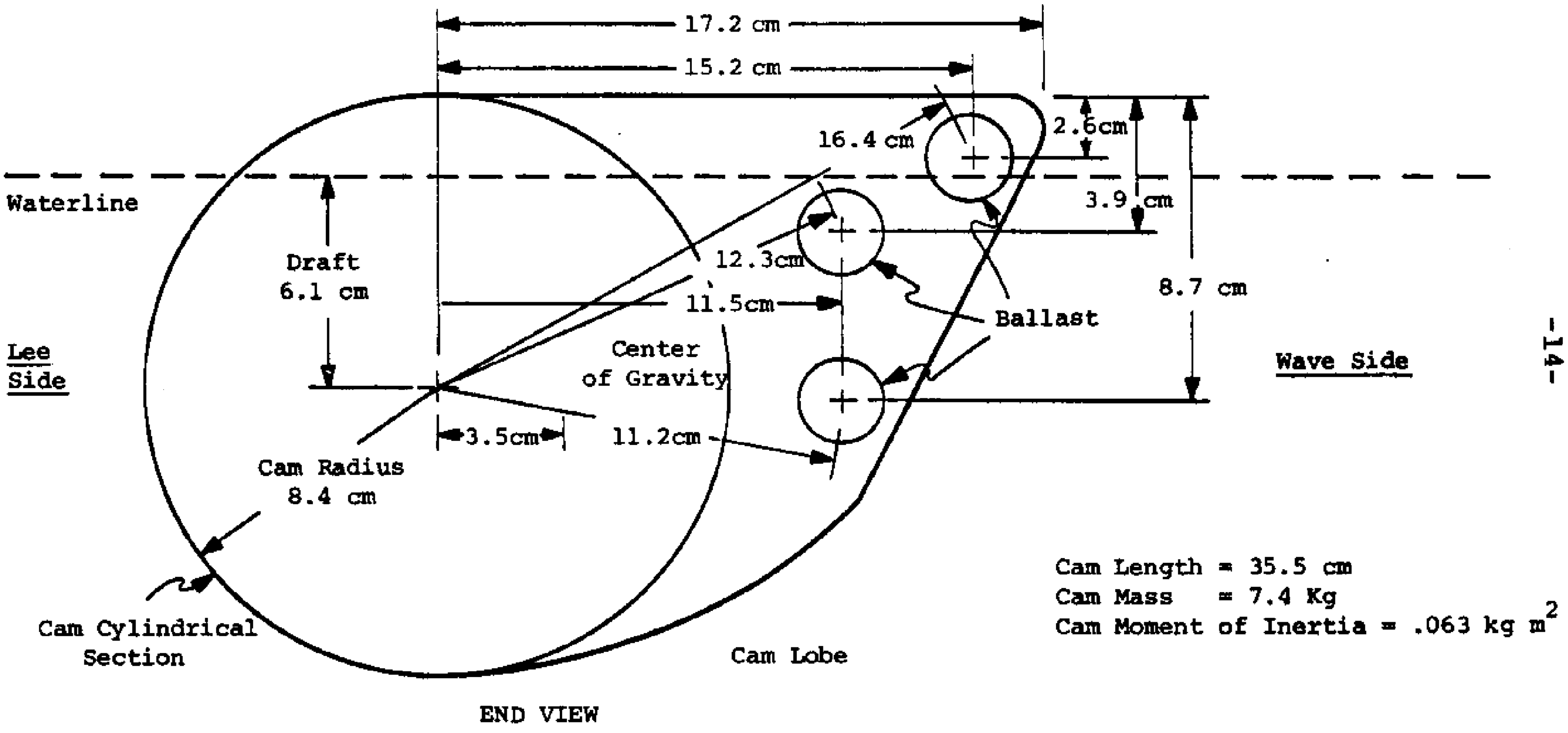


FIGURE 3.1 Cam Geometry

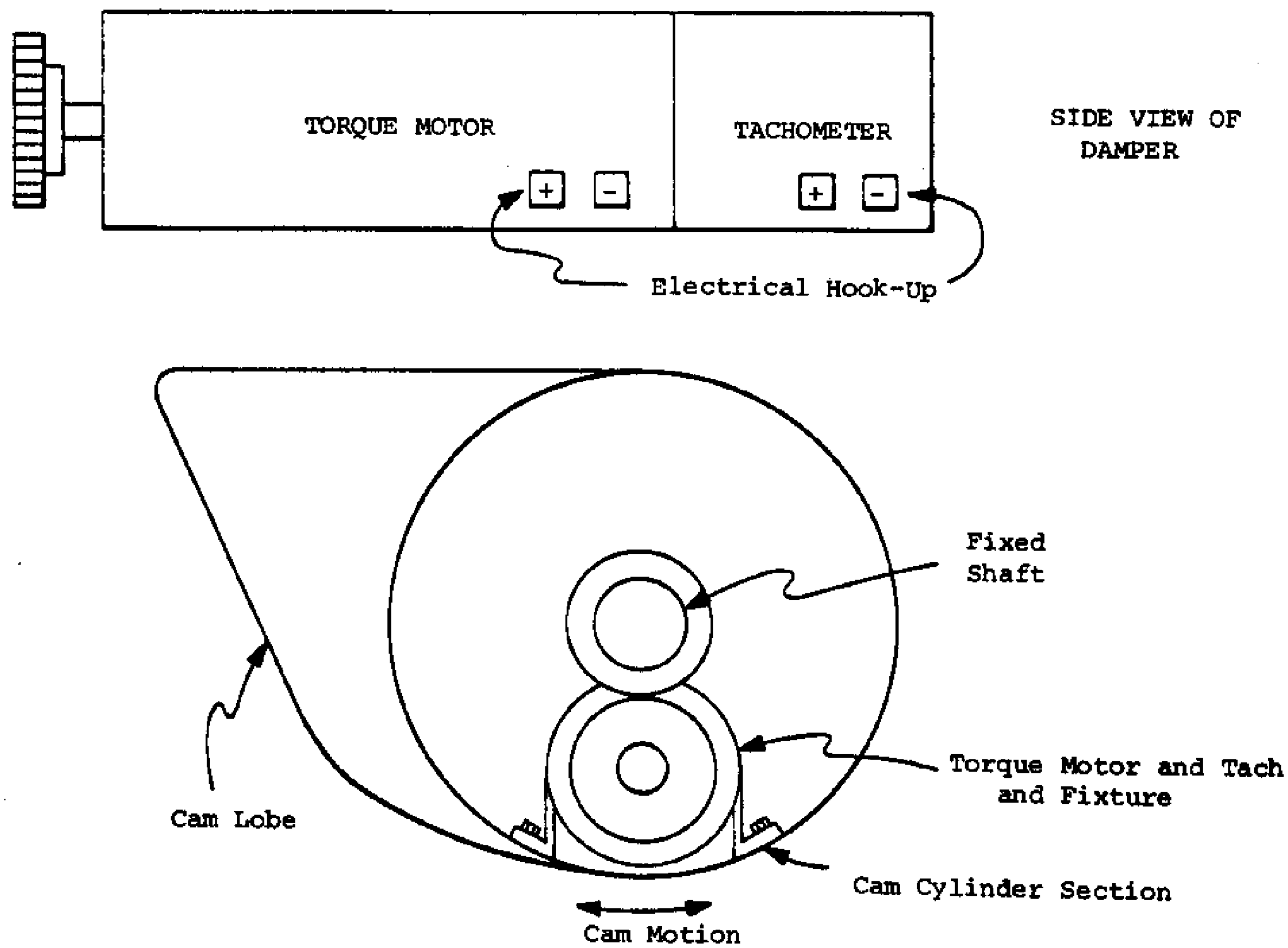
combination was mounted inside the cylindrical section of the cam and geared to the fixed shaft, which was placed along the axis of the cylinder (Figure 3.2). The mechanical arrangement required seals to prevent water entering the cylinder and bearings to allow the cam to rotate about the fixed shaft (Figure 3.3).

The shaft was supported in an aluminum frame for the wave channel tests. Several different arrangements of aluminum frames were used for the tests in the towing tank. The basic requirements for the frames were that they should be rigid and lightweight.

The power absorbed by the cam was measured by means of a torque meter (strain gages) on the fixed shaft and a sensor (potentiometer) was arranged to measure the angular movement of the cam relative to the fixed shaft. The work done per cycle was obtained by determining the area of the torque-angular displacement diagram. In some early experiments the area was measured using a planimeter on the photographic record of an oscilloscope trace. For many of the later tests a digital data acquisition system using a mini-computer was utilized to measure transducer signals and to provide numerical integrations of the results.

The strain gages used to measure torque were placed on the fixed shaft outside the cylinder (Figure 3.3) so that the friction of the seals and bearings would be included in the torque measurements.

Before each series of experiments the instrumentation was calibrated by applying known torques and displacements to the cam. The calibration constants were set into the computer data system.



END VIEW OF DAMPER MOUNTING

FIGURE 3.2 Electromechanical Cam Damper

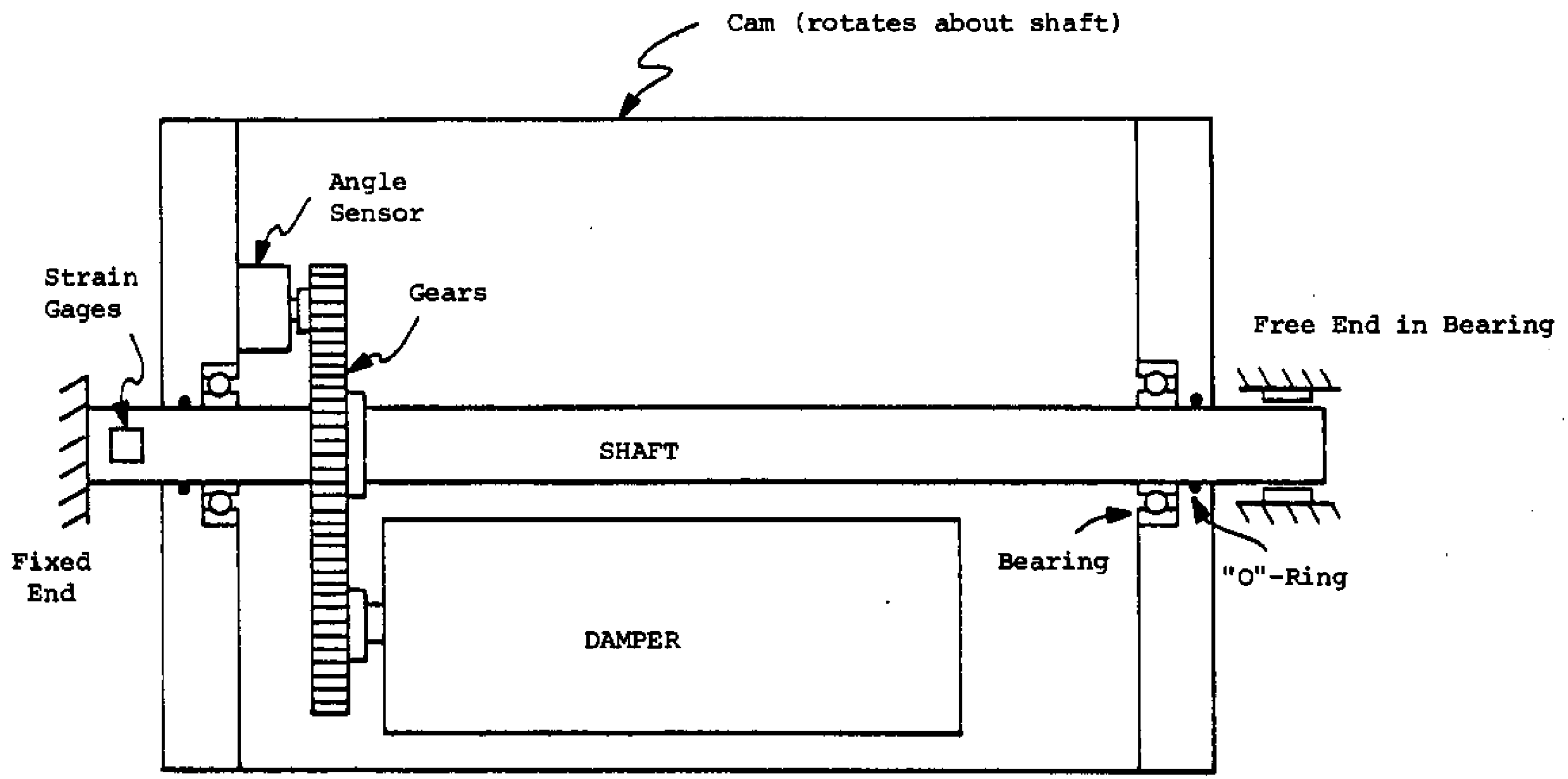


FIGURE 3.3 Load Cell and Angle Sensor (Work Measuring Equipment)

4. EXPERIMENTS IN THE TWO-DIMENSIONAL WAVE CHANNEL

THE WAVE CHANNEL

The experiments were conducted in a wave channel of approximate size 23 m (75 ft.) long, 0.38 m (15 in.) wide, and 0.76 m (30 in.) deep with a long sloping beach at the far end. The water depth can be varied but was standardized for the experiments at 0.38 m (15 in.). The waves are produced by a wave paddle with a slight forward lean at the surface and driven by a variable speed motor through a crank mechanism. The crank radius may be changed to vary the wave height. The wave maker could provide regular waves of periods between 0.6 and 2 seconds. Waves of shorter period than 0.6 seconds were found to be too irregular for use.

The walls of the tank are heavy plate glass to facilitate observations. The glass plates are supported by a welded steel frame with a steel track along the length of the channel to support instrumentation. Two resistance wave probes with calibrating fixtures and a hot pen recorder are available for wave height measurements.

TEST PROCEDURE

The performance characteristics of the cam were determined in a two dimensional situation in the wave channel. For the device operating at a fixed axis position (depth) and with a specified ballast the performance characteristics of importance were the cam efficiency, the angular amplitude, and the torque amplitude for range of the independent variables wave period, wave height, and damping setting. However, for most of the tests the wave height was set at a value of approximately 2 cm.

The efficiency of a wave conversion device is defined as

$$\varepsilon = \frac{\text{power absorbed}}{\text{power of the incident waves}} \tag{4.1}$$

where power absorbed is the work done per period/wave period and power of the incident waves is given in Chapter 2.

$$P = \frac{\rho g^2 H^2 T}{16\pi} \times P_* \tag{2.4}$$

where H is the incident wave height

T is the wave period

P_* is a factor depending on wave length, (wave period), and water depth.

To determine the incident wave height it is necessary to separate the incident wave from the sum of the incident and reflected wave traces that appear on the recorder paper. When the cam has been operating steadily for several minutes a standing wave pattern is superimposed on the progressive incident waves. This standing wave is generated when the wave reflected from the cam returns to the wavemaker and is reflected from it. At some wave periods the standing wave system is very clear and the wave recording device can be utilized to measure both in the incident and the reflected wave. When steady conditions have been established, the wave probe is moved forward slowly about one wavelength and the envelope of the trace is used to determine the incident and the reflected wave heights, Davis (1978). This method is not accurate when the standing wave is not stable. A technique was developed for such situations by taking the wave measurements and cam performance data before the reflected wave from the cam reached the wave probe. By stopping the wave maker at the appropriate time it was found that a fixed wave probe could measure the incident and reflect waves separately.

The power absorbed by the damper system can be predicted from wave height measurements using a power balance around the cam assuming that there is no wave dissipation then

Power Extracted = Incident Power - Reflected Power - Transmitted Power

The incident power and reflected power can be measured by the wave probe ahead of the cam while a wave probe downstream of the cam can be utilized to determine the transmitted power. A comparison of the efficiencies based on the cam instrumentation

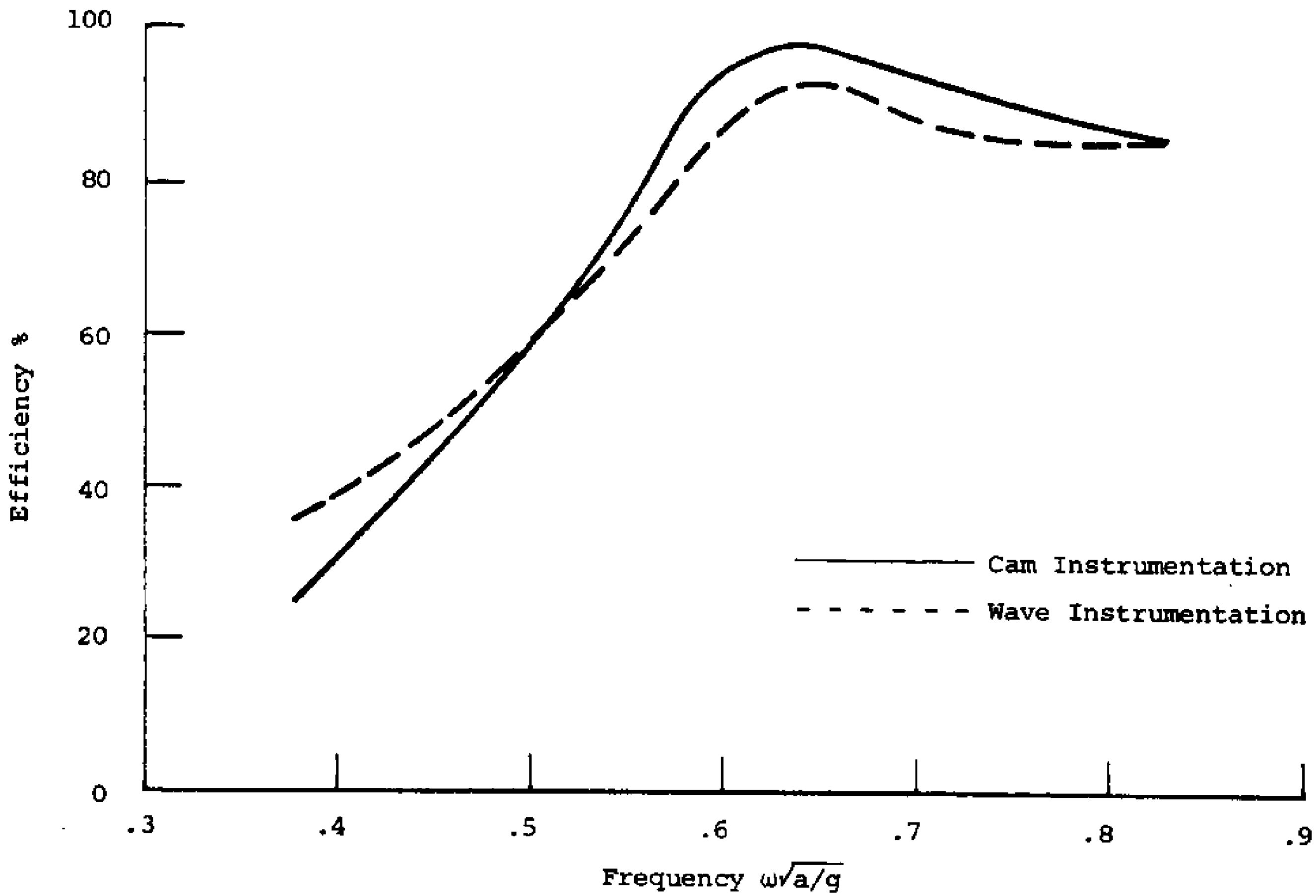


FIGURE 4.1 Comparison of Wave and Cam Power Measurements

and the wave data is shown on Figure 4.1. It was found that the agreement was reasonable over the whole range of periods and damping settings. During all tests in the wave channel the power absorbed by the cam was measured using the instrumentation in the cam, and by wave height measurements, as a check.

Non-Dimensional Performance Parameters

The non-dimensional parameters utilized have to present the performance data for the cam and to predict the operation of full scale devices were based on those developed in the hydrodynamic studies of Mynett (1978). The non-dimensional groups are as follows:

$$\text{efficiency} = \varepsilon = \frac{\text{power absorbed by the damper}}{\text{power of incident waves}} \tag{4.2}$$

$$\text{frequency,} \quad \hat{\omega} = \omega \sqrt{\frac{a}{g}} \tag{4.3}$$

where a is the radius of the cylindrical section of the cam.

$$\text{angular response,} \quad \hat{\theta} = \frac{\theta a}{H} \tag{4.4}$$

where θ is twice the angular amplitude of the rolling motion.

$$\text{moment of inertia,} \quad \hat{I} = \frac{I}{\rho a^4} \tag{4.5}$$

where I is the moment of inertia per unit length of the cam about the axis of the cylinder.

$$\text{damping, } \hat{\lambda} = \lambda/\rho a^4 \sqrt{g/a} \tag{4.6}$$

where λ is the damping factor for unit length of cam,

$$\text{for linear damping, } \lambda = \frac{2}{\pi^2} \frac{(\text{work}) \times T}{\theta^2} \tag{4.7}$$

In most of the performance characteristics the efficiency of the cam (or the response), is plotted against frequency with damping as the parameter. In some of the earlier tests the damping was adjusted until maximum efficiency was achieved so that the performance was presented at optimum damping. In the later tests damping was set at specified values.

The moment inertia of the cam was measured in air by the bifilar pendulum method of Timoshenko, as described by Wiegel (1964). Adjustments were made to the moments inertia by adding small masses at known distances from the axis.

EXPERIMENTAL RESULTS

Initially tests were conducted at the damping level for maximum efficiency. In later tests damping was varied in order to provide more complete performance characteristics.

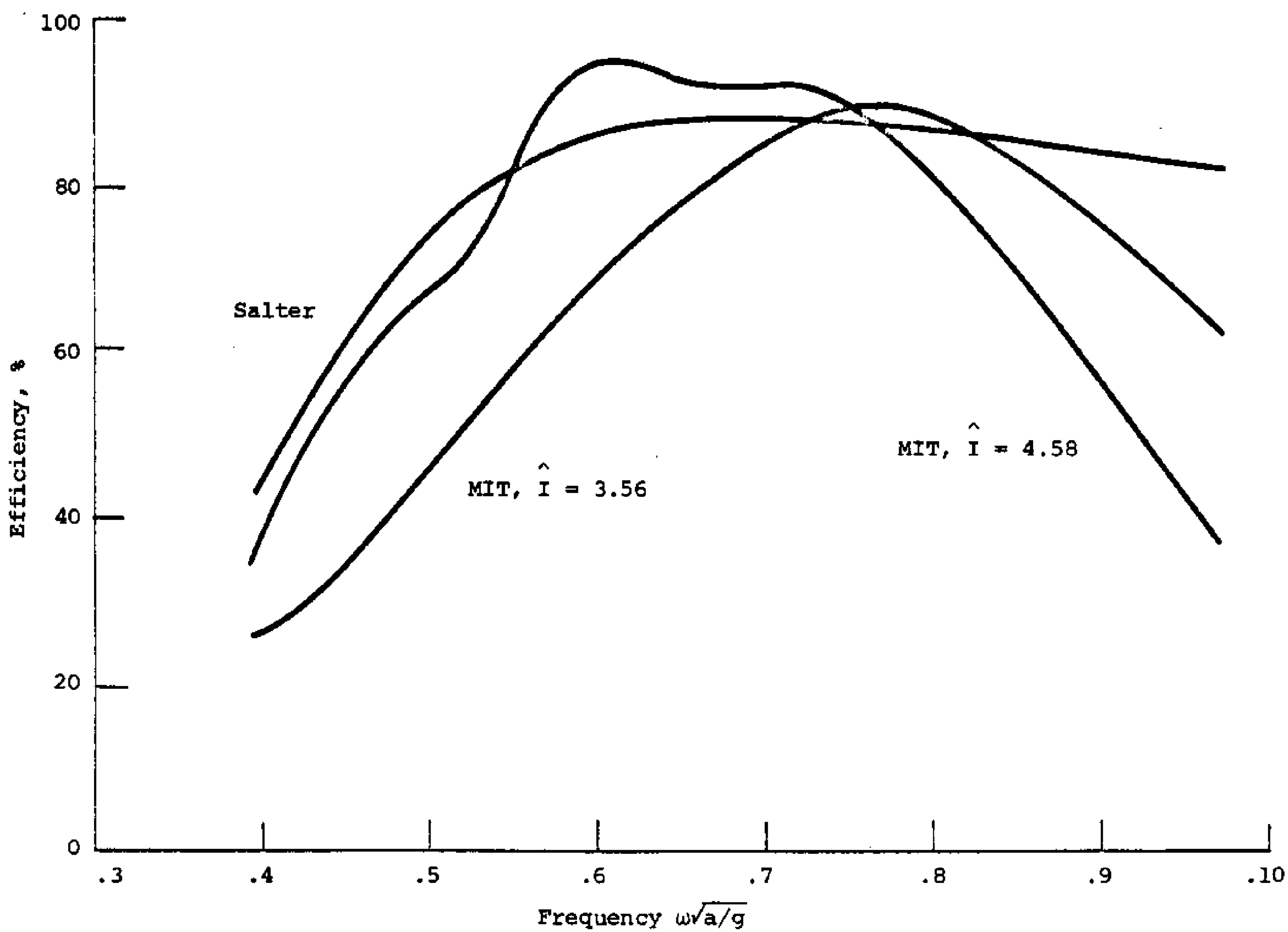


FIGURE 4.2 Comparison of Salter Experimental Results with Experiments at MIT, at Optimum Damping

All the tests were conducted with essentially the same geometrical arrangement and attitude. The depth of the fixed shaft and the angle of the lobe of the cam were set at fixed values for all tests in the wave channel and the towing tank. When changes in inertia were made, the cam was then trimmed to obtain the desired attitude.

All the tests in the wave channel were conducted with a wave height of approximately 2 cm corresponding to H/a of 0.24. At this value of wave height the efficiency was obtained at optimum damping for the range of wave periods. These results, obtained by Davis (1978), are presented on Figure 4.2 for two valuesof non-dimensional moment of inertia. The efficiency data published by Salter et al (1976) is also shown on the same figure. It can be seen that the excellent efficiency levels claimed by Salter have been confirmed, although Salter's results indicate a wider frequency range of high efficiency than our results have shown. The influence of changes in moment of inertia are as expected with the larger moments of inertai having their peak efficiencies at lower frequencies.

Some changes were made to the distribution of mass in the cam to improve the trim for the tests in the towing tank. The cam was tested in the two dimensional wave channel after those modifications were made, partly as a check of the instrumentation. More complete characteristics were obtained and are presented in Figure 4.3. The characteristics are presented as response plotted against frequency for three values of damping. Contours of efficiency are also presented.

The results indicate that there is a region of high efficiency around $\omega\sqrt{a/g}$ of 0.65 - 0.8. The magnitude of the response, $\theta a/H$, is about 0.9 in the region of highest efficiency. This indicates that at high efficiency the periphery of the cylindrical section of the cam moves a distance (θa) which is approximately equivalent to the wave height.

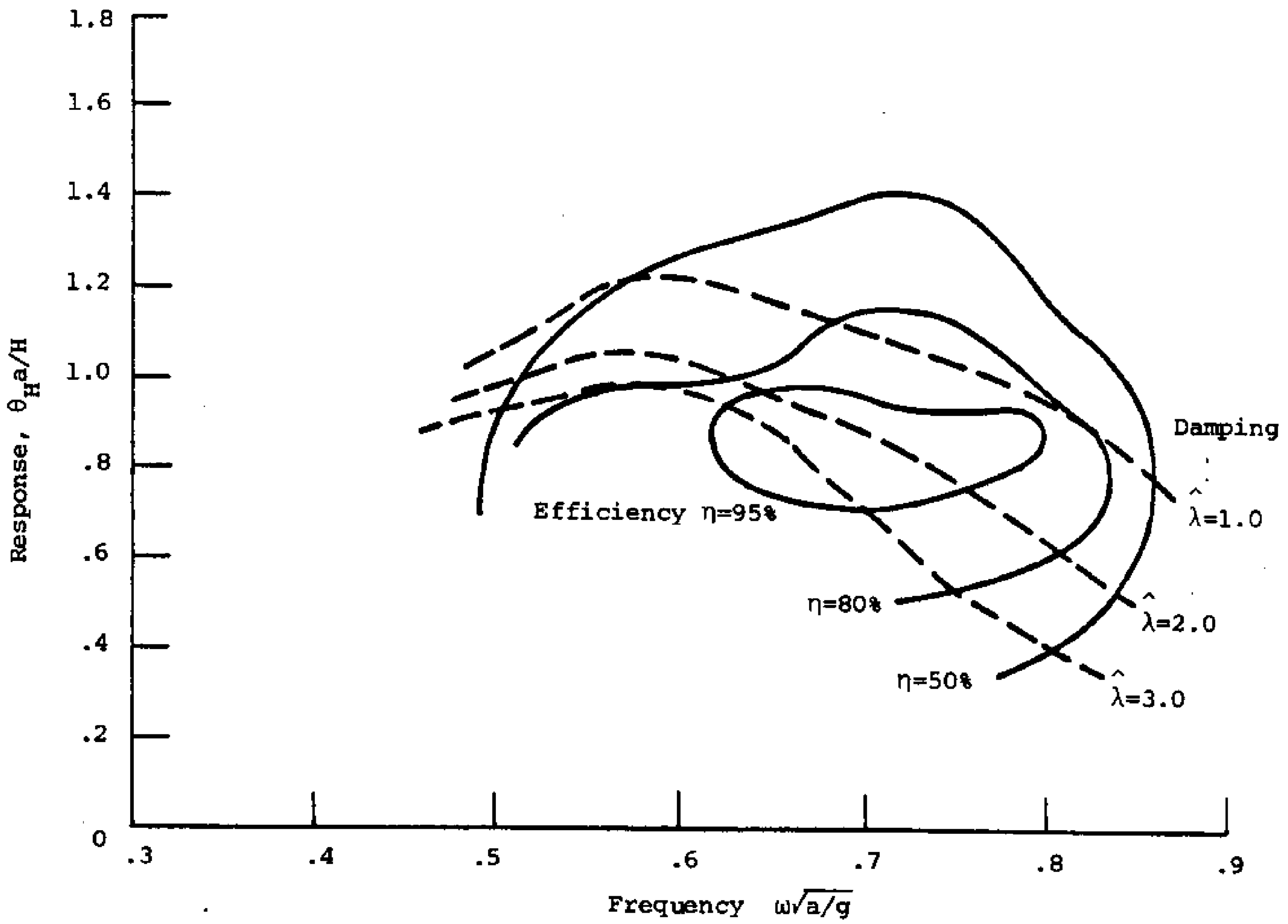


FIGURE 4.3 Performance Characteristics of the Cam Model in the Two-Dimensional Wave Channel, $\hat{I}$ = 5.46

5. TESTS IN THE TOWING TANK

THE TOWING TANK

The ship model towing tank is 2.62 m (8 ft. 7 in.) wide and 32.9 m (108 ft.) long and normally operates with a water depth of 1.22 m (4 ft.), which can be varied. There is a beach at one end and a wave paddle at the other. The paddle is hinged at the lower end and actuated by a servo-controlled hydraulic cylinder at the top. A sine wave signal generator in the control room drives the servo for regular waves. If random seas are desired, magnetic tapes of various sea states can be played on a recorder to control both the frequency and the amplitude of the wave paddle.

The instrumentation at the wave tank includes wave height gages and force blocks (force measuring transducers) together with suitable recording equipment. The standard of the instrumentation and of wave paddles and its controls were very much higher than with the wave channel. There were less scatter and more reproducible results with experiments in the towing tank.

There was one drawback with the measurements in the towing tank, namely that there was no reasonable method of checking the results of the cam instrumentation with measurements of the incident, reflected, and transmitted waves because the cam did not span the tank. However, at various times the instrumentation was recalibrated and also tested in the smaller wave channel for consistency.

TEST PROCEDURE

In the first series of tests the cam was rigidly supported from the walls of the wave tank, while in later tests the cam model was allowed to float with various mooring arrangements.

In all the tests with regular waves the measurements were taken before any reflected waves returned to the cam from the beach or the paddle. Careful timing was required to ensure that this was achieved.

The measurement of wave height utilized the fact that there was excellent control of the wave paddle. The wave maker was calibrated by a test run with the cam removed and a wave probe at the position of the cam. The setting of the wave maker to produce a 2 cm wave was determined for a range wave periods. As a check wave probes were placed in the tank during operation and records were taken.

At each wave frequency the damping levels were set at the selected values on the servo-controller for the torque motor and the torque and angular displacements were measured. In some of the earlier tests the work per wave cycle was determined using a photograph of the oscilloscope trace, while in later tests the digital data acquisition method was used.

In some tests with the cam held rigidly, the sway forces (along the axis of the towing tank) were measured using a force block. Attempts to measure the heave forces were not very successful because it was difficult to device a fixture rigid enough to do this.

Several methods of attaching the moorings to the bottom of the towing tank were tried. Initially eye bolts in the bottom of the tank were used, then a rectangular steel frame was used, finally ten and fifty pound weights were used as anchors.

Tests in Random Seas

Magnetic tapes to drive the wave paddle are used in the towing tank to provide Pierson-Maskowitz Spectra for the testing of models. These tapes provide simulations of random seas for a selected set of spectral peak frequencies, ω_p. The value of significant wave height, H_s, for each value of ω_p can be adjusted. With the testing of ship models the usual procedure is to adjust the value of significant wave height to correspond to the frequency, ω_p, in the Pierson-Maskowitz Spectrum. However, for the testing of the cam it was decided to carry out the tests with a fixed value of significant wave height of 3.2 cm for all values of frequency.

As in the case of regular waves the efficiency was determined as a function of non-dimensional frequency $\omega\sqrt{a/g}$, where the frequency $\omega = \omega_p$. The damping was selected to provide peak efficiency.

Tests with the cam showed that with random seas the instrumentation data should be collected for ten minutes, which corresponds to one hundred minutes in the ocean. The ten minute testing time was carried out in segments of thirty seconds in order to avoid reflection problems from the model and the beach. There was a waiting period between each segment to allow the tank to settle. The computer data acquisition system was used to calculate the total work during the test periods.

TEST RESULTS

Tests were conducted with cam axle supported rigidly from a structure spanning the towing tank. In addition experiments were carried out with various forms of moorings. Both rigid and moored systems were tested with regular and random seas.

Rigid Frame Tests

Initial tests in regular waves with the cam model placed centrally in the towing tank showed that the model was absorbing more wave power than was available in a two dimensional

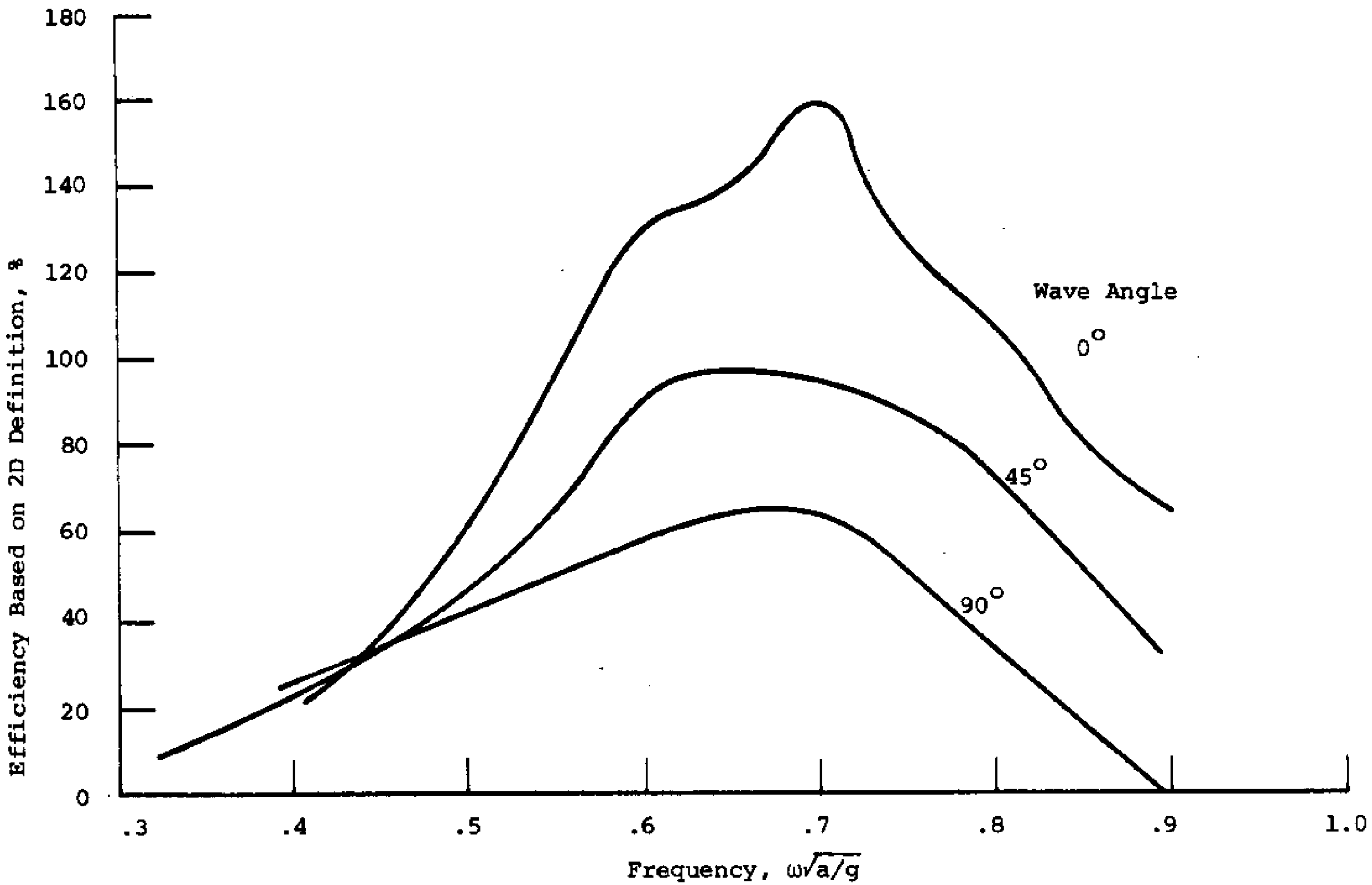


FIGURE 5.1 Cam Efficiency Measured at Various Wave Incidence Angles

strip the width of the cam. Using the two dimensional definition of efficiency, values of 150% were regularly measured, Figure 5.1. The instrumentation was checked by recalibration to determine the instrument constants and by returning it to the two dimensional wave channel for comparison with the method using incident, reflected, and transmitted wave heights. No discrepancies were observed, so that it was concluded the unexpected results were correct; that is to the experimental accuracy of about $\pm$ 5%. Further experiments with the model placed close to a wall of the tank gave very similar results to the tests when the model was put in the middle of the tank.

Two recent theoretical studies have indicated how in open seas efficiencies greater than the 100% (based on two dimensional definition) might be explained. Budal (1977) predicted that the regular spacing of wave energy devices could increase the efficiency and Standing (1978) indicated that diffraction effects could raise the efficiency above two dimensional levels. A satisfactory explanation of this phenomenon was not obtained during the test program.

The non-dimensional performance characteristics for the model cam placed on a rigid frame over the towing tank are presented on Figure 5.2. The characteristics are rather different from those measured in the wave channel (given in Figure 4.3). However, the peak efficiency is at a similar

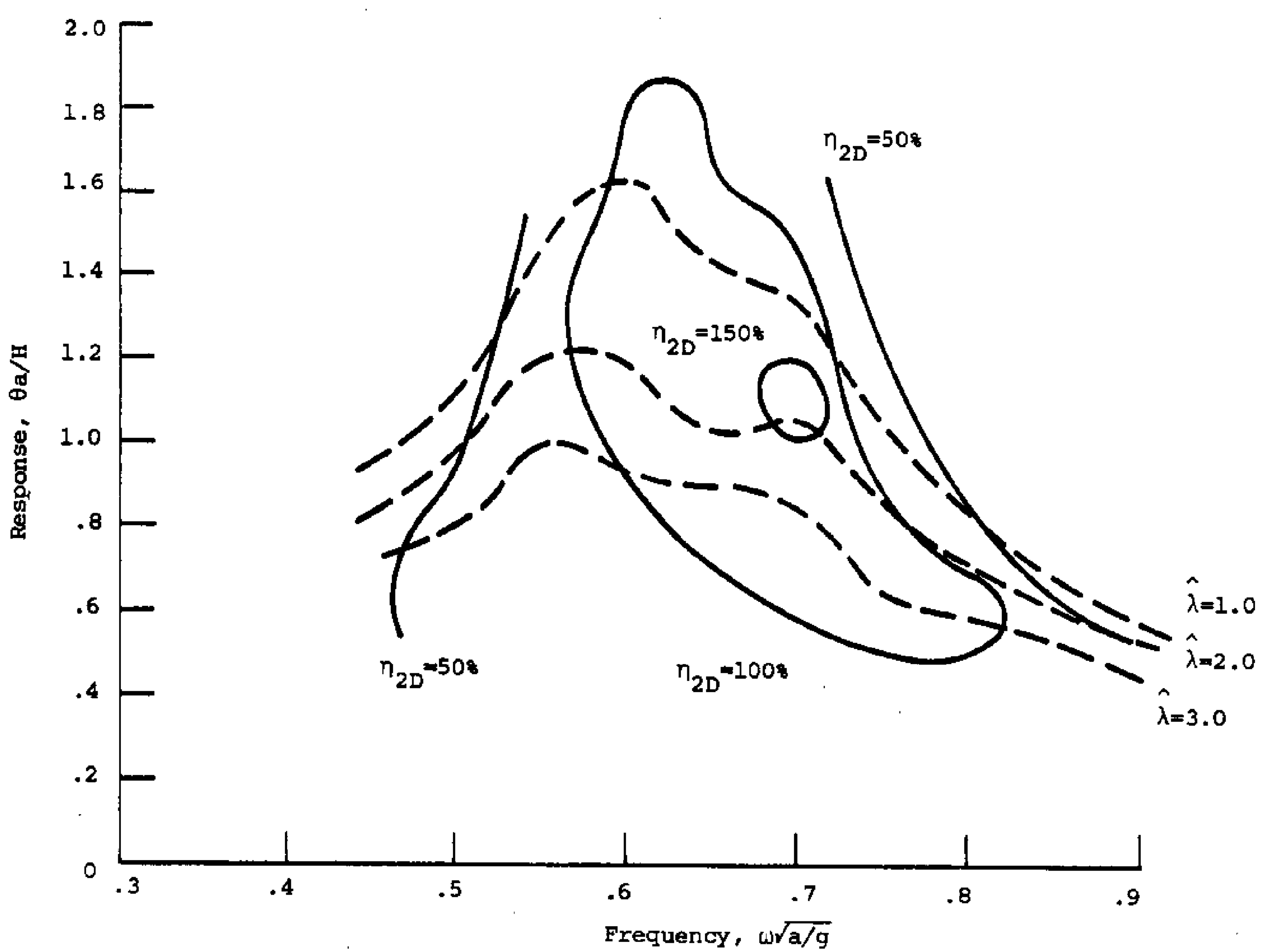


FIGURE 5.2 Performance Characteristics of the Cam Model in the Towing Tank, Rigid Frame

frequency of $\omega\sqrt{a/g} = 0.7$ and the response in the region of highest efficiency is only slightly higher than in the two-dimensional case.

Tests were conducted to determine the influence of angle of the wave crests to the cam. The model, placed in the center of the tank, was supported at 45° and perpendicular to the tank. The level of accuracy of the tests with the inclined model was lower than the other tests because of reflections from the walls. The results, presented in Figure 5.1, show the performance deteriorates when the waves are inclined to the model. Again there was an unexpected result. The efficiency when the model was placed perpendicular to the waves was much higher than anticipated. This result appears to be associated with two factors:

a. The buoyancy effect of the lobe of the cam.

b. The wavelength/cam length.

When the cam was set with axle of cam lower in the water than the standard setting, such that the buoyancy exerted a small moment about the axle, it was found that the efficiency was very low.

At wavelengths commensurate with the length of the cam it was observed that the cam efficiency was low. It appears that the cam performance when placed perpendicular to the wave crests can be explained qualitatively in terms of the quasi-static buoyancy forces.

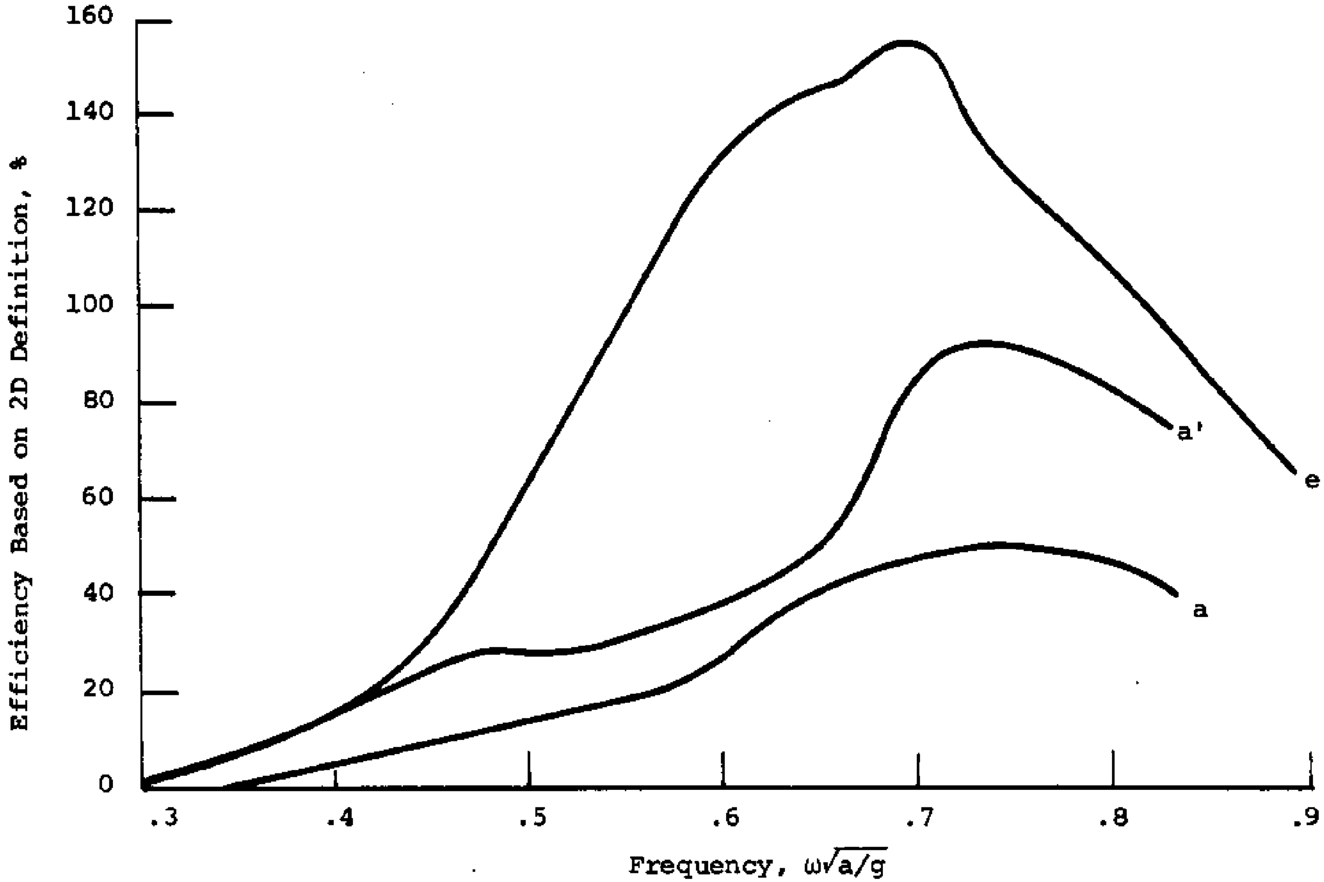


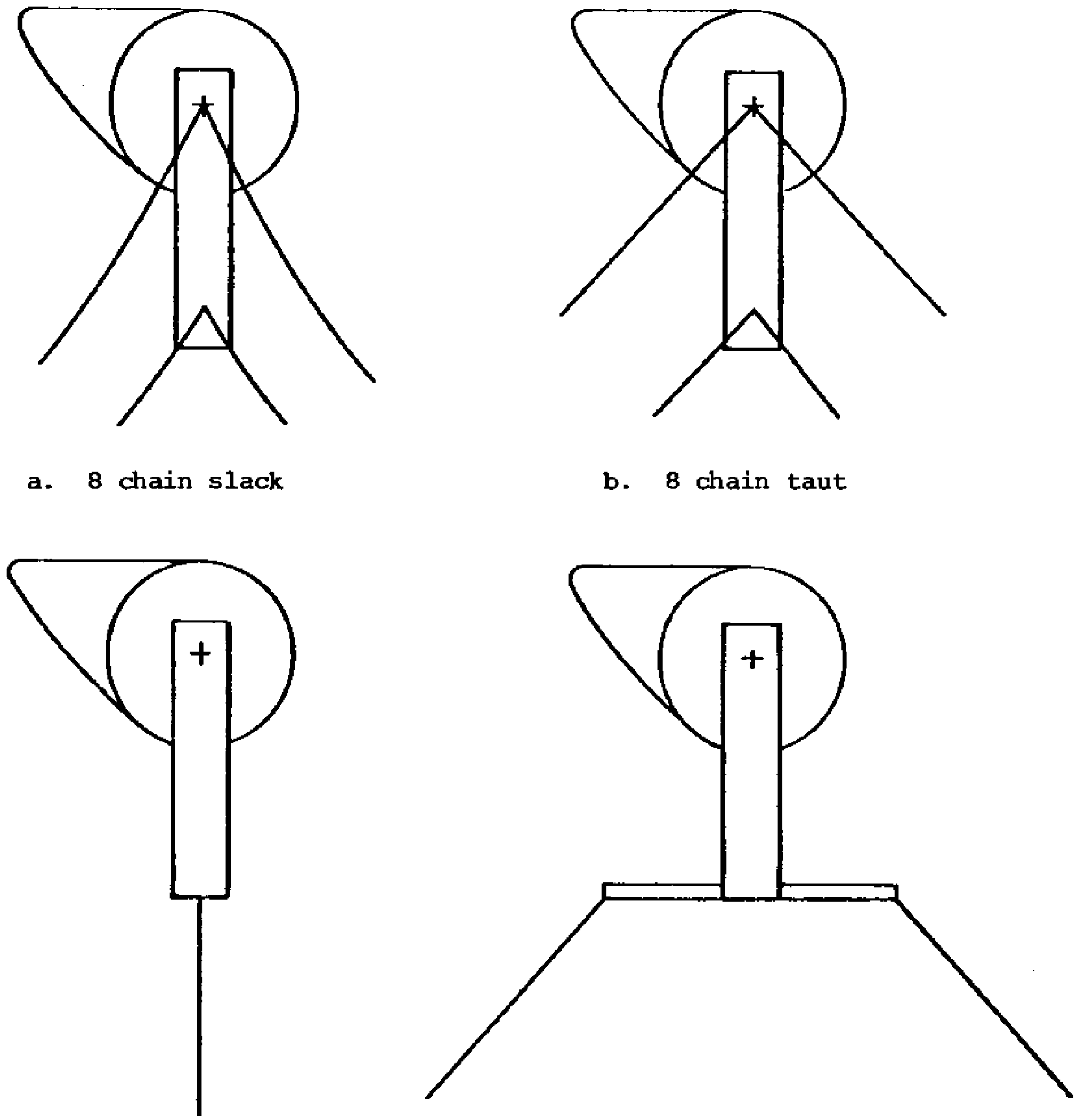
FIGURE 5.3 Efficiency of Moored Cam Systems
a - slack moorings, a' - slack moorings, added mass and damping
e - fixed frame

Moored Model Tests

Preliminary calculations indicated that the natural frequencies of the floating model cam in several modes would be close to the frequency for best efficiency as a power absorber. The implication was that the cam's response in these modes would reduce the efficiency of the cam as a wave power absorber. Initially attempts were made to change the natural frequencies of the offending modes. Later various taut line moorings were devised to produce a stable platform for the cam axle.

The tests with the slack mooring, as anticipated, demonstrated that natural frequencies of the cam and its support frame in heave (vertical motion) and in roll were close to the natural frequency of the cam as a power absorber. In regular waves the frame heaved and rolled excessively, and, as a result, the efficiency was low, Figure 5.3. Horizontal flat plates were added to the frame in order to introduce more damping and to increase the effective mass of the cam and frame. The efficiency of the cam was improved but the dimensions of plates were such that the solution was impractical.

Several forms of tautline mooring were tested, Figure 5.4. With tautline moorings the cam is pulled to the desired freeboard by the moorings. In order to achieve this situation the cam must be much lighter than the desired displacement.



a. 8 chain slack

b. 8 chain taut

c. 1 chain taut

d. 4 chain taut

FIGURE 5.4 Arrangements of Moorings Tested in the Towing Tank

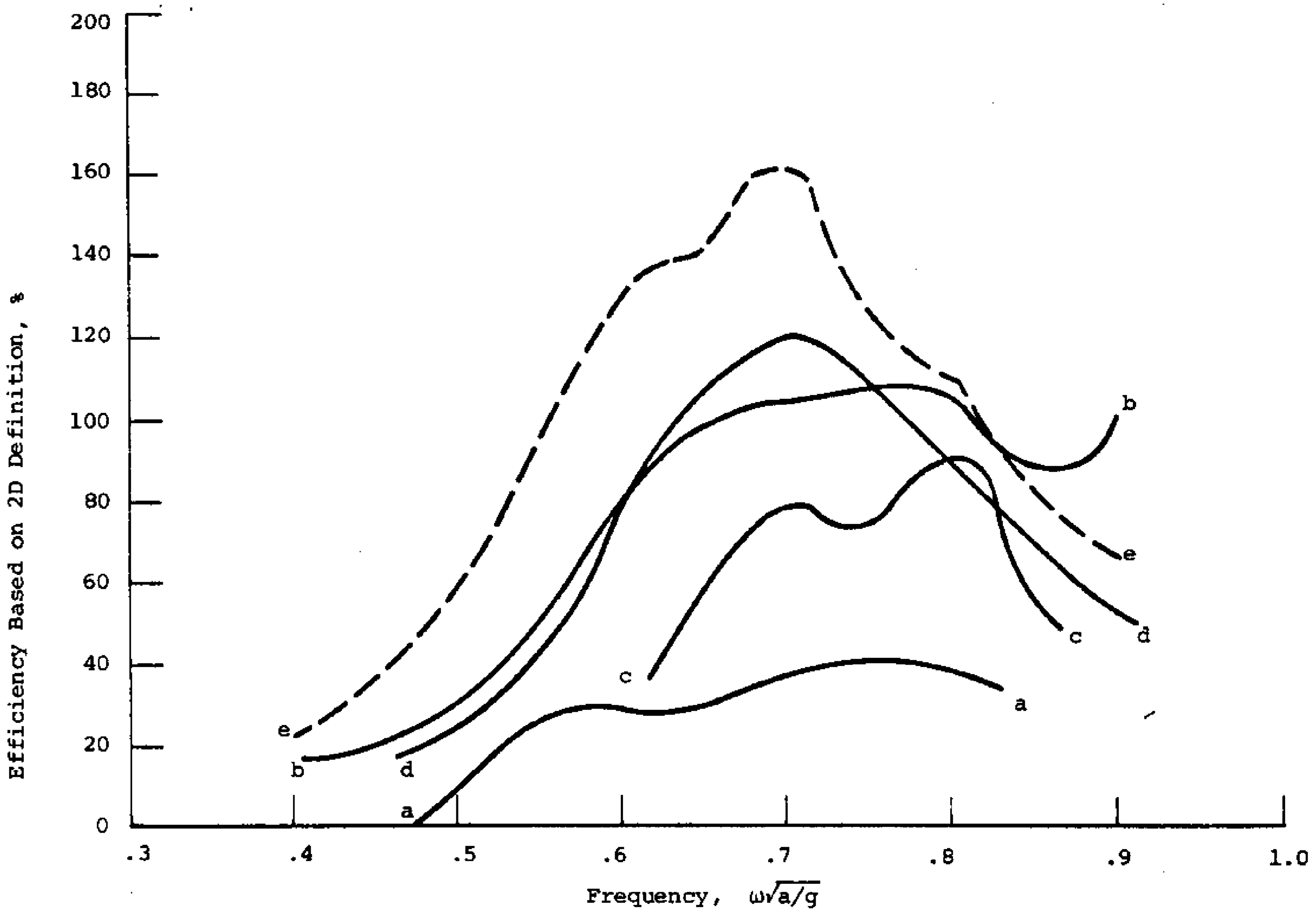


FIGURE 5.5 Efficiency of Moored Cam Systems

a. slack moorings, b. 8 chain taut moorings, c. single taut mooring,
d. 4 chain taut mooring, and e. rigid fram

The initial tautline mooring had eight chains (Figure 5.4b) to hold the support frame of the cam more-or-less rigidly in roll and heave. The results were good although some motion remained and the efficiency level was below the level for the rigid frame, Figure 5.5.

A simpler tautline mooring with a single chain was devised (Figure 5.4c). The heave motion was small but the frame supporting the cam was not restrained in roll and there was an undersirable response in the operating range. The offending mode had the frame acting as a rigid pendulum and the hinge at the point where the mooring chain was shackled to the frame.

A four chain tautline mooring was then devised to constrain the motion of the cam in heave and roll, Figure 5.4d. The chains were attached to the frame from moment arms on the frame. The performance with the four chain mooring was slightly better than with the eight chain system over most of the frequency range, Figure 5.5. Full characteristics of the cam performance with the four chains mooring system are presented in Figure 5.6.

Attempts to simplify the four chain mooring arrangement were unsuccessful as these systems allowed the frame supporting the cam to respond to the wave forces; this invariably reduced the efficiency of cam as a power absorber.

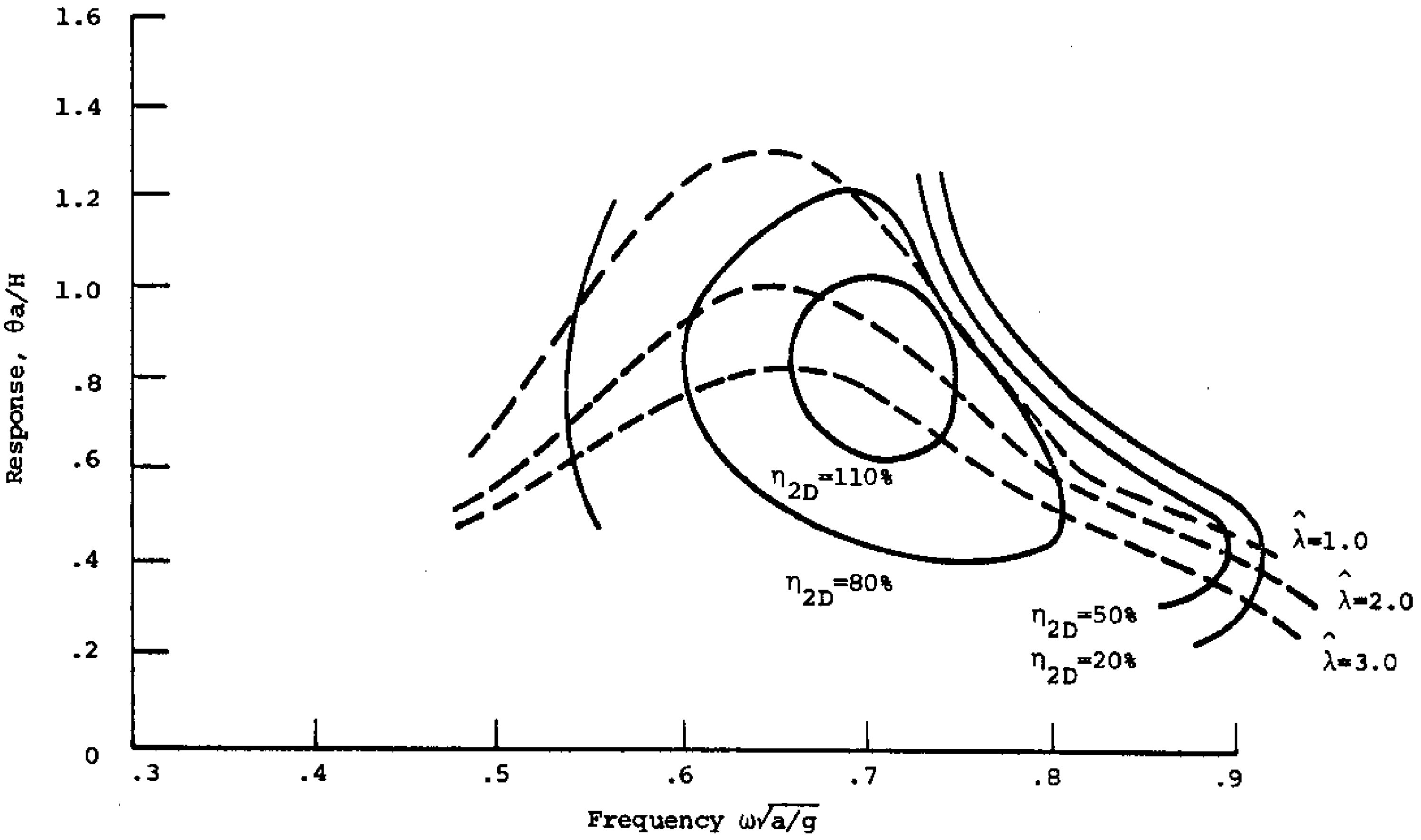


FIGURE 5.6 Performance Characteristics of the Cam Model in the Towing Tank, 4 Chain Taut Mooring

Testing in Random Seas

Tests were conducted with the fixed frame and with the four chain tautline mooring in simulated random seas. The absorber was set at the level which gave maximum efficiency, based on preliminary experiments. The friction at the seals of the cam had the undesirable effect of eliminating the motion, (and hence the absorbed power) of the smaller waves. The measured efficiency was therefore smaller than was anticipated from the tests in regular waves.

The efficiency of the cam in random seas is presented in Figure 5.7 for the fixed frame and the moored cases. The frequency plotted is that of the value at the peak of the spectrum. It can be seen that there is only a small difference between the performance with a rigid frame and with a tautline mooring.

Observations during the tests with random seas suggested that the power output in the ocean would be extremely variable, with relatively long times (several wave periods at the period corresponding to the peak of the spectrum) during which no power would be developed. This suggested that an energy storage system capable of providing continuous power for at least one or two minutes in the full scale situation would be required to provide a suitable wave power source.

Tests were conducted in simulated storm conditions to determine how the cam would operate. The cam was moored with four weights each of which was about one third of the displacement of the cam in simulated sea state 7 (significant wave height equivalent to 4m). Under these conditions, with the power absorber set at level for maximum efficiency, it was observed that the cam would occasionally reach the stop which had been placed to prevent it turning completely over. In addition the two mooring weights ahead of the cam were sometimes lifted slightly. The motion of the cam was not excessively vigorous, nor did it appear to be destructive.

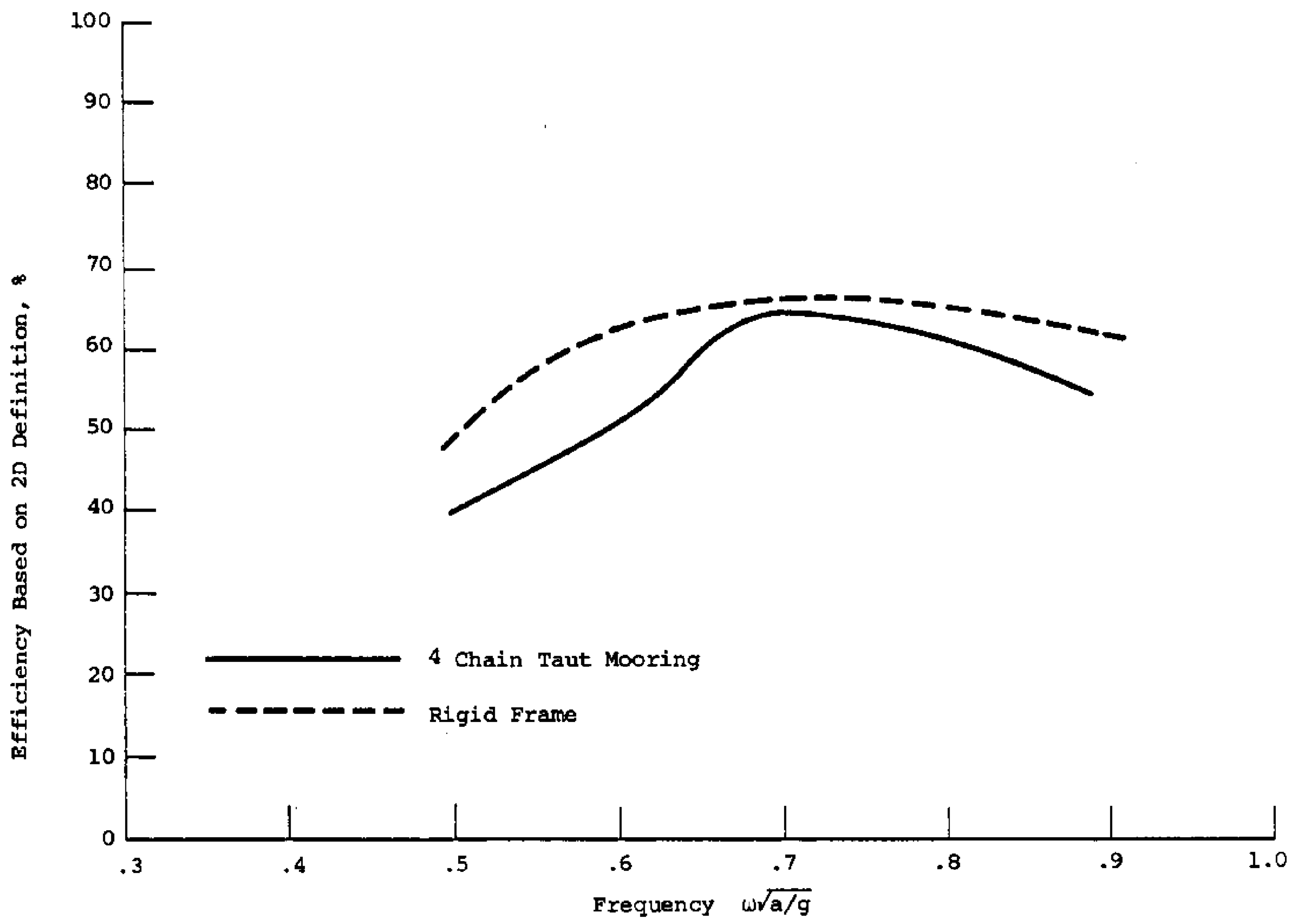


FIGURE 5.7 Performance of the Cam in Random Seas

6. ENGINEERING EVALUATION OF SALTER CAM SYSTEMS

The two-dimensional and simulated three-dimensional tests carried out with a model of the Salter cam have provided information which can be used in a preliminary engineering evaluation of a wave energy conversion system. Additional data, such as wave power statistics at various sites, construction costs, energy conversion methods, and cable costs are obviously also required to facilitate a preliminary design and to estimate the probable cost of the system and its operation.

Salter has advocated the use of a long string of cams (ducks) with a back bone or spine, for ocean applications. It was decided at the outset of this study that the Salter concept was perhaps for rather large average power outputs and would require the solution of formidable engineering problems associated with response of the string of cams and the strength of the spine at each cam. Instead, for this engineering evaluation, it was decided to consider a single cam with an energy conversion system inside the device, a suitable mooring system, and an electrical cable to shore connected to one or more cams.

A taut mooring system similar to that investigated experimentally was chosen, Figure 6.1.

PREDICTED POWER OUTPUT

The overall performance of the wave energy conversion system depends on the wave power at the site, the performance of the cam, and the characteristics of energy conversion arrangements within the cam.

The performance of the cam system was calculated for Nauset Beach, Cape Cod, Massachusetts. From the estimated performance for this site it was assumed that the performance could be scaled for many coastal (particularly East Coast) sites, with good

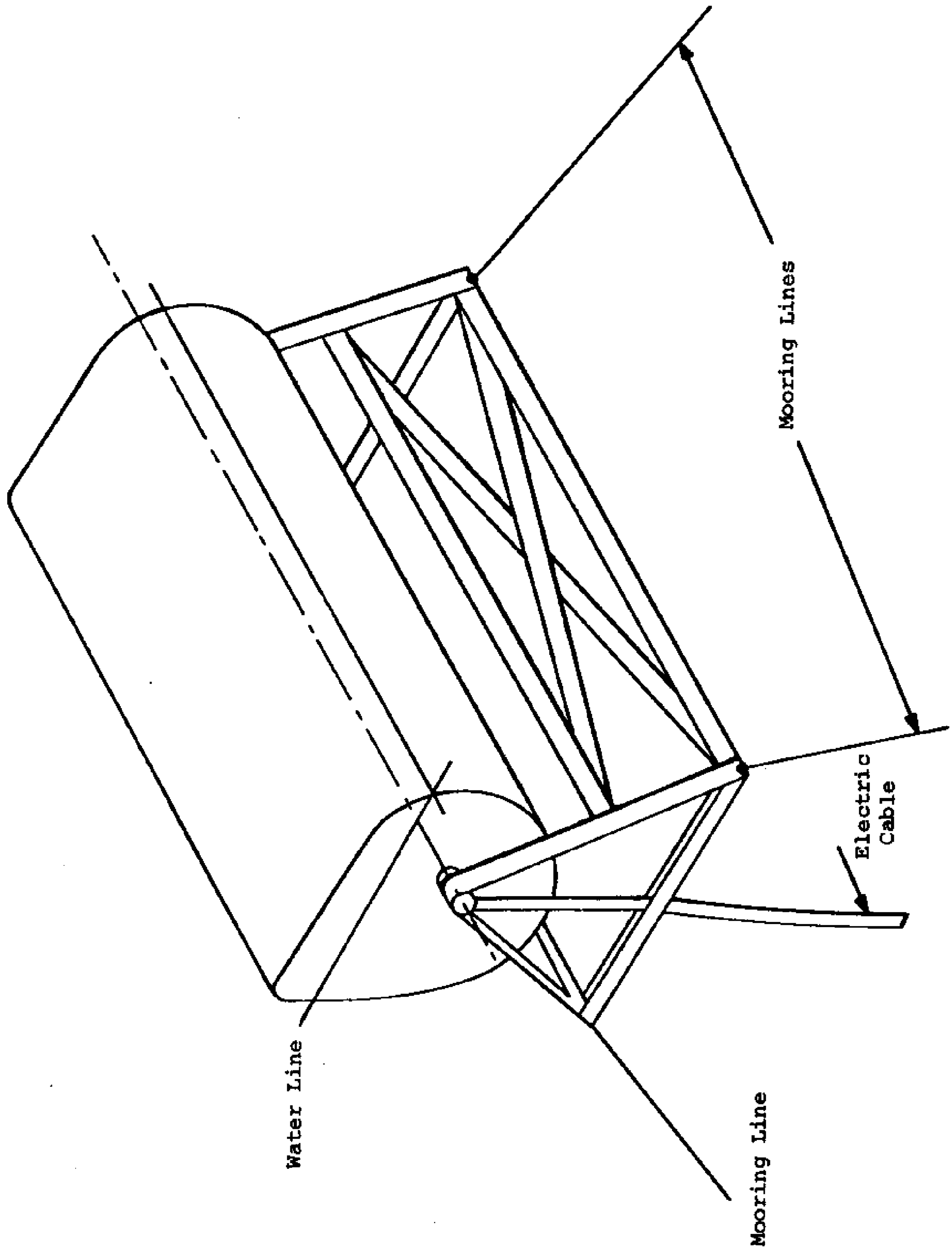


FIGURE 6.1 Possible Arrangement of a Moored Cam

accuracy. In addition, it was anticipated that the performance of cam systems for some open ocean sites could be scaled with acceptable accuracy. These assumptions imply that the annual distributions of wave heights and wave periods are similar at the various sites, scaled only for average power level and average wave period.

Site Wave Data

At Nauset Beach, Cape Cod, there was a bottom mounted pressure gage about 1.3 Km offshore in 10 meters of water. Linear wave theory had been used to translate the pressure variations into wave heights. It was assumed that the data taken for the shallow depth of 10m would be representative of more practical sites where the water is deep enough to accommodate cams of radi 6 to 10 meters.

The distributions of incident wave power for the Nauset Beach site are presented in Table 6.1 (Dingwell, 1977). It can be seen that much of the energy is concentrated in waves having periods of 8 - 10 seconds with the average of 8.5 seconds. The annual average power is approximately 8 kW/m.

Cam Performance

The performance characteristics of the model cam in random seas, with a taut mooring are presented in Figure 5.7. The results can be scaled to any size of cam. The performance characteristics for the full size cam are based on the assumption that the system can be operated at optimum power; this corresponds to optimum damping in the model tests.

Energy Conversion

The process of converting the power absorbed by the cam into electricity is a challenging engineering problem. Salter

Period T,s

Significant Wave Height, H_s, ft

	0-1	1-2	2-3	3-4	4-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12-13	13+	ROW TOTAL
0.5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
1.5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
2.5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
3.5	0.00	1.21	3.17	1.21	0.63	0.24	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
4.5	0.00	0.29	0.39	0.51	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	5.46
5.5	0.00	0.14	1.05	1.57	2.15	0.87	0.81	0.40	0.00	0.00	0.00	0.00	0.00	0.00	1.18
6.5	0.00	0.15	1.14	2.45	1.22	1.11	0.00	0.53	0.00	0.34	0.00	0.00	0.00	0.00	7.16
7.5	0.00	0.05	0.53	0.75	1.74	0.36	0.87	1.53	1.18	0.00	0.00	0.00	0.00	0.00	7.50
8.5	0.00	0.23	1.25	2.30	2.60	3.92	5.27	5.14	2.77	3.23	1.50	0.00	0.00	2.03	7.07
9.5	0.00	0.24	1.11	1.41	1.22	2.07	1.49	0.52	2.77	2.71	1.03	0.00	1.78	2.07	30.23
10.5	0.00	0.50	1.09	1.65	1.78	0.65	0.92	0.64	0.00	0.93	0.00	0.00	0.00	0.00	10.50
11.5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	5.37
12.5	0.00	0.41	1.36	1.45	1.21	0.37	1.61	2.14	0.89	1.11	0.00	0.00	0.00	0.00	0.00
13.5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.60	0.00	0.00	0.00	0.00	0.00	10.57
14.5	0.00	0.24	0.00	0.79	0.21	0.61	0.40	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
15.5	0.00	0.00	0.00	0.00	0.50	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	7.20
16.5	0.00	0.12	0.00	0.44	0.22	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
17.5	0.00	0.00	0.00	0.20	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.78
18.5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
19.5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
20.5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
21.5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
COLUMN TOTAL	0.00	3.66	11.10	14.53	13.00	10.21	11.37	10.92	7.60	8.91	2.52	0.00	1.78	4.10	

THE AVERAGE POWER OVER THE PERIOD MEASURED IS 2.42 KILOWATTS/ft, 8 kW/m

TABLE 6.1

Available Power Distribution for Nauset Beach, Cape Cod in Percent of Total Power originally proposed a hydraulic system to convert the rocking motion of the cam into rotary motion to drive the electric generator. He proposed a rotary pump capable of operating with small angular motions.

Baux (1978) has also concluded that a hydraulic system should be used to drive the generator. A multicylinder hydraulic pump and a swash plate motor were proposed to drive the generator. Baux used information from conventional pumps and motors to predict the performance characteristics of his proposed system. The swash plate motor would be controlled to provide constant rpm drive to the generator. Predicted characteristics based on Baux proposed system are presented in a non-dimensional form in Figure 6.2. It can be seen that the predicted efficiency is a function of the angular amplitude and the wave period. The design amplitude and design period would be selected to optimize the overall power output from the cam and converter.

Predicted Overall Performance

The overall performance of the cam and conversion system was computed for the Nauset Beach site and then scaled for an offshore site. The wave data from Table 6.1 was first combined with the predicted performance of the hydraulic/electrical energy conversion system within the cam (given in Figure 6.2) in order to establish an average conversion efficiency of the conversion system. At the best matching point the average conversion efficiency was found to be 72%. The average output of a cam system at the Nauset site is therefore reduced from the available average power from the waves of 8kW/m to the following:

$$\text{Average power output} = 5.76\ \eta_{cam},\ \text{kW/m} \qquad (6.1)$$

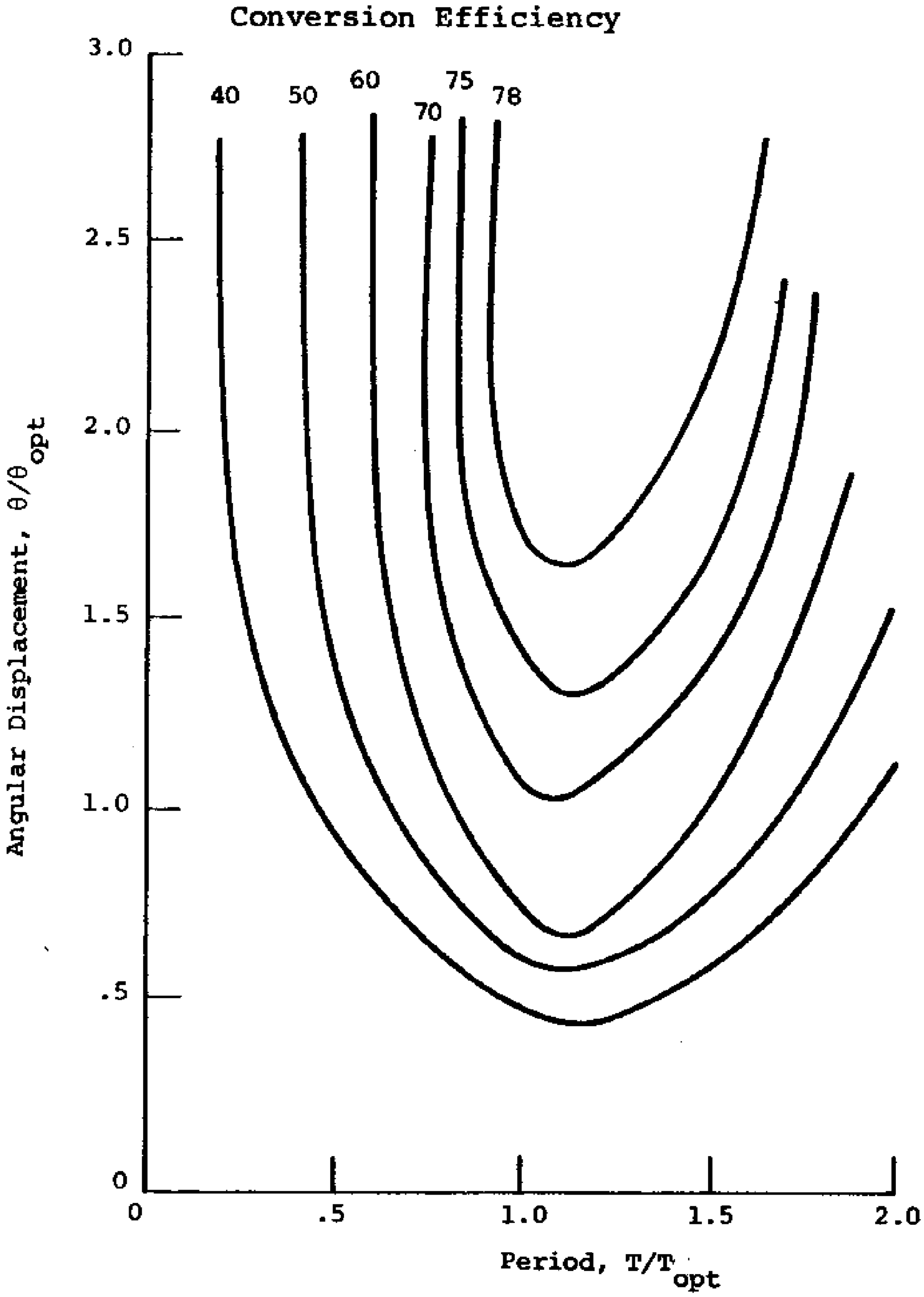


FIGURE 6.2 Predicted Performance Characteristics for the Energy Conversion System

The value of the cam efficiency, η_{cam}, depends on non-dimensional wave frequency, $\omega\sqrt{a/g}$, as shown on Figure 5.7; it is therefore a function of cam size at the selected site.

The peak efficiency of the model cam in random waves occurs at $\omega\sqrt{a/g} = 0.7$. At the Nauset Beach site the average wave frequency, ω_p, is 0.74, corresponding to a period of 8.5 seconds, hence the radius of the cylinder, a , for maximum efficiency is 8.8 m. The cam efficiency is obviously reduced when the size of the cam is both increased and decreased from this value. Cams of the size for maximum efficiency and smaller were examined for this study. Larger cams would obviously be uneconomical.

The performance of the cam system was calculated for the Nauset Beach site assuming that the cam and conversion equipment were capable of absorbing the power from the largest waves. This implied that the electrical conversion system was rated at the maximum power output of the waves; this is very costly. In addition the performance was recalculated using the assumption that the device was rated at a lower value, so that it was not able to generate the power from the largest waves, however, the conversion costs would be lower. The calculations were carried out for wave height limits of 3, 2.5, and 2 times the average wave height at the site. The annual average power output from the cam system for these limits is presented in non-dimensional form in Figure 6.3.

The performance of the cam system at an ocean site, 200 km from the coast, was scaled from the performance data calculated for the Nauset Beach site. At the North Atlantic ocean site the average annual value of the incident wave power given in Table 2.1, was 37.1 kW/m and the average wave period was 6.3 seconds. The cam cylinder radius, a, for maximum efficiency at the ocean site was 4.8 m.

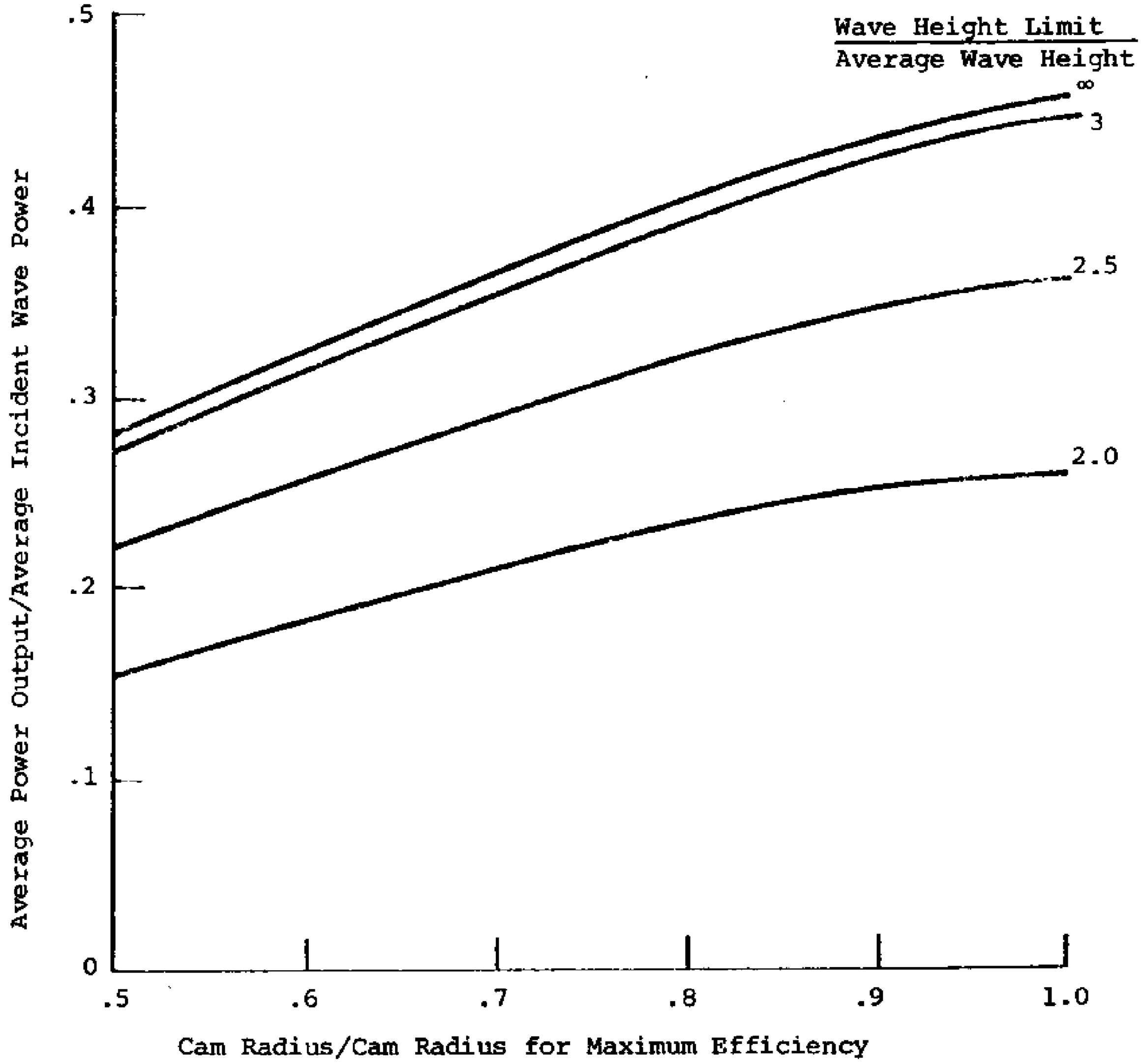


FIGURE 6.3 Electrical Power Output Characteristics

COSTS OF THE WAVE POWER SYSTEM

The capital investment of the cam system would include the cost of the structure, conversion equipment, mooring, and electrical power cable. The annual cost of operating the system would include the maintenance and insurance costs.

Structural Costs

The cam structure was analyzed as a vessel using ship structural strength methods. For simplicity it was assumed that the cam was floating with the lobe in the position to give maximum wave and still water bending moments and minimum modulus. This would be a more severe case than the typical operating arrangement.

The length of the cam was selected as that length which would require an effective plate thickness of 1 cm to support the bending moments. An effective thickness of 1 cm was assumed to be the minimum thickness plate that would be used for a cam in the ocean. Longer cams would obviously require thicker plates. The weight estimate for steel was increased by 40% above the minimum steel weight to allow for framing, bulkheads, and ends. The supporting structure for moorings and cam axle frame was assumed to weigh 10% of the basic cam weight, because the frame can be designed using mainly tension members.

The cost per tonne (1000 kg) of welded steel depends on the complexity of the structure. It is anticipated that the cam and associated structure would be considered as a relatively simple fabrication task costing about $1500 per tonne. Dingwell (1977) assumed 562$/ton and Bisceglia (1978) assumed 1071$/ton. Cam dimensions, weights, and steel costs are presented in Table 6.2.

Table 6.2 Predicted Steel Costs for the Cam Structure

	$\omega\sqrt{a/g}$	.50	.55	.60	.65	.70
Nauset Beach	Radius a, m	4.5	5.4	6.5	7.6	8.8
	Length, m	67	73	79	85	91
	Displacement, tonnes	5,326	8,357	13,103	19,273	27,665
	Steel Weight, tonnes	272	356	464	583	723
	Steel Cost, 10^3\$	408	534	696	874	1,084
Ocean Site	Radius a, m	2.5	3.0	3.5	4.2	4.8
	Length, m	51	55	60	65	69
	Displacement, tonnes	1,251	1,943	2,885	4,501	6,241
	Steel Weight, tonnes	115	149	190	247	299
	Steel Cost, 10^3\$	172.7	223	284	370	449

Cost of Conversion to Electricity

The cost of providing a hydraulic pump and motor to drive the generator is difficult to establish because such a system has not yet been developed. Bisceglia (1978) concluded that the cost of hydraulic systems of about the size contemplated would be 64$/kW. Dingwell (1977), using data from a windpower study, expected the cost of the hydraulic drive to be 130$/kW at 500 kW and 120$/kW at 1500 kW.

The cost of electrical generators designed for 3600 rpm were given by Bisceglia as 60$/kW while Dingwell indicated values of 88$/kW for 500 kW generators and 52$/kW for 1500 kW generators.

In this study the values used were those selected by Dingwell have been used because they are more realistic. The cost of energy conversion was assumed to follow the general form of power equipment, in terms of the rated power, P_{max}; namely an equation of the type

$$\text{Cost} = A\, P_{max}^{n}$$

Using Dingwell's data the equation was

$$\text{Cost} = 795\, P_{max}^{-.21}\ ,\quad \$/\text{kW where } P_{max} \text{ is in kW.} \tag{6.2}$$

The Cost of Electrical Transmission

The cost of electrical transmission line to the shore was expected to depend on the power level carried and whether a-c or d-c is being carried. The selection of alternating or direct current is influenced by the distance that the power is transmitted. Over short distances a-c would be selected while d-c is preferred for long distances.

Wechsler et al (1974) suggested that the cost of d-c cables with auxiliary equipment and including installation would be 31.1/M, \$/kW km, where M is the maximum power carried by the cable in megawatts. The implication of the equation is that cable costs are 31,100 \$/km, independent of the power carried. For the purposes of this evaluation it will be assumed that costs of the cable will be 40,000 \$/km, to allow for inflation.

Mooring Costs

The costs associated with mooring the cams include the cost of materials, costs of placing the mooring, and the towing costs. It is very difficult to estimate these costs as they are site dependent and there is very little information in the published literature that can be used. It has been assumed that the cams would be mooring in relatively shallow water ($<$ 200 feet).

The cost of the mooring materials was based on the cost of the largest manufactured chain to provide a taut mooring with each leg of the mooring 500 feet in length. With a taut mooring arrangement the maximum load is a combination of buoyancy lift of the cam and the wave force acting on the cam. The buoyancy lift force depends on the design value of the tension in the taut mooring and would be about the same magnitude as this design value. Davis (1978) measured the maximum wave forces (in sway) on the model cam in the towing tank and concluded that at maximum load they were given by

$$\text{Force} = 0.035\ \rho L a H g \qquad (6.3)$$

where

H is the wave height

a is the cam radius

L is the cam length

It is obvious from this equation that the wave forces in sway, even with very large wave heights (20m), are much smaller than the buoyancy lift force, which is about half the displacement. The sum of the tensions in the chains were therefore assumed to be equivalent to the displacement of the cam, to allow for the chain geometry. Costs of chain were obtained from the manufacturers and found to be equivalent to approximately 1500 $/tonne of chain.

The costs of placing the moorings were very difficult to determine. From discussions with consultants in the field it was concluded that piles rather than anchors would be used and the placement cost for cams of the size considered here would be about one million dollars. This figure was assumed to include the cost of towing the cam to the site. The cost of providing moorings for the various sizes of cams is presented in Table 6.3. It appears that providing moorings for the cams with a taut mooring design is a formidable problem because the steady forces are very large.

Table 6.3 Predicted Mooring Costs for Cam Systems

	$\omega\sqrt{a/g}$	.50	.55	.60	.65	.70
Nauset Beach	Radius a, m	4.5	5.4	6.5	7.6	8.8
	Displacement, tonne	5,326	8,357	13,103	19,273	27,665
	Mooring cost, 10^3\$	1,353	1,554	1,868	2,277	2,833
Ocean Site	Radius a, m	2.5	3.0	3.5	4.2	4.8
	Displacement, tonne	1,251	1,943	2,885	4,501	6,241
	Mooring cost, 10^3\$	1,082	1,129	1,191	1,298	1,414

Other Capital Costs

The costs of special equipment, such as welding jigs, were not included in the cost of steelwork because it was concluded that the cam form was relatively simple to fabricate. The cost of corrosion protection was assumed to be a part of the steelwork costs.

Total Capital Investment for the Wave Energy System

The methods of predicting the equipment and installation costs described in the previous paragraphs have been used to predict the total capital investment for the wave energy systems.

The cost per kW of average power output was obtained utilizing the predicted costs and predicted cam performances at the Nauset Beach site but three kilometers from the shore, and at a site 200 kM from the shore as described earlier.

The costs at the open ocean sites were high because of the high cost of providing cables to the shore. In view of the high cable costs it was decided to estimate the costs for five cams feeding electricity into a single cable.

The predicted costs of the cam system at the Nauset Beach site and at 200 km from the shore are presented in Figure 6.4. The costs are presented as functions of cam radius and limiting wave height. It was assumed that the power conversion equipment would be rated at the power level corresponding to the limiting wave heights. The minimum predicted capital investment costs for the various sites are presented in the accompanying table.

Table 6.4 Minimum Capital Investment Cost for Wave Energy Conversion Systems

Site	Power kW/Cam	Radius, m	Rated Power / Average Power	Cost $/kW
Nauset Beach	282	7.6	9.0	12,967
Ocean	1,143	4.8	9.0	10,667
Ocean, 5 Cams	1,143	4.8	9.0	4,305

Operating Costs

The main annual costs of operation for wave energy systems would be the maintenance costs for the moorings, energy conversion system, and the steelwork. In addition there would be insurance.

The maintenance cost on the cam and equipment was expected to be very much higher than land based systems and a value of 8% of the capital cost (including spares) was assumed. The insurance

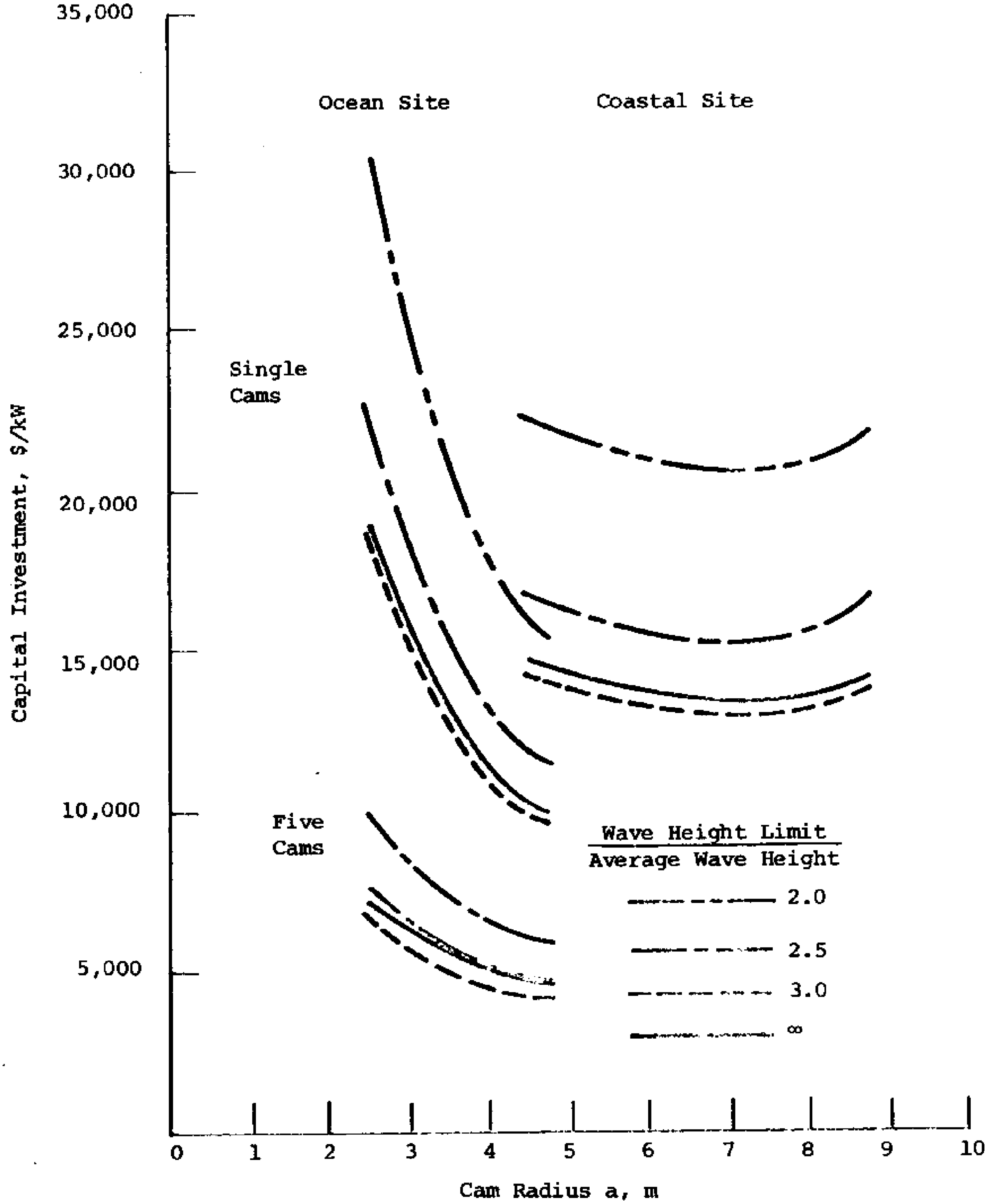


FIGURE 6.4 The Predicted Capital Investment Costs of Salter Cam Wave Energy Systems

would also be high, and a figure of 2% was selected. A total of 10% of the capital and installation cost was therefore assumed for the annual operating costs.

Indirect Costs

Utility power plants have a substantial indirect cost, including architect and engineering fees, contingencies, spare parts, and interest during construction, which sometimes total 50% of the direct cost. Such costs are not likely to be significant in wave energy systems because a large proportion of these indirect costs for conventional plants are associate with the fact that each power plant is different. Standardization is expected with wave energy systems. No indirect costs were assumed.

Predicted Cost of Electrical Production

The cost of electricity from the wave energy system will be based on the busbar energy cost (BBEC) given in $/kWh by

$$\mathrm{BBEC} = \frac{(\mathrm{CI})(\mathrm{FCR}) + (\mathrm{AC})}{\mathrm{CAP(avg)} \times 8760} \tag{6.4}$$

where

CI is the capital investment, $

FCR is the fixed charge rate, per year

AC is the annual costs, $/y

CAP(avg) is average annual output, kW

The fixed charge rate was taken as 0.15 (DeMeo and Bos, 1978) and annual capacity factors corresponding to 50, 70 and 90% of the average power were assumed. The costs of electrical power for the optimum designs of Table 6.4 in mills/kWh (10^{-3} $/kWh) are presented in the accompanying table.

Table 6.5 Busbar Energy Cost, Mills/kWh

Capacity Factor	50%	70%	90%
3km from shore 1 cam	740	529	411
200km from shore 1 cam	609	435	338
200km from shore 5 cams	246	175	136

DISCUSSION

The values in Table 6.5 should be compared with the anticipated costs of electric production using fossile fuels in 1995 of 50 - 90 mills/kWh (DeMeo and Bos, 1978).

The capacity factor of new plants would be low initially (50%) because of operational difficulties and because the energy source is variable. In a mixed power system with fossil fueled power plants and nuclear units the capacity factor for the wave energy systems might eventually become high (80-90%) because the wave energy system would have small variable costs.

The use of storage offers greater operational flexibility, but whether or not the additional expense is justified will depend on the cost of providing the storage and on the characteristics of individual utilities. Storage has not been considered in this study.

In predicting the power output of the various cam systems it has been assumed that the crest length of the significant waves are much larger than the cam length and also that the waves approach the cam normally. The information on crest length suggests that representative lengths of the crests are about three times the wave lengths. The crest lengths of the significant waves in the North Atlantic should therefore be about 200 meters. Hence a 50-80 meter cam would probably have acceptable performance and capture most of the energy in the waves.

The direction of the significant waves relative to the cam depends on the site and, to some extent, on the type of mooring. In addition the influence of the direction of the waves on cam performance depends on the length of the cam. A cam which as a length much shorter than the wave length was demonstrated experimentally to operate at 90° to the incident waves with about 50% of the efficiency of the cam operating normal to the incident waves. This suggests that a 50 - 80 m cam is perhaps too long for a site where there is appreciable change in prevailing wave direction throughout the year.

The influence of seas with short crest length and changes in wave direction will certainly reduce the electrical output of Salter cam devices. There is insufficient experimental evidence to determine the magnitude of these power reductions. However, they could easily result in a 50% reduction in average power output and this would raise the cost of electrical power by 100%.

In addition there is considerable uncertainty about the various costs of the device. The most significant uncertain costs and probably the most serious engineering problem would be associated with mooring arrangement. The magnitude of the steady forces in the taut mooring and the difficulty of providing suitable anchors to sustain those forces will probably make the Salter wave energy device unattractive, at lease as a single unit.

7. CONCLUSIONS

The experimental study of a model of a Salter cam in a two dimensional wave channel demonstrated that in regular waves the efficiency of the device was indeed high. The wave frequency for peak efficiency was close to the natural frequency in roll of the device, and this depended on the geometry of the cam and on the moment of inertia of the device about the fixed axle. At each wave frequency and wave height there was a level of power output (damping) which maximized the efficiency of the device.

The performance of the model cam in the towing tank provided a test of the device in conditions more nearly representative of the ocean. With regular waves and the cam held rigidly the power absorbed was greater than the power in a two dimensional strip the width of the cam. This result was not satisfactorily explained.

With the model freely floating there was a considerable reduction in power output because the device was responding in heave, sway, and roll. It was found that the performance could only be improved to the level approaching that of the fixed cam by using taut moorings. Taut moorings were investigated and a suitable mooring arrangement was developed.

The cam model was tested in the towing tank in simulated random seas. The measured efficiency was considerably reduced compared with the performance in regular seas.

The model tests in random seas were used to predict the performance of a full sized cam in the ocean. Wave data for a coastal site (Nauset Beach, Cape Cod) and an ocean site in the North Atlantic were utilized to predict the average power output of cams having a range of sizes.

The cost of fabricating the cam, providing the energy conversion, mooring the device, and providing a cable to shore were estimated. The predicted performance and costs were utilized to estimate the cost of electricity at the shore.

The costs of power production from waves at the coastal site were predicted to be prohibitively high. More reasonable electrical power costs were predicted for an ocean site, provided several cams could be serviced by a single electrical cable. However, the predicted costs of electrical production were about 140 - 250 mills/kWh, which is considerably larger than predicted costs of electricity produced by more conventional methods. There was considerable uncertainty about the cost and power predictions because of the novelty of the arrangement.

A general conclusion of this study (and this has application to other wave energy conversion systems) is that the conversion efficiency of the device is not really the most important parameter. The more important technical and economic considerations are associated with the feasibility of conversion into electrical power, the strength of the device to withstand storms, the provision of a satisfactory mooring arrangement, and the construction of a cable to bring the electrical power ashore.

REFERENCES

Baux, J. F. P.	1978	The Feasibility of Wave Energy Converters, S.M. Thesis, Department of Ocean Engineering, MIT.
Budal, K.	1977	Theory for Absorption of Wave Power by a System of Interacting Bodies, Journal of Ship Research, Vol. 21, No. 4.
Bisceglia, S. V.	1978	Feasibility of Cam Type Wave Generators, Ocean Engineer Thesis, Department of Ocean Engineering, MIT.
Davis, N. B	1978	Performance of a Salter Cam-Type Wave Energy Extractor, An Experimental Evaluation, S.M. Thesis, Department of Ocean Engineering, MIT.
DeMeo, E. A. Bos, P. B.	1978	Perspectives on Utility Central Station Photovoltaic Applications, Report No. ER-589-SR, Electric Power Research Institute.
Dingwell, R. E.	1977	Predictions of Power Production by a Cam Type Wave Energy Converter for Various Locations, M.S. Thesis, Department of Ocean Engineering, MIT.
Hogben, N. Lumb, F. E.	1967	Ocean Wave Statistics, HMSO, London.
Mollison, D. Buneman, O. P. Salter, S. H.	1976	Wave Power Availability in the N.E. Atlantic, Nature, Vol. 263, No. 5574.
Mynett, A. E. Serman, D. D. Mei, C. C.	1979	Characteristics of Salter's Cam for Extracting Energy from Ocean Waves, Applied Ocean Research, Vol. 1, No. 1.

Nath, J. H. Williams, R. M.	1976	Preliminary Feasibility Study for Utilization of Water Wave Energy, ERDA Contract WA-76-3102.
Salter, S. H. Jeffrey, D. C. Taylor, J. R. M.	1976	The Architecture of Nodding Duck Wave Power Generators, The Naval Architect.
Standing, R. G.	1978	Application of Wave Diffraction Theory, National Maritime Institute, England, Report NMI R32.
Thompson, E. F.	1977	Wave Climate at Selected Locations Along U.S. Coasts, U.S. Army Corps. of Engineers, Coastal Engineering Research Center, Technical Report No. 77-1.
Wechsler, L. Brown, C. E. Sundaram, T. R.	1974	Engineering Analysis of Systems for Extracting Useful Energy from the Sea, Tenth Annual Conference, Marine Technology Society.
Wiegel, R. L.	1964	Oceanographical Engineering, Prentice Hall.

NOMENCLATURE

a	cam radius, m
d	water depth, m
C_g	group velocity, equation 2.2, m/s
g	gravitation acceleration, m/s^2
h	wave height, equation 2.6
H	wave height, peak to trough, m
H_s	significant wave height, m (or feet)
I	moment of inertia, kg m^2/m
$\hat{I}$	non-dimensional moment of inertia $I/(\rho a^4)$
L	wave length, m
P	probability
P	wave power, kW/m
$\bar{P}$	average wave power, kW/m
P_{max}	maximum electrical power, kW
P_*	wave power correction for depth, equation 2.5
s	spectral density, equation 2.11, ft^2s
T	wave period, s
$\bar{T}$	average wave period, s
V_k	wind velocity, knots
z	variable in equation, 2.6, m
ε	cam efficiency
θ	angular movement of cam, peak to trough, radians
$\hat{\theta}$	non-dimensional angular response, $\theta a/H$

λ	damping coefficient, N ms/m
$\hat{\lambda}$	non-dimensional damping coefficient, $\lambda/(\rho a^4\sqrt{g/a})$
ρ	water density, kg/m^3
ω	angular frequency, radians/s
$\hat{\omega}$	non-dimensional frequency, $\omega\sqrt{a/g}$
ω_p	angular frequency at spectrum peak, radians/s