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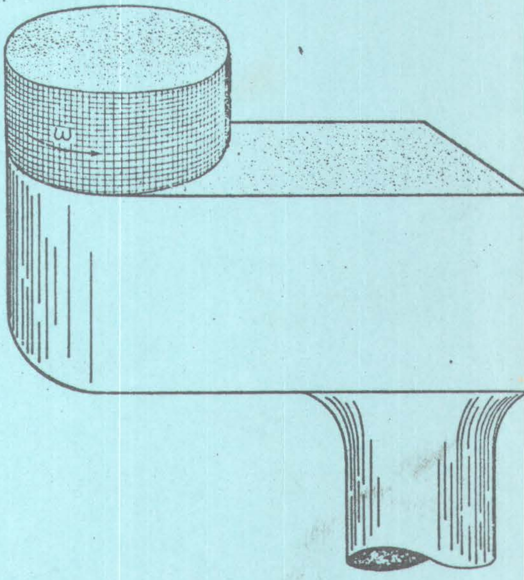
SEA SOLAR PLANTS
A FEASIBILITY STUDY

Ocean Engineering Design Project



Ocean Engineering Programs
School of Engineering
OREGON STATE UNIVERSITY
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SEA SOLAR PLANTS
A FEASIBILITY STUDY

Ocean Engineering Design Project

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FORWARD

This publication is the accumulation of the individual final reports of the thirteen students involved in Oregon State University's Ocean Engineering Design Class from March 27 to June 5, 1974.

Ocean Engineering involves the application of the tools of all traditional engineering disciplines for the planning, design, construction, inspection and maintenance of systems used with and for the development and protection of the oceanic environment. A relatively new and extremely challenging area within this broad field is the economical extraction of energy from the sea. The storehouses for this energy within the sea include currents, waves, and thermal gradients among others. The Ocean Engineering Design class for this term limited study solely to thermal energy conversion.

Student backgrounds entering the class included Ocean Engineering, Civil Engineering, Mechanical Engineering, Economics, Oceanography and Marine Resources Management. This diversity of background and experience provided for a full range of student viewpoints which greatly enhanced the advancement of our technical understanding of thermal energy conversion concepts. With few exceptions, the students had no prior exposure to sea thermal energy conversion and had to build individually into the problem from their own background.

The unifying goal within the class was to establish whether or not a Sea Solar Power Plant concept was feasible, and, if so, whether or not the concept could be made practical. More specifically, the class divided into two basic groups; the first concerned with the feasibility of physically sizing, constructing, operating and locating the plant, while the second concerned itself with the impact of the plant on the local environment and with the impact of that environment on the plant.

Throughout the short study period, time was our most precious resource. The procurement of outside lecturers, literature searches, and even individual tasks were adopted, initiated, and ended based on a productivity per unit time input criterion. This rather unique project set-up prompted the initiation of the PERT analysis of our efforts by Allen Boyce.

This then was our general approach for a first step into a very large field. It is hoped that this study will help to form a base for future studies at OSU and elsewhere to expand and refine our findings.

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Many thanks are also due to Miss Janice Baker and Mrs. Susan Ellinwood for their excellent typing of this report, and for their help in its organization and publication.

Finally, our deepest thanks are extended to Dr. Larry Slotta, the class project director. His continued help with, and criticism of our ideas and concepts contributed more than any other single factor to our overall learning process and in particular to our understanding of the sea thermal power problem.

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by

Allen Boyce

ABSTRACT

A Procedure Evaluation and Review Technique (PERT) Analysis is applied to the conceptual design of a Solar Sea Power Plant (SSPP) because of the limited time within which the project has to be completed. A PERT analysis rather than a Critical Path Method (CPM) was used because of the variability in time duration of activities required to complete a task. A Probabilistic Factor is also applied to some activities to account for repititions that may be required in an effort to reach an optimal design. A time chart is constructed and manpower leveling applied to estimate the least possible duration of the project and best allocation of manpower to required activities.

SEA SOLAR POWER PLANT - PERT ANALYSIS of PLANT FEASIBILITY STUDY

by

Allen Boyce

INTRODUCTION

The tasks undertaken in the Sea Solar Power Plant Feasibility Study are by no means intended to exemplify all those tasks required for the design and construction of a Solar Sea Power Plant. Tasks were chosen scrutinously in an effort to concentrate on those areas of greatest interest and concern, yet still allowing for the conceptual design as a whole. The initial tasks chosen were as follows:

1. A Study of Thermal Cycle requirements for low temperature gradient power production.
2. An investigation into different possible working fluids.
3. A possible power output evaluation.
4. A complete design of Heat Exchangers.
5. An analysis of pumping requirements.
6. A size and configuration design.
7. Stationing on position design.
8. Design of screening devices.
9. Biofouling Prevention Methods study.
10. Energy trade-off analysis.
11. Maricultures Study.
12. Environmental Impact Statement.

These tasks were broken down into component activities to give the level of control necessary for management of the project.

A three week orientation and organiaation period is included on the PERT network chart as an informational sidelight. This period was not considered as a part of the PERT analysis. The activities were:

A. Introduction to project by Dr. Slotta

B. Thermodynamics lecture by Dr. Zaworski

C. Heat Transfer lecture by Dr. Larsen

D. Wave Forces & Mooring design lecture by Dr. Nath

E. Large Volume pump design lecture by Associate Professor Tom Heinecke

F. Literature review of publications by Massachussetts University and Carnegie-Mellon Institute of Technology on the subject

G. Organization period

PERT ANALYSIS

A PERT Analysis is applied to this project because of the indeterminacy of exact times for the duration of activities necessary for the completion of the project. As an example of the typical situation encountered, consider the task of "sight selection." In order to find a sight suitable for power production with the once through cool system of a solar sea power plant, it is necessary to first situate the plant in an area where an adequate temperature gradient exists between the surface warm water and lower depth cool waters. It is also necessary to have adequate depth to keep the cool water uptake a distance from the bottom without exposing the primary structure to the influence of surface currents and wave forces. Once these two criteria have been met it is then necessary to establish the nearest pop$\underline{\text{n}}$ centers and power transmission requirements. It may happen that there are no areas within the regions established by criterias 1 and 2 which meet 3. It is then necessary to reevaluate and adjust the initial criterion for 1, 2, and 3 and begin the sight selection process over again or continue on the basis of the reevaluation. The readjustment of the criterion for any activities that are dependent on 1, 2, or 3 may also be necessary. Even though initial design criteria were established, they are of necessity prone to alteration. The time duration for completion of any activity or group of activities comprising a task has therefore, become indefinite as has the time duration for completion of the entire project. This is typical of a research oriented project delving in a new field. A CPM analysis requires exact estimates of activity times which is not possible here. Also, a PERT analysis allows for the application of probabilistics.

PROBABILISTIC FACTORS

There are several places in the design where loops are encountered. This is caused primarily by the dependency of the power output on the sizing of subsequent units. To site the major example the pumping power requirements are dependent on the heat exchanger sizes which are dependent on the power output desired. Therefore, for a given net power output we must increase the gross power production of the plant by the amount of power required for pumping which in turn, increases the heat exchanger sizes and the pumping requirement. This interdependency contradicts a basic assumption of the PERT method which states that the precedence relationships of project activities must be completely represented by a noncyclical network graph in which each activity connects directly to its immediate predecessor (Weist and Levy, 1969). Neither PERT nor CPM allows for conditional activities or cyclical orderings.

To account for the possibility of having to repeat activities a probabilistic factor may be applied to the initial time durations arrived at using the conventional PERT assumptions. It should be noted that the probabilistic factor may not be applied to all the activities within a loop since the need for revamping any designs is dependent on the magnitude of the correction when compared to the overall design criteria. It can be seen that some design aspects within a loop are sensitive to subsequent changes while others are not.

To account for the probability of repeating an activity more than once in coming up with an optimal design "Probability Factors" will be given for each activity. More than one probability factor may be applied to some activities which are contained in more than one loop. In this manner an activity duration may be arrived at without distorting the initial time duration estimates arrived at using the convention PERT assumptions.

Another alternative is to simply lump dependent activities into one activity and/or increase the pessimistic time estimate for the activity(s) duration. Neither alternative is considered viable because the relationship of the schedule to the actual process being modeled is lessened.

It is more commnn to find activity interdependencies existing because of lumping activities together which affords the simple remedy of breaking the activities into smaller tasks.

Exact probability factors were resolved from estimates given by the individuals working on those phases of the project within each loop. The procedure was to estimate as close as possible the least number of itterations required to insure a nearly 100% chance of completion. If some doubt did exist, but was not substantial enough to assume a higher number of itterations, the 100% factor was adjusted as can be noted for loop 1 and loop 2. The general probability equation used was arrived at by consideration of conventional applications of probability to problems of this nature. From the general equation the appropriate probability factor 'X' to be applied to each loop was solved by trial and error.

PROBABILITY FACTORS

$$\text{General Egn:} \quad P_n = \sum_{i=0}^{n-1} X(Y)^n$$

X....probability of no itterations
Y....probability of having to itterate
P_n...probability of n itterations

LOOP 1: Activity #30 back to Activity #3

Assume 98% chance of only four itterations therefore, Probability Factor equals .75 by trial & error.

$$P_4 = \sum_{i=0}^{4-1} X(Y)^n$$
$$= .75 + .75(.15)^1 + .75(.15)^2 + .75(.15)^3 = 98\%$$

Activities Affected: 3,9,11,13,14,16, 18,20,22,23,24,30.

LOOP 2: Activity #46 back to Activity #40

Assume 99% adequate design in three itterations. Probability Factor equals .80 by trial and error.

$$P_3 = \sum_{i=0}^{3-1} (.20)^{n-1}$$
$$= .80 + .80(.20) + .80(.20)^2$$
$$= 99\%.$$

Activities Affected: 40, 42, 43, 44, 45, 46.

LOOP 3: Activity #56 back to Activity #40

Assume 100% chance of adequate design in three itterations. Probability Factor = .90.

$$P_3 = \sum_{i=0}^{3-1} .90(.10)^{n-1}$$
$$= .90 + (.90)(.10) + (.90)(.10)^2$$
$$= 99.9\%$$

Activities Affected: 40, 42, 43, 44, 45, 46, 51, 52, 55, 56.

TABLE 1. NOTATIONS

Abbreviation	Definition
H.E.	Heat Exchanger
W.W.	WarmmWater
C.W.	Cold Water
W.F.	Working Fluid
THRM'L.	Thermal
EXT.	Extraneous
PT.	Point
COMB.	Combined
CONV.	Conventional
EMBED.	Embediment
E.I.	Environmental
EFF.	Effect
MARI.	Mariculture

PROBABILITY OF MEETING DUE DATE

General Egn: $Z = \frac{D-Te}{S_t}$

Z.........Probability number
D.........Due Date
T_e.......Sum of time estimates on critical path
S_t.......Standard Deviation of activities on critical path

D = 6 wks x 16 hrs/wk x 4 men = 384 hrs

$T_e = \Sigma Te_{critical\ path}$ = 387 hrs

$S_t = [\Sigma V t_{critical\ path}]^{1/2}$ =

$$Z = \frac{384 - 387}{11.35} = -.23$$

Probability of meeting due date = 40%

The above analysis assumes four of the men working on the project are available to work on critical activities. In reality because of the constraints imposed by splitting into groups to handle different tasks this may be optimistic (see resource allocation constraints). The extreme would be to assume only one man works on each activity. To complete the project in six weeks this individual would have to work 64-1/2 hours per week. The closest estimate to the actual situation is 2-3 men per activity. At three men per activity on the critical path a work load of 22 hours per week is required to complete the project in six weeks.

RESOURCE ALLOCATION

A man power leveling was performed for the time table established from the PERT time estimates in an effort to optimize manpower utilization over the span of the project. The initial group of ten men is split into two groups of five each, a structural design group and an environmental group. The work of the men in these groups is restricted to activities that fall under their specific category. The category which each of the activities comes under is fairly evident. Biofouling and screening were included under the environmental groups tasks. It is also assumed that once a man is assigned toa task he must stay on it to completion--unless two men are already assigned to it. These constraints were made to try to fit as closely as possible the actual conditions of the project. A man is assumed to work about sixteen hours per week.

It was found that over the first half of the project that the structural group carries the majority of the work load while the environmental group waits for needed information. During the later portion of the project the critical activities are shifted from the structural group to the environmental group. The primary reason for this is the prerequisite constraints of the activity.

Through an allocation of manpower a considerable shortening of the project duration can be accomplished. The total time duration for the project was

TABLE 2. PERT TIME ESTIMATES

#	Activity	Activity times t_o	t_m	t_p	Variance V_t	Expected Time	Probability Factor	Adj. Expect Time
1	Pumping Study	30	48	56	18.78	46	--	46
2	Temp. Theory	4	8	11	1.36	8	--	8
3	Power Out	6	13	15	2.25	12	.75	16
4	Temp. Sea	3	4	8	.69	5	--	5
5	Bathemetry	2	5	8	1.00	6	--	6
6	Upwelling	3	7	12	2.25	7	--	7
7	Biofoul Envr.	6	10	13	1.36	10	--	10
8	Working Fluid	20	32	42	13.44	32	--	32
9	H.E. Matr'l	8	11	15	1.36	12	.75	16
10	Inter. Flow	5	9	13	1.78	9	--	9
11	H.E. Surface	14	20	25	2.25	10	.75	13
12	Pop$^{\underline{n}}$ Centers	3	6	7	.44	6	--	10
13	Boiler	16	21	26	2.78	21	.75	28
14	Condenser	16	21	26	2.78	21	.75	28
15	Trans. & Dist.	8	10	11	.69	10	--	10
16	W.W. Flow	3	5	6	.25	5	.75	7
17	W.W. Pipes	2	5	7	.69	5	--	5
18	C.W. Flow	4	6	9	.69	6	.75	8
19	C.W. Pipes	6	8	10	.44	8	--	6
20	W.F. Flow	5	6	7	.11	6	.75	8
21	W.F. Pipes	4	5	6	.11	5	--	5
22	W.W. Pumps	2	2	2	0.00	2	.75	3
23	W.C. Pumps	2	2	2	0.00	2	.75	3
24	W.F. Pumps	3	4	5	.11	4	.75	5
25	Sediment	3	5	7	.44	5	--	5
26	Currents	5	10	12	1.36	10	--	10
27	Shipping	3	5	7	.44	5	--	5
28	Biofoul Growth	12	18	21	2.25	18	--	18
29	Site	11	18	20	2.25	17	--	17
30	Thrml. Cycle	17	24	30	4.69	24	.75	32
31	Boil'r Housing	6	9	15	2.25	10	--	10
32	Condnsr Hsn'g	6	12	17	3.36	12	--	12
33	Support Syst.	5	6	9	.44	6	--	6
34	Biology @ Site	28	36	45	8.03	36	--	36
35	Machinery	4	4	4	0.00	4	--	4
36	Ext. Pipe	4	8	12	1.78	8	--	8
37	Weight	8	14	16	1.78	14	--	14
38	Power/Cost	8	24	30	13.44	23	--	23
39	Bouyancy	4	8	12	1.78	8	--	8
40	Shape	8	12	16	1.78	12	(.80)(.90)	17
41	Maricultures	8	14	16	1.78	14	--	14
42	Waves	7	8	9	.11	8	(.80)(.90)	11
43	Wind	9	10	11	.11	10	(.80)(.90)	14
44	Current Forces	7	8	9	.11	8	(.80)(.90)	11
45	Misc. Forces	8	10	14	1.00	10	(.80)(.90)	14
46	Stability	12	30	42	25.00	23	(.80)(.90)	32

TABLE 2. cont.

#	Activity	Activity Times t_o	t_m	t_p	Variance	Expected Time	Probability Factor	Adj. Expect Time
47	Entrmnt. Fouling	8	10	14	1.00	10	--	10
48	Entrmnt. Biota	8	10	12	.44	10	--	10
49	Biofoul-Struc.	3	5	8	.69	5	--	5
50	Biofoul-Operat'n	5	8	17	4.00	8	--	8
51	Pt. Mooring	10	12	17	1.36	13	.90	15
52	Dyn. Position'g	10	12	17	1.36	13	.90	15
53	Screens	4	12	19	6.25	12	--	12
54	Biofoul Prevent'n	12	18	24	4.00	18	--	18
55	Mult. Pt. Mooring	5	8	9	.44	8	.90	9
56	Comb. Pos. Syst.	6	9	11	.69	9	.90	10
57	Conv. Anchoring	8	12	14	1.00	12	--	12
58	Embed. Anchoring	8	12	18	2.78	12	--	12
59	E.I. Entrainmnt.	4	6	8	.44	6	--	6
60	E.I. Impingement	4	6	8	.44	6	--	6
61	Sea Circulation	3	5	10	1.78	6	--	6
62	E.I. Circulation	6	12	18	4.00	12	--	12
63	Upwell Eff. Mari.	6	12	14	1.78	12	--	12
64	Imping. Eff. Mari.	2	3	4	.11	3	--	3
65	Biofoul Eff. Mari.	3	6	7	.44	6	--	6
66	E.I. Biofoul Prvnt	3	6	7	.44	6	--	6
67	Mari. Struc.	6	12	15	3.36	12	--	12
68	E.I. Mari.	3	4	5	.11	14	--	14
69	E.I. Statement	10	12	18	1.78	13	--	13
70	Bene/Cost Mari.	8	10	15	1.36	11	--	11
71	Write-Up	65	8	108	51.36	80	--	80

TABLE 3. BOUNDARY TIME TABLE

#	ACTIVITY	EARLY START	LATE START	EARLY FINISH	LATE FINISH	FLOAT
1	Pumping Study	0	51	46	97	51
2	Temp. Theory	0	0	8	8	0
3	Power Out	0	24	3	40	24
4	Temp. Sea	0	93	6	99	93
5	Bathemetry	0	93	6	99	93
6	Upwelling	0	244	7	251	244
7	Biofoul Envr.	0	217	10	227	217
8	Working Fluid	8	8	40	40	0
9	H.E. Matr'l	40	40	56	56	0
10	Inter Flow	40	47	49	56	7
11	H.E. Surface	56	56	69	69	0
12	Pop$^{\underline{n}}$ Centers	6	99	12	105	93
13	Bioler	69	69	97	97	0
14	Condenser	69	69	97	97	0
15	Trans & Dist	12	105	22	115	93
16	W.W. Flow	97	100	104	107	3
17	W.W. Pipes	97	102	102	107	5
18	C.W. Flow	97	99	105	107	2
19	C.W. Pipes	97	99	105	107	2
20	W.F. Flow	97	97	105	105	0
21	W.F. Pipes	97	100	102	105	3
22	W.W. Pumps	104	107	107	110	3
23	C.W. Pumps	105	107	108	110	2
24	W.F. Pumps	105	105	110	110	0
25	Sediment	22	100	47	125	78
26	Currents	22	115	32	125	93
27	Shipping	22	120	27	125	98
28	Biofoul Growth	10	227	28	245	217
29	Site	32	125	49	142	93
30	Thrml Cycle	110	110	142	142	0
31	Biol'r Housn'g	142	144	158	154	2
32	Condns'r Housn'g	142	142	154	154	0
33	Support System	142	148	148	154	6
34	Biology @ Site	49	201	85	237	152
35	Machinery	154	158	158	162	4
36	Ext. Pipe	154	154	162	162	0
37	Weight	162	162	176	176	0
38	Power/Cost	176	283	199	306	107
39	Bouyancy	176	176	184	184	0
40	Shape	184	184	201	201	0
41	Maricultures	7	251	21	265	244
42	Waves	201	204	212	215	3
43	Winds	201	201	215	215	0
44	Current Force	201	204	212	215	3
45	Misc. Forces	201	201	215	215	0
46	Stability	215	215	247	247	0

TABLE 3. cont.

#	ACTIVITY	EARLY START	LATE START	EARLY FINISH	LATE FINISH	FLOAT
47	Entrnmt Fouling	201	237	211	247	36
48	Entrnmt Biota	201	237	211	247	36
49	Biofoul-Struc.	201	248	206	253	47
50	Biofoul-Operat'n	201	245	209	253	44
51	Ptl Mooring	247	260	262	275	13
52	Dyn. Position'g	247	269	262	284	22
53	Screens	247	247	259	259	0
54	Biofoul. Prvn't	209	253	227	271	44
55	Mult. Pt. Mooring	262	275	271	284	13
56	Comb. Pos. Syst.	271	284	281	294	13
57	Conv. Anchoring	281	294	293	306	13
58	Embed. Anchoring	281	294	293	306	13
59	E.I. Entrainmnt	259	287	265	293	28
60	E.I. Impingmnt	259	287	265	293	28
61	Sea Circulation	259	259	265	265	0
62	E.I. Circulation	265	281	277	293	16
63	Upwelling Eff. Mari.	265	265	277	277	0
64	Impinge Eff. Mari.	259	274	262	277	15
65	Biofoul Eff. Mari.	227	271	233	277	44
66	E.I. Biofoul Prvn't	227	287	233	293	60
67	Mari. Structure	277	277	289	289	0
68	E.I. Mari	289	289	293	293	0
69	E.I. Statement	293	293	306	306	0
70	Benefit/Cost Mari.	289	295	300	306	6
71	Write-Up	306	306	387	387	0

shortened from 387 hours to 117 hours. It can also be seen that three of the individuals in the environmental group are not needed until mid way through the project. A further reduction in the project duration could be realized if the constraint allowing individuals to work only on activities within their group was removed.

BEGINNING AND ENDING NODES

Often confusion is generated in the use of activity diagrams because of inadequate information about when an activity begins and when it is complete. To help alleviate this problem and further clarify activity descriptions the beginning and ending points for each activity are given in the following table. Immediate prerequisites to each activity are also given.

CONCLUSIONS

An activity diagram is of considerable aide as a management tool. It helps to clarify interdependencies of activities and time constraints which are crucial with a project of limited duration. The critical activities can be identified and the most judicious use of manpower applied in order to avoid time delays or unnecessarily overburdening any one individuals responsibilities. By referring to the bar chart at the completion of activities it is possible to see if the proposed schedule is being met. In the event that it is not, future slack periods where time can be made up are easily identified. If slack periods are not available then other management alternatives available such as eliminating portions of tasks, shifting the work load or working overtime can be considered.

The advantage of activity diagrams and bar charts is difficult to measure, but certainly clarity in activity relationships and time durations contributes to the smooth flow and ultimate success of a project.

The complete PERT Network is shown in Figure 1; Resource Allocation in Figure 2; and Time Allocation in Figure 3.

REFERENCE

Weist, J. and Levy F. A Management Guide to PERT/CPM. Englewood Cliffs, N.J. Prentice-Hall, Inc. 1969.

TABLE 4. BEGINNING & END NODE DESCRIPTIONS

Activity Description	Beginning & End Nodes	Prerequisites
1. Cold Water Pumping Methods:	Begin: Literature Study at Library after task commitments End: Recommendations for pumping	--
2. Temperature Range Theory Analysis:	Begin: Text Study of Workable Temperature ranges for power production after task commitments End: A workable theory for low temperature range power production	--
3. Gross Power Output Calculations:	Begin: Review of power production of preceding power units after task commitments End: Recommendation for magnitude of power production	--
4. Temperature Gradient of the Sea Study:	Begin: Collect published material on subject after task commitment End: When regions of adequate temperature range resolved	--
5. Bathemetry Investigation:	Begin: Survey Bathemetry Maps End: When regions of adequate depth resolved	--
6. Ocean Upwelling Study:	Begin: Search for published material on topic End: Documentation of effects	--
7. Biofouling as a Func. of Ocean Envir. Research:	Begin: Locate material on the subject End: Documentation -- possibly chart of depth vs. creatures	--
8. Optimum Working Fluid Investigation:	Begin: List all possible fluids End: Recommendation for specific fluid	--

TABLE 4. cont.

9.	Heat Exchanger Material Analysis:	Begin: From working fluid & power range (pressure drop) list compatible materials End: Recommendations for compatible materials	2
10.	Working Fluid Flow Circulation Design:	Begin: When working fluid & power output recommended. End: When flow pattern design requirement specified.	3,8
11.	Heat Exchanger Surface Configuration Design:	Begin: When flow pattern & material recommendations complete End: When optimal configuration for heat transfer documented	3,8
12.	Population & Industrial Centers Proximity Study:	Begin: Search for published material on regions near sea End: When regions located documented .	9,10
13.	Size Boiler Units:	Begin: Review recommendations for fluid & surface configuration End: When optimal unit size calculated & documented	4,5
14.	Size Condenser Units:	Begin: Review recommendations for fluid & surface configuration End: When optimal unit size calculated & documented	11
15.	Power Transmission & Distrib. System Design:	Begin: Review pop$^{\underline{n}}$ & power needs near coast and distance to depth End: Recommendation for type of transmission system	11
16.	Calculate Warm Water Flow Rates:	Begin: Review bioler & condenser size End: List flow rates for unit designs	12
17.	Design Piping System Warm Water:	Begin: Review bioler & condenser size End: When optimum theoretical piping design found	1,13,14

TABLE 4. cont.

18. Calculate Cold Water Flow Rates:	Begin: Review boiler & condenser sizes End: List flow rates for unit design	1,13,14
19. Design Piping System Cold Water:	Begin: Review boiler & condenser sizes End: When optimum theoretical piping design found	1,13,14
20. Calculate Working Fluid Flow Rates:	Begin: Review boiler & condenser requirements End: List flow rates through each unit	1,13,14
21. Design Working Fluid Piping System:	Begin: Review Boiler & condenser unit designs End: When piping design with material compatibility found	1,13,14
22. Design Pumping Units Warm Water:	Begin: When flow rates & piping system for warm water completed End: When power required calculated	1,13,14
23. Design Pumping Units Cold Water:	Begin: When flow rates & piping system for cold water completed End: When power required calculated	16,17
24. Design Pumping Units Working Fluid:	Begin: When flow rates & piping system for working fluid completed End: When power required calculated	18,19
25. Bottom Sediment Survey:	Begin: Search for bottom survey publications within power transmission regions End: When material on sediments in regions documented	20,21
26. Current Profile Analysis:	Begin: Search for current profile material within power transmission regions End: When material documented	15

TABLE 4. cont.

27.	Shipping Channels Investigation:	Begin: Search for shipping lane maps in power transmission regions End: When material documented	15
28.	Biofouling Creatures Growth Rate Survey:	Begin: Upon identification of biofouling creatures End: When growth rates for pertinent creatures established	7
29.	Site Selection:	Begin: When material on sediments, temperature, depth, shipping & currents compiled End: When adequate size documented	25,26,27
30.	Optimization of Thermal Cycle:	Begin: Review power & cycle requirements for units End: When requirements for optimum cycle at specific size met	22,23,24
31.	Size Boiler Housing:	Begin: When size of unit for optimum thermal cycle found End: When housing size & material documented	30,29
32.	Size Condenser Housing:	Begin: When size of unit for optimum thermal cycle found End: When housing size & materials documented	30,29
33.	Size Support System:	Begin: When size of thermal units found End: When recommendation for operation & required support system documented	30,29
34.	Biology of SSPP Sight Investigation:	Begin: When sight selected End: When all relevant obtainable material documented	29
35.	Size Extraneous Machinery Required:	Begin: Housing & support system requirements completed End: When major machinery placement decided upon	31,32,33
36.	Design Extraneous Piping Required:	Begin: Housing & support system requirements completed End: When piping sizes & placement specified	31,32,33

TABLE 4. cont.

37. Calculate Weight Distribution:	Begin: Unit sizes & extraneous requirements specified End: List weights	35,36
38. Power/Cost Trade-Off Analysis:	Begin: Review weights & material specifications End: Recommendation on economy of plant	37
39. Design Bouyancy Compensators:	Begin: Calculate distribution of weights End: Documented recommendations on method, size & placement	37
40. Design Configuration of Plant:	Begin: Review unit sizes & bouyancy compensator sizes of compensator End: Configuration recommendation	39
41. A Study of Maricultures:	Begin: Literature review of material on the subject End: Document possible maricultures at plant site	6
42. Calculate Wave Forces:	Begin: Look at wave theory in relation to design configuration & placement End: Documentation of forces	40
43. Calculate Wind Forces:	Begin: Collect material on wind in region of site End: Document forces due to plant configuration	40
44. Calculate Current Forces:	Begin: Review current data from site selection End: Document forces on plant due to configuration & placement in current	40
45. Calculate Other Source Forces:	Begin: List miscellaneous forces that could be acted End: Document calculated forces acting & conditions	40
46. Orientation Stability Analysis:	Begin: Compilation of all forces acting End: Stable orientation recommendation	42,43,44,45
47. Effect on Entrainment on Plant Operation:	Begin: Review Biology for types biota entrained End: List effects, cause, remedies	40,34

TABLE 4. cont.

48.	Effect of entrainment on Biota:	Begin: Review biology for types biota entrained End: List adverse effects & cause	40,34
49.	Structural Effects of Biofouling:	Begin: Review design configuration & biology of sight End: List adverse effects and further design considerations	40,34,28
50.	Biofouling Influence on Plant Performance:	Begin: Review plant flow & materials imposed End: List possible performance hindrance & cause	40,34,28
51.	Single Pt. Mooring Analysis:	Begin: Review stability requirements, forces & weight distribution End: Investigation into multiple pt. mooring	46
52.	Dynamic Positioning Analysis:	Begin: Review stability requirements End: Recommendation with power requirement specifications	46
53.	Design Screening Device:	Begin: Analysis of entrainment results & configuration effects End: Documented design for required screening	46,47,48
54.	Biofouling Prevention Methods:	Begin: When types & rates of growth of biofouling creatures on structure complete End: Documented recommendations from latest literature on innovative design	49,50
55.	Multiple Pt. Mooring Analysis:	Begin: When single pt. mooring analysis complete End: Searched into possible combined dynamic & pt. mooring systems	51
56.	Combined Positioning Systems:	Begin: When dynamic and multiple pt. investigation recommendations documented End: When review of all positioning system recommendations made	52,55

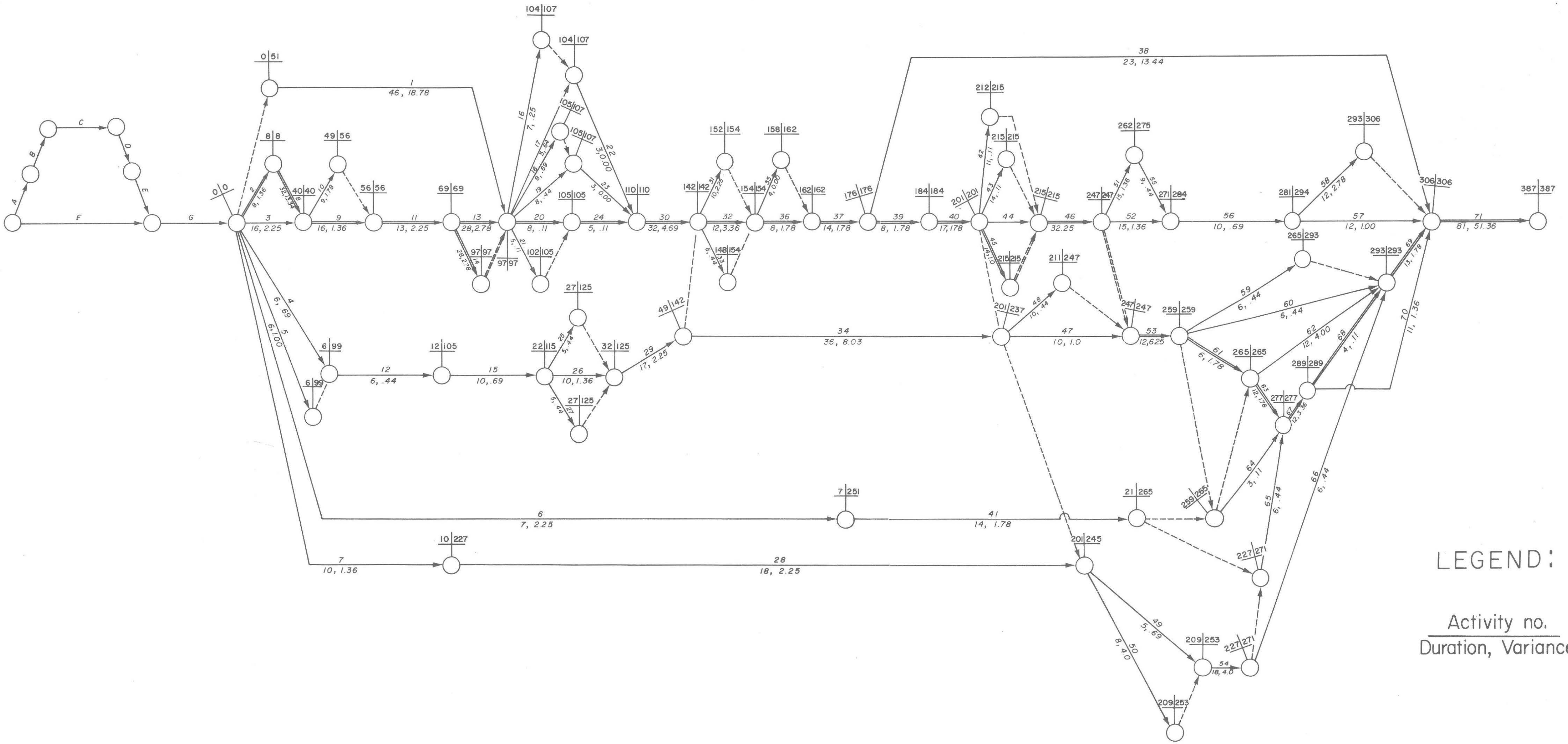
TABLE 4. cont.

57. Conventional Anchoring Design:	Begin: When design recommendations for positioning made End: When field thoroughly investigated & recommendations documented	56
58. Embedient Anchoring Design:	Begin: When design recommendations for positioning made End: When field thoroughly investigated & recommendations documented	56
59. Environmental Impact Entrainment:	Begin: Review effect entrainment on biota End: Documentation of impact concerns	52
60. Environmental Impact Impingement:	Begin: Review effect of impingement End: Documentation of impact concerns	52
61. Sea Circulation About SSPP:	Begin: Review screening device flow patterns End: Diagramming of flow pattern	52
62. Environmental Impact Circulation & Upwelling:	Begin: Review of upwelling & flow pattern End: Document impact considerations	40,61
63. Circulation & Upwelling Effect on Mariculture:	Begin: Review upwelling & flow pattern End: Document possible effects	40,61
64. Impingement effect on Mariculture:	Begin: When impingement effects on biota documented End: Adverse effects documented	40,53
65. Biofouling Prevention Influence on Mariculture:	Begin: When biofouling prevention method decided End: Impact consideration documented	40,54
66. Environmental Impact Biofouling Prevention Method:	Begin: When biofouling prevention method decided End: Impact consideratior documented	54

TABLE 4. cont.

67.	Mariculture Containment Structures:	Begin: When circulation, biofouling agents & screening effect known End: When possible structures conceptual design completed	63,64,65
68.	Environmental Impact Maricultures:	Begin: When type of culture & structures known End: When documentation of possible effects complete	67
69.	Environmental Impact Statement:	Begin: When all structural, operational, biology aspects documented End: Necessary considerations documented	59,60,62,68,66
70.	Benefit/Cost Analysis of Maricultures:	Begin: When mariculture type & structure investigation complete End: When analysis documented	67
71.	Write-Up:	Begin: When final recommendations complete End: When last page typed	38,57,69,70

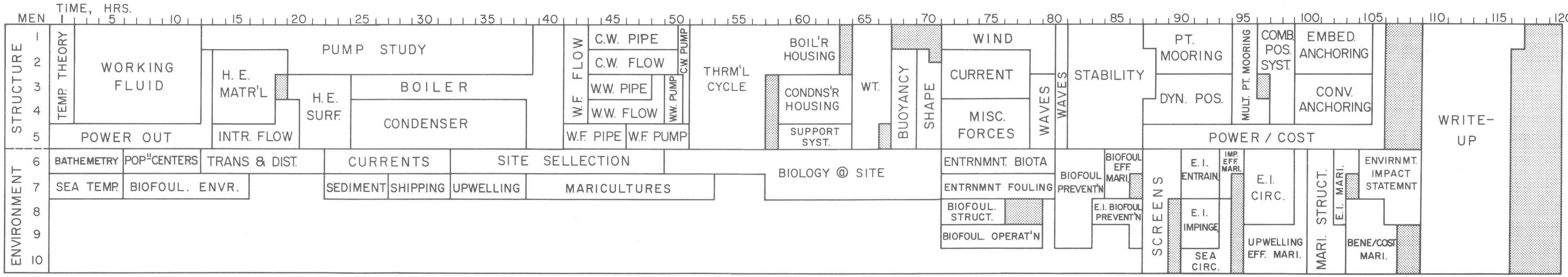
SOLAR SEA POWER PLANT PERT NETWORK



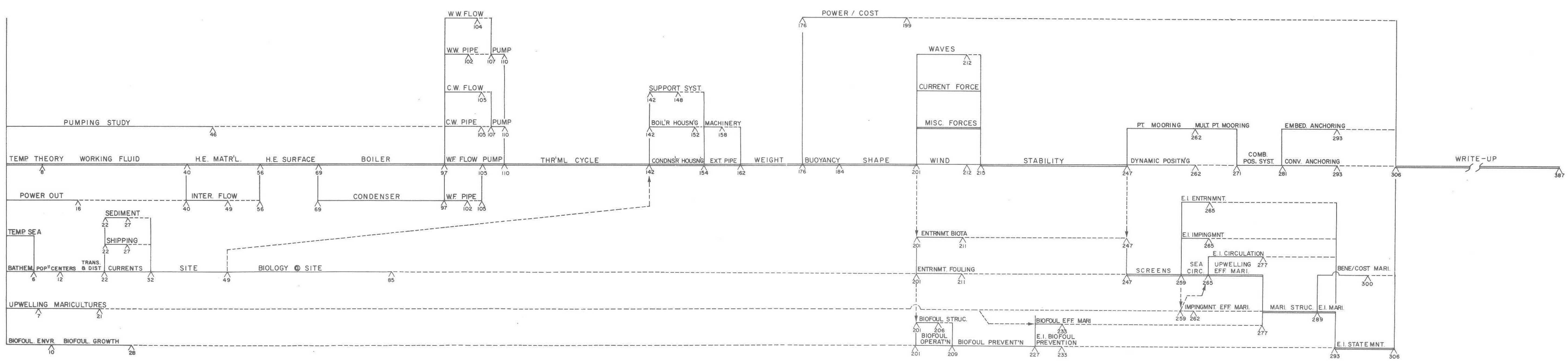
LEGEND:

Activity no.
Duration, Variance

SOLAR SEA POWER PLANT RESOURCE ALLOCATION



SOLAR SEA POWER PLANT TIME CHART



SEA SOLAR POWER PLANT - SITE SELECTION

by

Bob Arneson

ABSTRACT

The processes of selecting a site location for the Sea Solar Power Plant are discussed. One major area not included is political bearing on the siting. The location chosen is approximately 16 miles east of Miami, Florida in the Atlantic Ocean. It is positioned in the Florida Current which follows the continental shelf along the coast of Florida. The major reason for choosing a site in the current was that it offered the necessary temperature profile for operation of the sea solar power plant. A benefit received from the current is the dispersal of the warm condenser discharge water before it can be recycled into the boiler intake. The close proximity of major population centers provides potential customers for the power generated by the plant.

SEA SOLAR POWER PLANT - SITE SELECTION

by

Bob Arneson

INTRODUCTION

The process of site selection can be approached from a number of different viewpoints. Some of the factors which must be considered are the water depth, bottom conditions, current, wave data, temperature gradient, and proximity to population and shipping lanes. Of these the one which must be given top priority is the temperature gradient, since this is the theoretical basis for the solar sea power plant. Other groups have found suitable sites which exhibit this necessary temperature gradient. For the United States these include areas in the Gulf Stream and along the southern California coast (Solar Sea Power Plant Conference, 1973). This study has chosen a site in the Florida Current which flows north along the continental shelf between Florida and Cuba and the Bahamas. At approximately 33° N, which is just south of Cape Hatteras, North Carolina, the current leaves the shelf and goes northeast into deep water. It joins the Antilles Current to become the Gulf Stream (Stommel, 1958).

PROPOSED SITE

The exact location of the proposed plant site is 25° 53' N, 79° 50' W. This is 16 miles off the coast of Miami, Florida. A site temperature profile taken by the University of Miami on January 18, 1963 at 25° 36' N, 79° 53' W is shown in Table 1. This is close to the selected site, and is also within the Florida Current, so it may be considered representative of on site conditions.

It can be seen from Table 1 that temperature remains essentially constant down to 83 meters, where it reaches a fairly steep thermocline. From 377 meters to the bottom it is again essentially constant. The sounding at this position was 444 meters (University of Miami, 1964). The surface layer of 23°C water will be used for the warm water intake and the 8°C water taken from depth is the cold water condenser supply. Although there are seasonal variations of this temperature profile, they do not appear to be of sufficient magnitude to halt plant operations (Stommel, 1958).

TABLE 1. TEMPERATURE PROFILE IN THE FLORIDA STRAITS
(University of Miami, 1964, pp 20-22)

Depth, in meters	Temperature, in degrees centigrade
1	23.88
9	23.88
18	23.86
32	23.46
83	22.14
133	18.94
182	15.00
280	9.66
377	8.33
424	8.08
433	8.05

Since we are dependent on this temperature difference which exists in the current, any meandering by the current could seriously affect plant operations. In 1937, Church demonstrated that the position of the Gulf Stream is not always the same. In 1950, from results of the Multiple Ship Survey, meanders in the Gulf Stream were again observed (Stommel, 1958). More recent investigations have also been conducted on these meanders (Robinson, 1974). None of these studies showed data in the Gulf Stream south of Cape Hatteras. However, it appears that the Florida Current in the region of this site does not meander or shift its position radically (Stommel 1958). Therefore, it appears that the necessary temperature profile will remain on site throughout the year.

On Figure 1, it is seen that the depth of water at the plant site is 300 fathoms, or 1800 feet. This depth was chosen for several reasons. The water must be deep enough for the cold water pipe to penetrate to the bottom cold layers (8°C or less). If a position were chosen where only minimal clearance for the pipe was obtained, there would be a tendency for wave action, however slight, to cause grounding of the pipe. There would also be a suction of the bottom materials into the cold water intake. The selected location provides several hundred feet of separation from the bottom, thus eliminating these potential problems. Conversely, it would not be desirable to chose a site in very deep water. Added depth would increase the anchoring problems and costs.

By being only 16 miles off the coast of Miami, the plant will be easy to reach by both helicopters and small surface ships. Even though it is not planned to exchange crews more than once every several weeks, being relatively close to shore makes this operation a short and simple procedure. Further, transportation of produced power to shore by cable over that distance will be relatively inexpensive, easy, and will exhibit minimum line loss. The plant site is also in the vicinity of large population centers to consume the electrical output.

Even though the plant is going to be submerged most of the time, it would still not be desirable to place it in a shipping lane. Some deep draft ships would pose potential hazzards, especially in adverse weather conditions. Further, the plant would pose a hazzard to all shipping when surfaced. The chosen site location is several miles west of the shipping lanes. (Atlas of Pilot Charts, 1955). However, it would still be prudent to have a lighted buoy marking the position of the plant.

The current is the resource which causes the favorable temperature profile. It is also what causes the problems of positioning the plant. The magnitude of the current can be seen on the profile in Figure 2 from University of Miami May data, 1964.

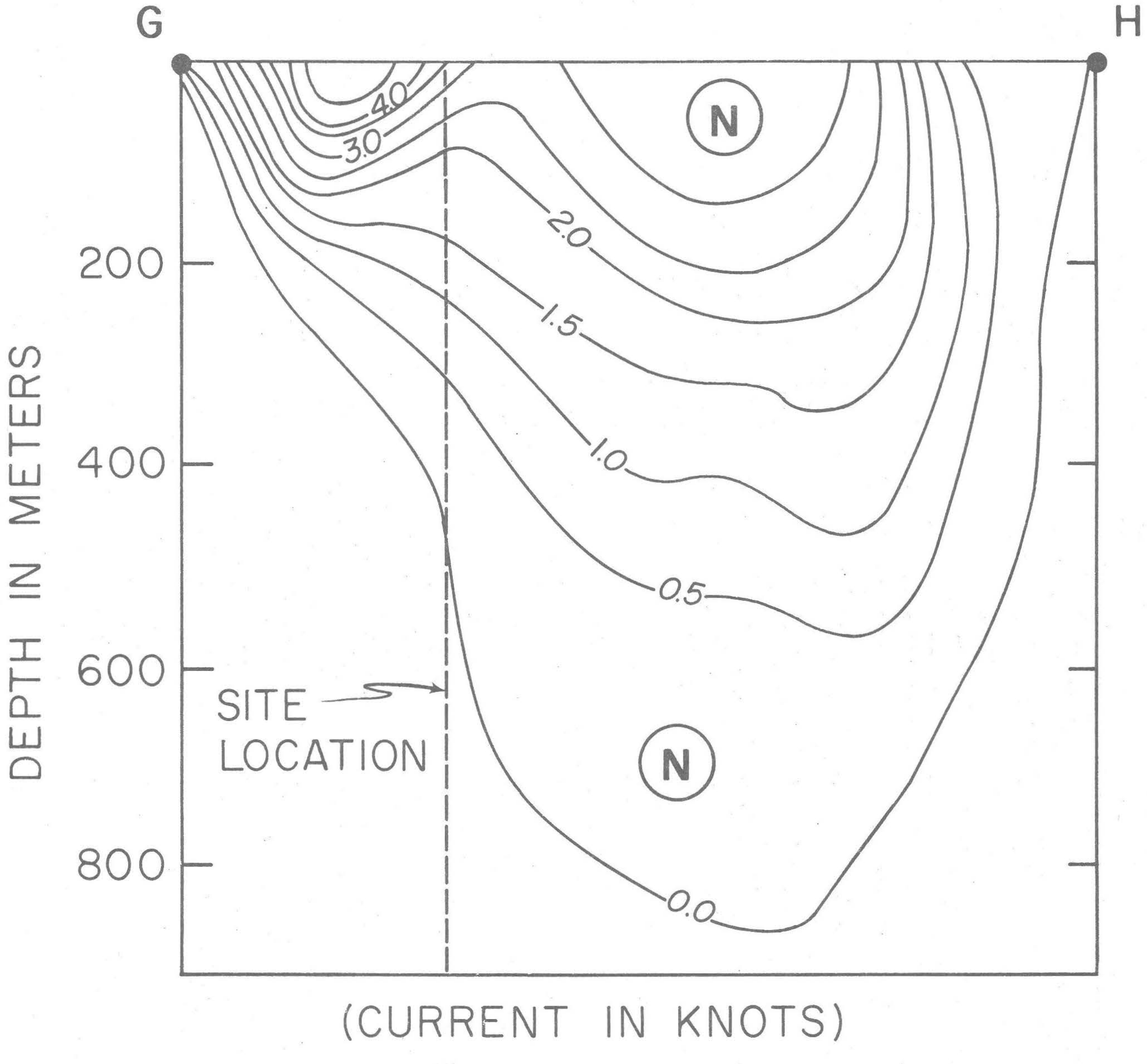


FIGURE 1. May Plant Current Profile (U.S. Naval Oceanographic Office, 1965)

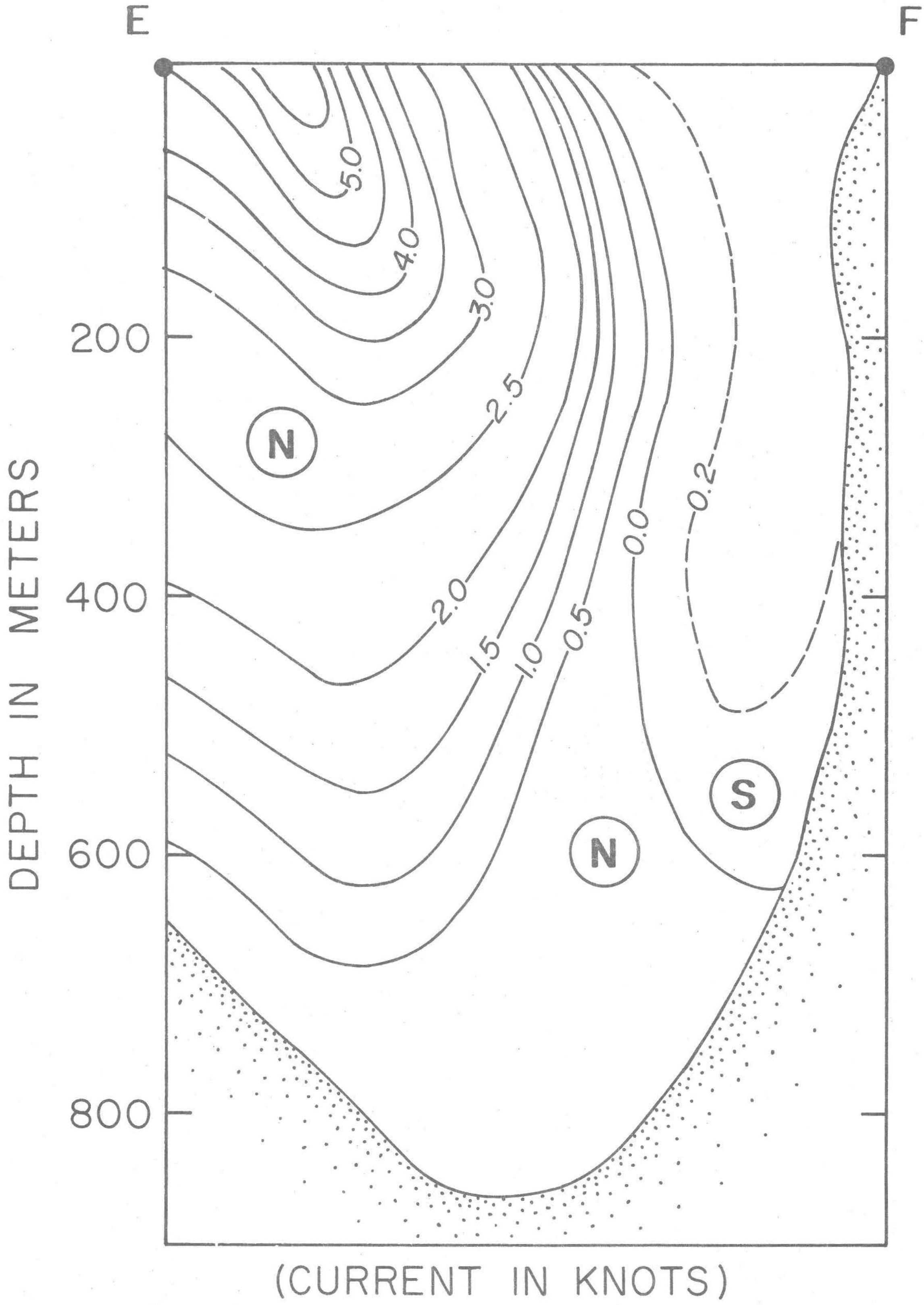


FIGURE 2. August Plant Current Profile (U.S. Naval Oceanographic Office, 1965)

The coordinates of points G and H are 25° 55' N, 80° 04' W and 25° 47' N, 79° 16' W, respectively. The dotted line shows the location of the plant. At this point the current decreases from 3.5 knots at the surface to zero at 500 meters. The N indicates that the general direction of flow is to the north. The flow in the current is always northerly; however, there is evidence to indicate slight southerly flow on the edges of the current. This phenomenon, occuring in August, is shown on Figure 3. The coordinates of points E and F are 25° 39' N, 79° 56' W and 25° 27' N, 79° 16'W, respectively. Although the speed of the current is continuously changing, a seasonal comparison of surface current shown on Table 2 indicates the mean speed and percentage of the time at a certain speed to be relatively constant.

Another asset provided by the current is a mixing and dispersal of the warm water as it leaves the condenser. This should preclude recycling of condenser discharge to warm water intake. Due to the great depth to the cold water and the fact that the warm water is less dense. There should be no problem of the warm water sinking and being recirculated into the cold water intake.

TABLE 2. SURFACE CURRENTS IN THE FLORIDA STRAITS.
(U.S. Naval Oceanographic Office, 1965)

Season	Directional mean speed, in knots	Speed, in knots	Time at Speed (%)
Winter (January, February, March)	2.5 N 2.0 NE	>4	5
		3.0-3.9	24
		2.0-2.9	31
		1.0-1.9	23
		0.1-0.9	17
Summer (July, August, September)	2.7 N 2.0 NE	>4	8
		3.0-3.9	29
		2.0-2.9	29
		1.0-1.9	18
		0.1-0.9	16

The current is a major drag force on the plant. The other factors to consider are the wave action and wind forces. If the plant were a floating or semi-submerged design it would be prudent to design for the most severe conditions. A margin of safety would also have to be included. Since the plant will be submerged with the exception of crew changes and some maintenance, it is not necessary to take the most sever conditions as design factors. It is desired to operate the plant at a depth below the affect of surface forces and wave action. If the surface conditions are worse than those designed for, the plant can remain submerged until conditions permit surfacing for required crew changes.

The wave and wind forces for the area of the plant vary seasonally as the current does. The roughest month is September with the waves being greater than 12 feet three percent of the time with a maximum height of 28 feet. The wind is Beaufort Force 7-12, three percent of the time, or greater than 28 knots (U.S. Naval Oceanographic Office, 1963).

Political and social considerations have not been mentioned. They are many and varied and it is felt that they are beyond the scope of this report. For those wishing to investigate this area, background should be provided by the oil companies from their vast experience with the oil platforms.

CONCLUSIONS

1. The selected site (25° 53'N, 79° 50'W) will provide a stable thermal gradient on a year round basis sufficient for plant operation.

2. Other site advantages, including: local water depth, proximity to major population centers and to shore, local current to remove exhaust flows and site seperation from shipping lanes, far outweigh the one main site disadvantage of increased drag force from relatively high local currents.

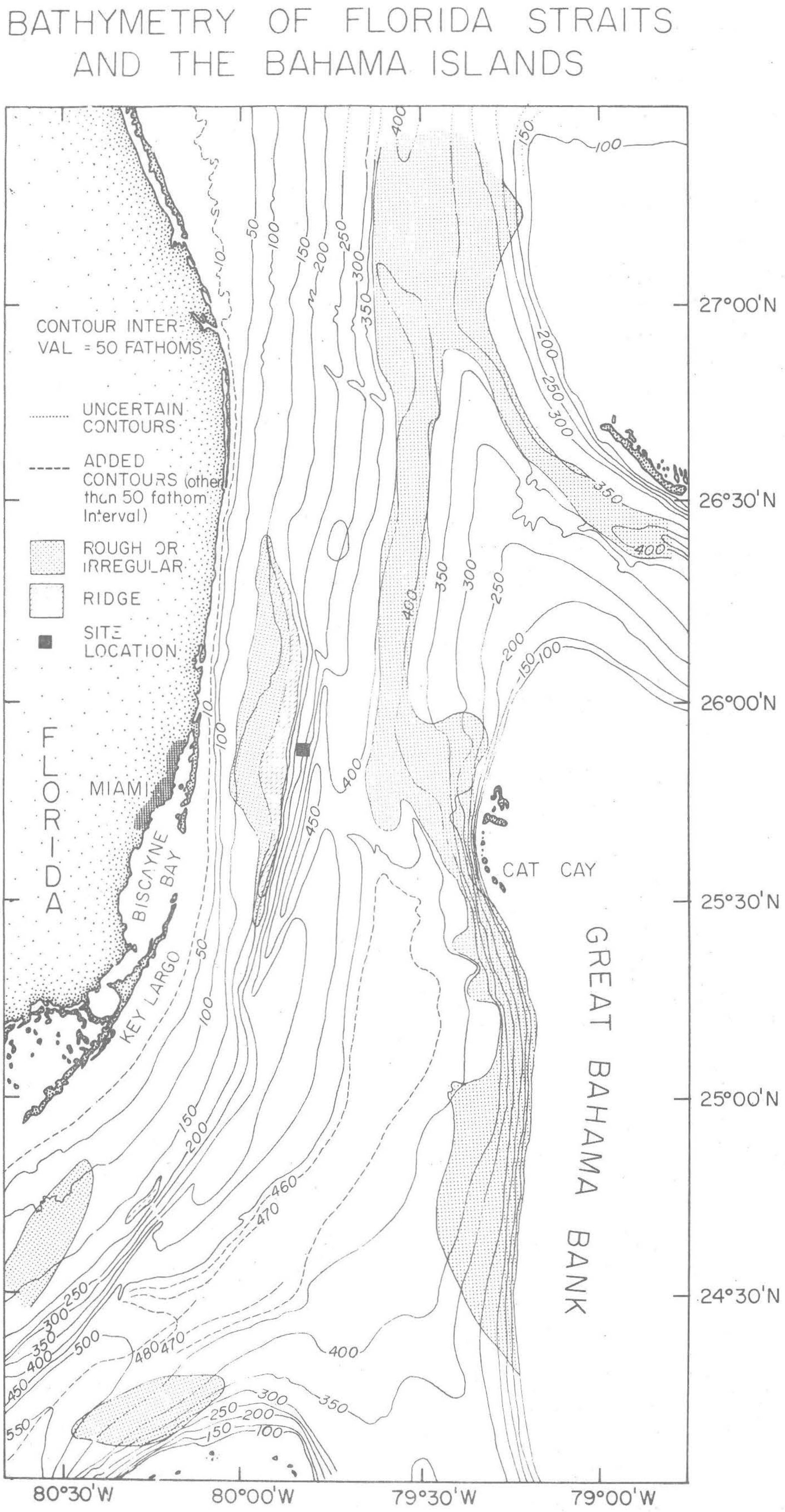


FIGURE 3. Bathymetry of Florida Straits and the Bahama Islands (U.S. Naval Oceanographic Office, 1965)

REFERENCES

Atlas of Pilot Charts, Central American Waters and South Atlantic Ocean, H.O. Pub. No. 576, Second Edition. 1955. U.S. Navy Hydrographic Office, Washington, D.C. March, 1955.

Hurley, R.J. "Topography of the Straits of Florida." Investigations of American Tropical and Subtropical Seas. Institute of Marine Science, University of Miami, Miami, Florida. July 1963.

Oceanographic Atlas of the North Atlantic Ocean, Section I Sea and Swell. Pub. No. 700, U.S. Naval Oceanographic Office, Washington, D.C. 1963.

Oceanographic Atlas of the North Atlantic Ocean, Section II Tides and Currents. Pub. No. 700, U.S. Naval Oceanographic Office, Washington, D.C. 1965.

Robinson, A.R., Luyten, J.R., and Fulister, F.G. "Transient Gulf Stream Meandering. Part I: An Observational Experiment." Journal of Physical Oceanography. 4(2): 237-254, April, 1974.

Stommel, Henry. The Gulf Stream. London: Cambridge University Press. 1958.

Sverdrup, H.V., Johnson, M.W. and Flemming, R.N. The Oceans. Englewood Cliffs, N.J.: Prentice Hall, Inc. 1942.

University of Miami. "A Report of Data Obtained in the Florida Straits and Off the West Coast of Florida, January - June, 1963." 64(1). Institute of Marine Science, University of Miami, January 1964.

"Workshop On: Power Plant Siting, Economic and Political Problems, Environmental Considerations." Proceedings, Solar Sea Power Plant Conference and Workshop. Carnegie - Mellon University, Pittsburg, Pennsylvania, June, 1973.

SEA SOLAR POWER PLANT - HEAT EXCHANGER ANALYSIS

by

Dan Ladd

ABSTRACT

This discussion shows the procedure used in analyzing the operating characteristics of a particular heat exchanger design. The design used was chosen for its ease of construction and maintenance. The operating parameters varied were the condensing and evaporating temperatures, the water velocity through the exchangers, and the exit water temperatures. The dewired design parameters were the heat exchanger volume, surface area, flow area and pumping power requirements for particular operating conditions.

The conclusions reached were:

(1) Surface area and pumping requirements go down for higher heat exchanger-water temperature differences.

(2) Surface area decreases, while pumping power increases for increasing water velocity.

(3) Flow area decreases for increasing water inlet-outlet temperature differences.

Finally, the optimum design consists of balancing the heat exchanger operating characteristics within the physical limitations of a sea solar power plant design.

SEA SOLAR POWER PLANT - HEAT EXCHANGER ANALYSIS

by

Dan Ladd

INTRODUCTION

One of the most important considerations in any attempted design of a sea solar power plant, is the design of the heat exchangers, as these represent the largest and most expensive components of any conceivable plant. This discussion shows the procedure used analyzing the operating characteristics of a particular heat exchanger design, operating at the conditions of a particular site location. The operating parameters varied were the condensing and evaporating temperatures, the water velocity through the exchangers, and the water exit temperatures. The heat exchange design parameters desired were the volume, surface area, flow area, and pumping power requirements for particular operating conditions. Nomenclature used for this report is listed in Table 1.

TABLE 1. NOMENCLATURE

Symbol	Definition
Q	Heat flow rate, $\frac{Btu}{sec}$
Q_{evap}	Heat flow rate at the evaporator
Q_{cond}	Heat flow rate at the condensor
h	Specific enthalpy, $\frac{Btu}{lb}$
$\dot{M}$	Mass flow rate, $\frac{lb}{sec}$
U	Overall heat transfer coefficient, $\frac{Btu}{Hr\ ft^2 °F}$
A_{ht}	Heat transfer area, referred to U, ft^2
LMTD	Logrithmic mean temperature difference, °F
$\dot{V}$	Volume flow rate, $\frac{ft^3}{sec}$
C	Specific heat, $\frac{Btu}{lb\ °F}$
T_{exit}	Exit temperature
T_{in}	Inlet temperature
G	Area volume flow rate, $\frac{lb}{ft^2 sec}$
f	Friction factor
ρ	Density, $\frac{lb}{ft^3}$

AMMONIA CYCLE

Figure 1 is a skematic of the heat engine cycle proposed for the Sea Solar Power Plant (SSPP). Ammonia was chosen as the working fluid as it appears that this is the most satisfactory fluid, from purely efficiency considerations (Olson, et al., 1973). Thermodynamic analysis of the ammonia cycle results in the following equations.

$$Q_{evap} = \dot{M}_{NH_3}\ (h_1 - h_4) \quad \ldots\ldots (1)$$

$$Q_{cond} = \dot{M}_{NH_3}\ (h_3 - h_2) \quad \ldots\ldots (2)$$

$$Work_{net} = \dot{M}_{NH_3}\ [(h_2 - h_1) - (h_4 - h_3)] \quad \ldots (3)$$

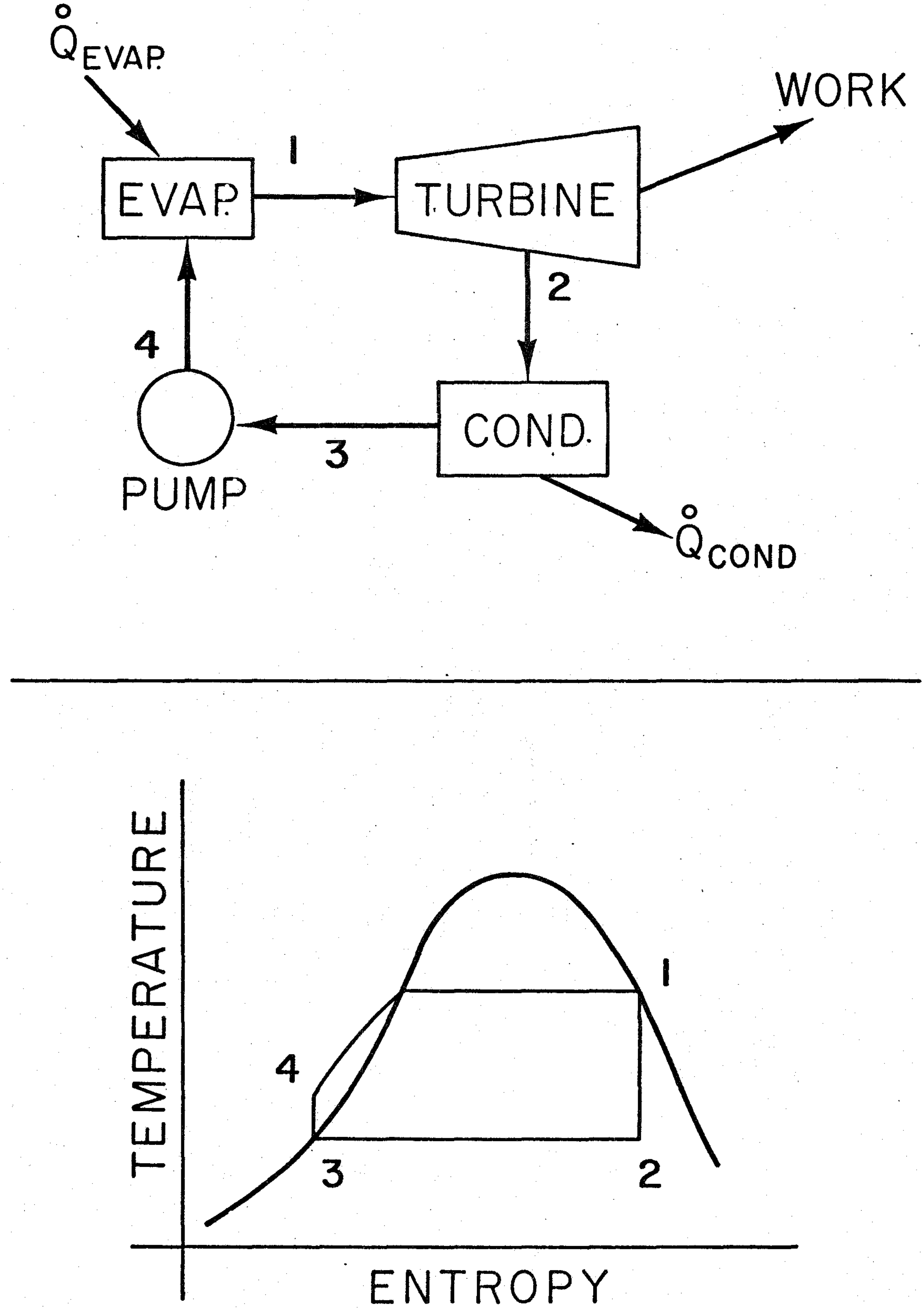


FIGURE 1. Ideal Ammonia Cycle

Figure 2 is a result of calculations using the above equations. The temperature difference refers to difference between the cycle working fluid temperatures and the seawater ambient temperatures in the condensor and evaporator where seawater temperatures are taken to be 47°F and 73°F for cold and warm water intakes respectively. It is assumed that these temperature differences are the same for both the evaporator and condensor. Thermodynamic data was taken from Van Wylen - Sonntag (1965). Turbine and pump were assigned efficiencies of 85%. This figure is representative of existing designs and is adequate for the level of calculations presented.

HEAT EXCHANGER DESIGN

Figure 3 is a cross section of the heat exchanger design used. This design was chosen for its ease of construction and maintenance, and because of its relatively low cost with reasonable heat transfer characteristics (Anderson, Sept. 1973). Aluminum was chosen as the heat exchanger material primarily because of its good heat transfer characteristics. Work done by J.H. Anderson (Sept. 1973) considered many different materials, including plastics, but aluminum seemed to be least costly overall. The corrosion problems associated with aluminum in seawater seem possible to overcome (Godard, et al., 1967, and Anderson, Sept. 1973).

For simplicity, the design of the condensor and evaporator were taken to be identical, as were the properties of seawater at the condensing and evaporating temperatures. Figure 4 shows the variation in overall heat transfer coefficient (U) and the friction factor (f) as a function of seawater velocity. U and f are defined by the following equations.

$$Q = U\ A_{ht}\ (LMTD) \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

$$\text{or, } U = \frac{Q}{A_{ht}\,(LMTD)} \quad \ldots\ldots\ldots\ldots\ldots\ldots (4a)$$

$$\text{and, Pumping Power} = \frac{G^3 A_{ht} f}{2p^2} \quad \ldots\ldots\ldots (5)$$

$$\text{or, } f = \frac{(\text{Pumping Power})\ 2p^2}{G^3 A_{ht}} \quad \ldots\ldots\ldots (5a)$$

U is calculated as the inverse of the sum of the individual heat transfer resistances. The assumptions involved in the calculation of U are as follows. The condensing and evaporating coefficients were taken to be the same and equal to

$$800\ \frac{\text{Btu}}{\text{hr ft}^2\ {}^\circ\text{F}} .$$

Although this is certainly wrong, any analysis that involves changing the particular inside heat transfer coefficients involves the ammonia velocity which depends on the individual heat exchanger geometry, for which this analysis hopes to solve. In any case, the inside heat transfer coefficients have negligible effect on the overall resistance, and the value picked is certainly on the conservative side. The inside and outside fouling resistances were taken to be

$$.005\ \frac{\text{hr ft}^2\ {}^\circ\text{F}}{\text{Btu}} .$$

The water side resistances were taken from Table 5.4.1, lines 26 to 29, J.H. Anderson (Sept., 1973) as were the friction factors. These values evolved from the data in Figure 32 of Kays & London (1965).

Using eq. (4), and knowing the required heat flux from Figure 2, Figures 5 and 6 can be drawn showing the requirements of the product U x A_{ht} for varying condensing (evaporating) temperatures and for seawater exit temperatures. From Figure 5, Figure 7 can be drawn. Although Figure 7 is drawn for a seawater exit temperature of 50°F and velocity of 4 ft/sec, it shows the general trend, that both condensor area and pumping power drop to a minimum at a condensing temperature of about 55°F. This curve stems from two separate forces. Initially, the increasin

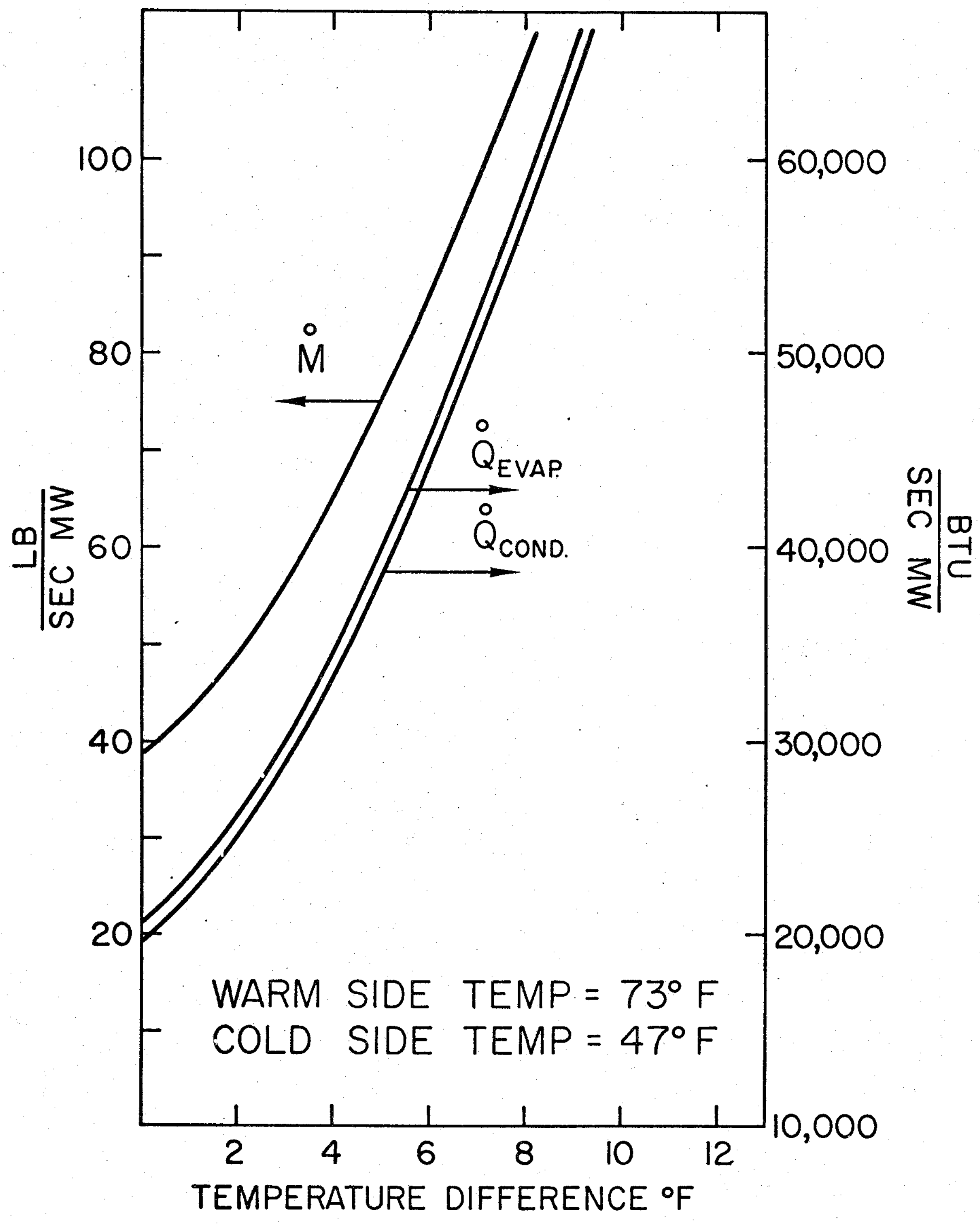


FIGURE 2. NH_3 Flow Rate and Heat Flux vs. Condensor-Evaporator Seawater Temperature Difference

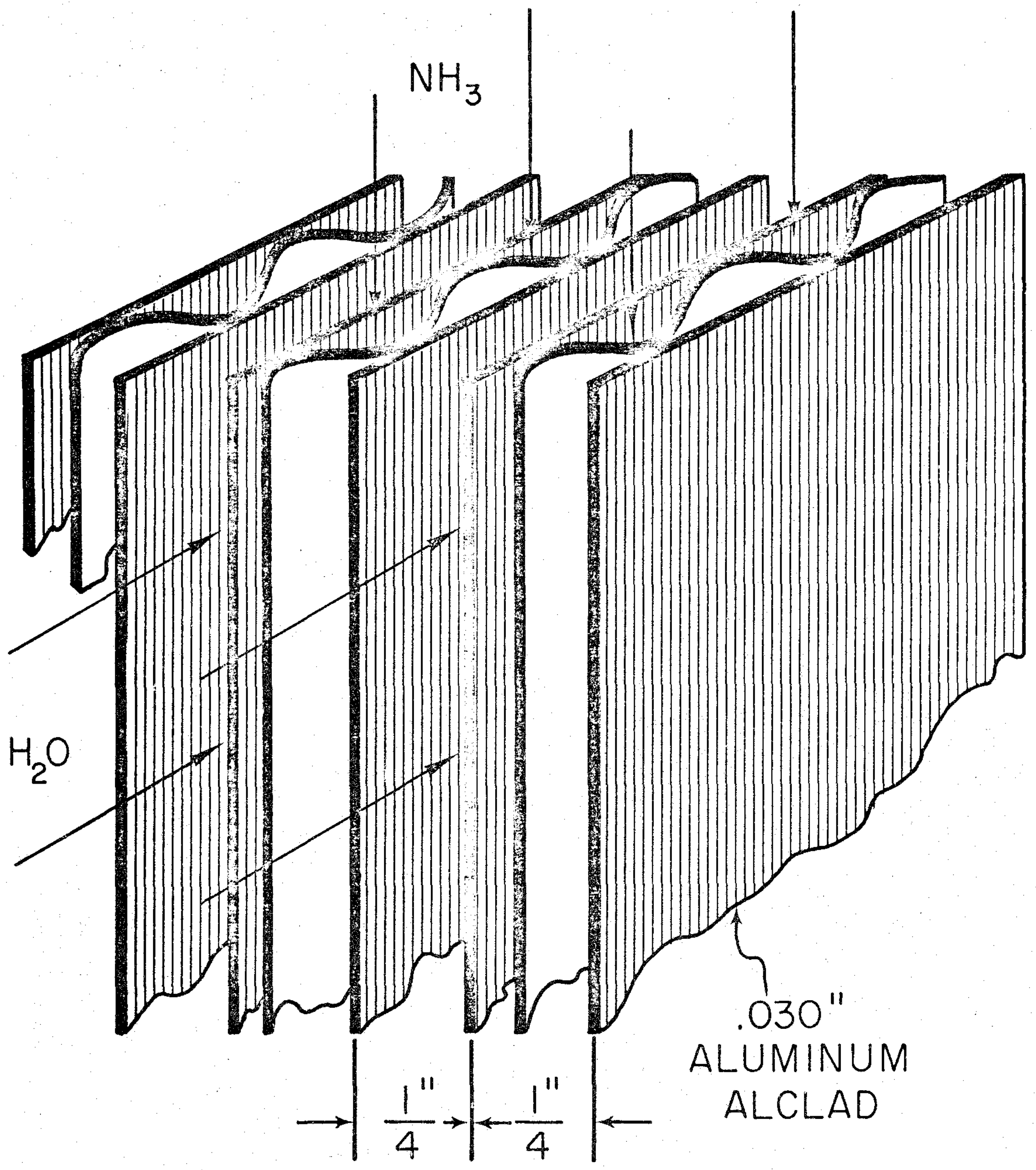


FIGURE 3. Condensor Cross Section Design

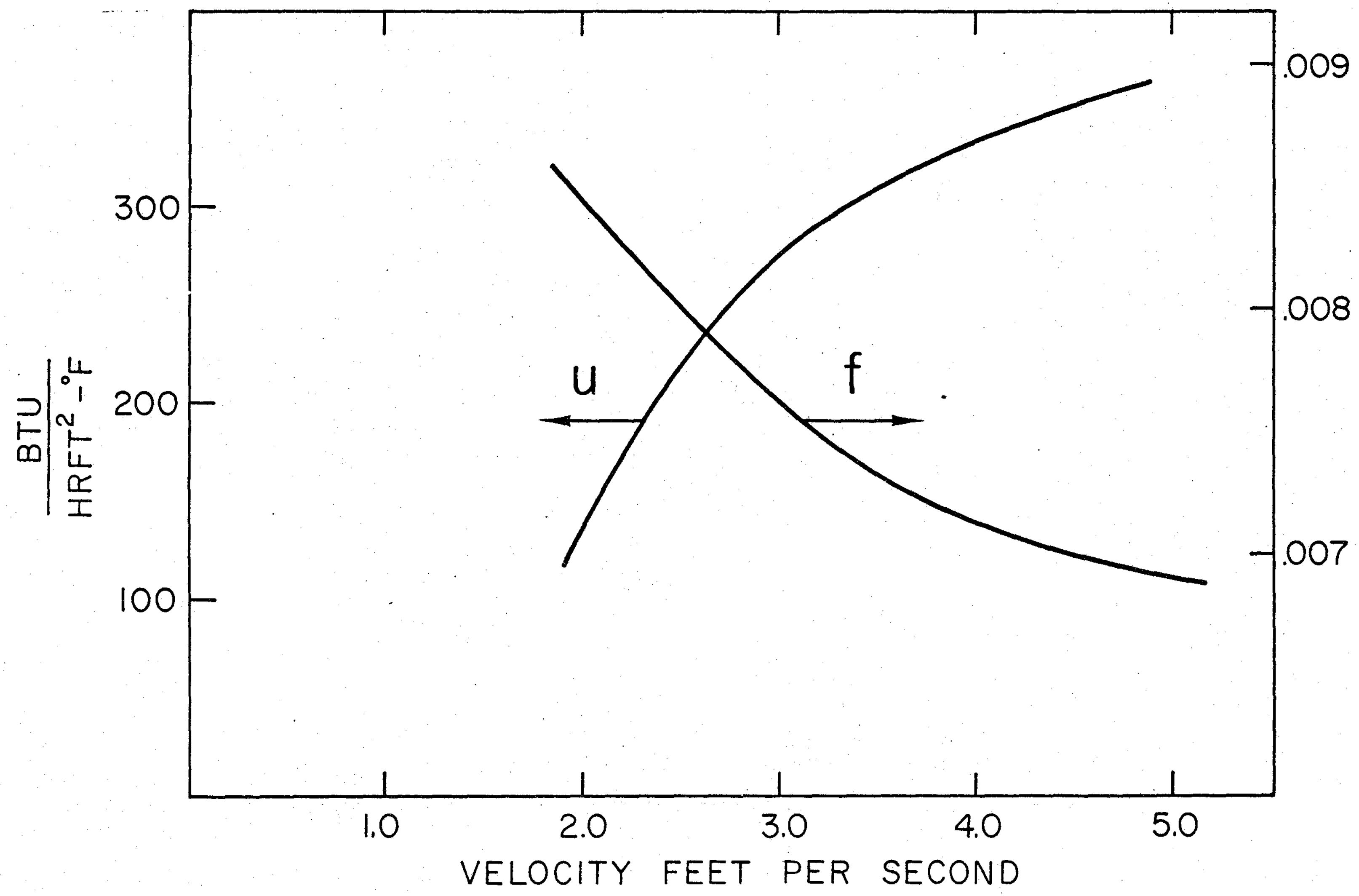


FIGURE 4. Heat Transfer Coefficient and Friction Factor vs. Seawater Velocity

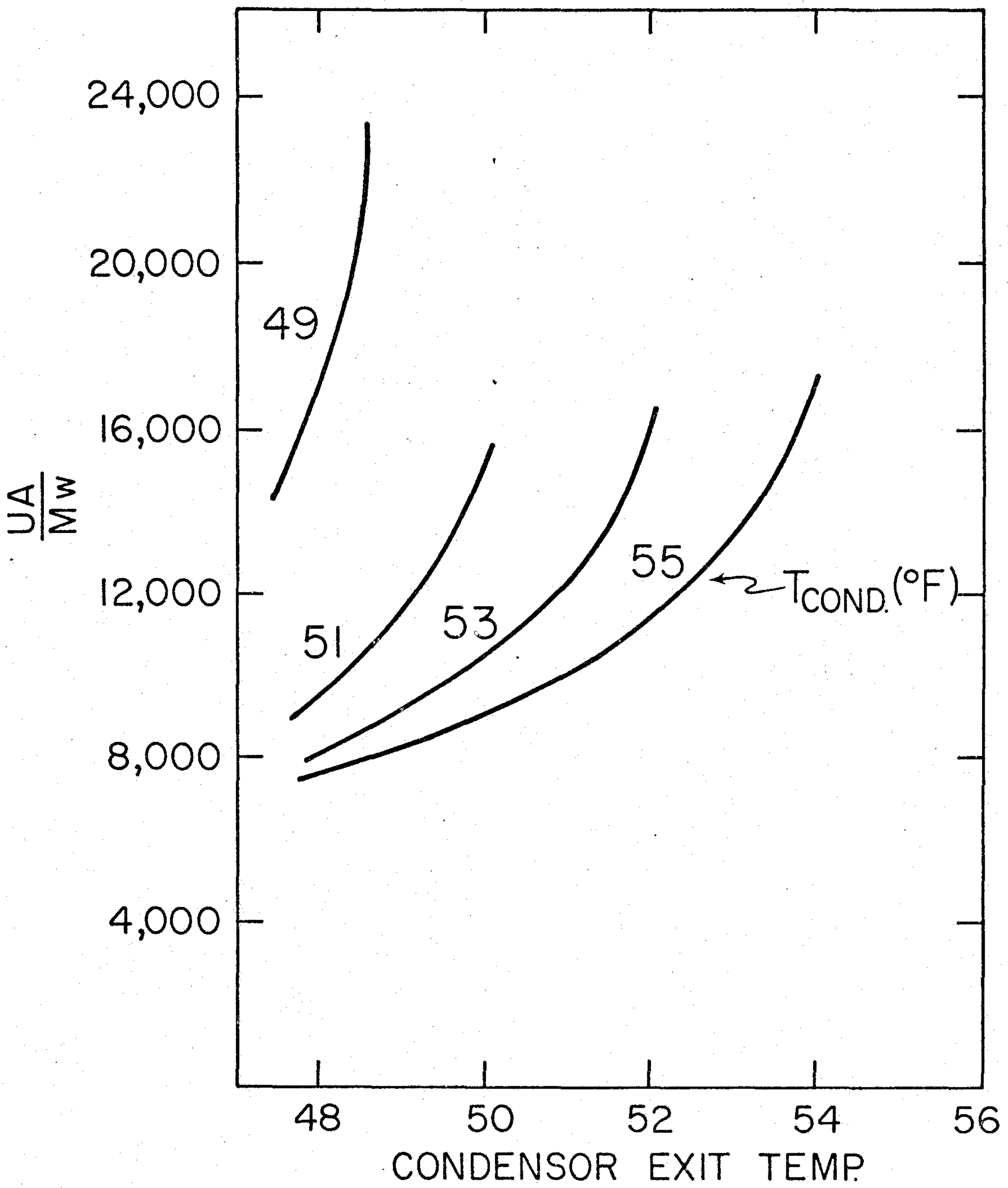


FIGURE 5. UXA_{HT} vs. Condensor Exit Temperature

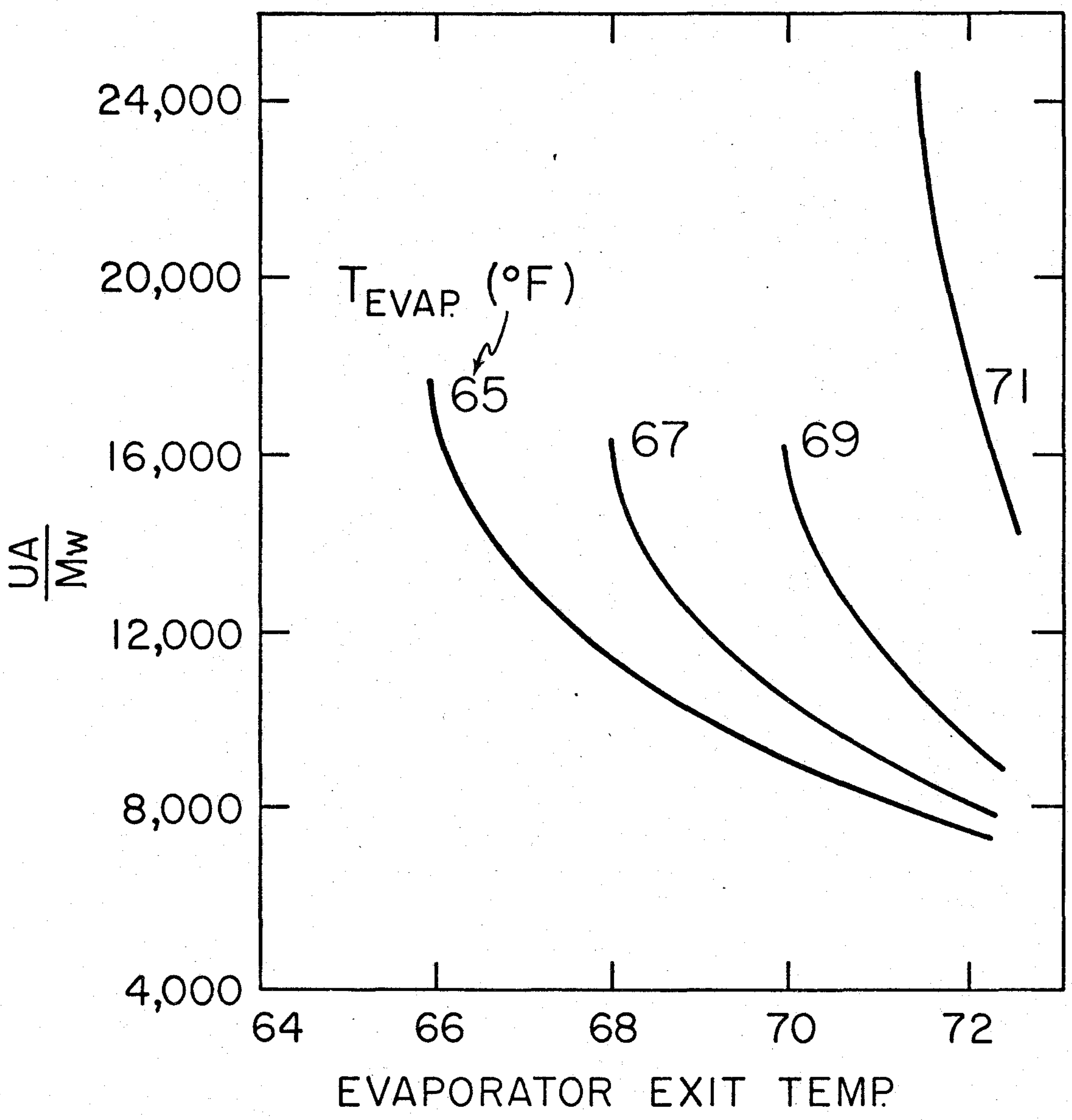


FIGURE 6. UXA_{HT} vs. Evaporator Exit Temperature

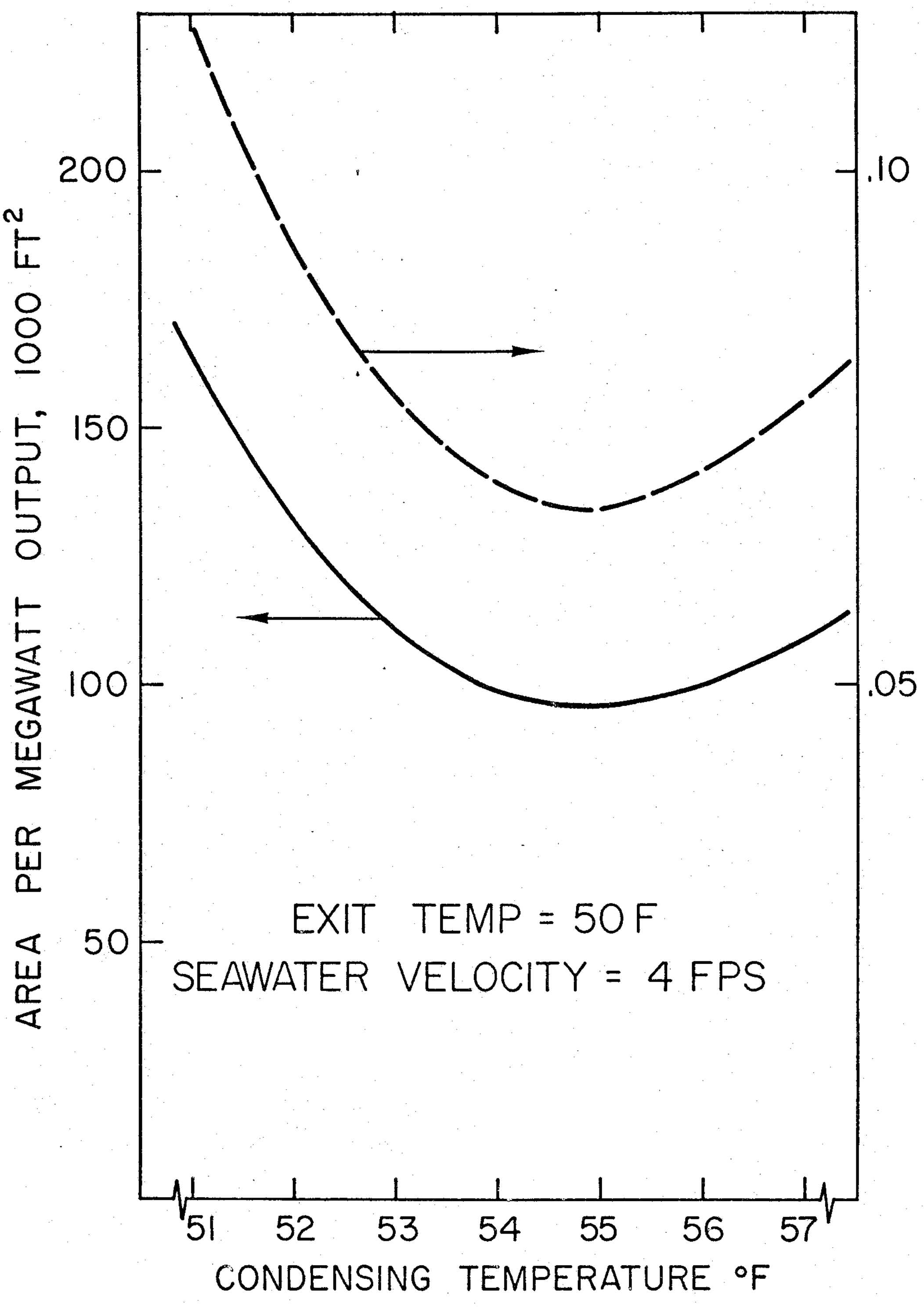


FIGURE 7. Pumping Power Requirements vs. Output Power for Varying Condensing Temperatures

condensing temperature increases the logrithmic mean temperature difference so that the area decreases. However, at above 55°F, the amount of heat rejected needed to produce one megawatt of power increases so drastically (Figure 2), as to more than cancel out the effect of the increasing LMTD. In the simplified analysis used, the 55° condensing temperature corresponds to a 65° evaporating temperature, with the exception that both the area and the pumping power would be about three percent higher for the evaporator.

Figures 8-11, show the effect of increasing water velocity on the area and pumping power requirement of the condensor for various T_{cond}. While the pumping power goes up, for increasing velocity, the area goes down. The pumping power was calculated from equation (5), assuming an 85% pump efficiency. Entering and leaving losses were ignored, as these are somewhat a function of actual plant layout.

Figures 8-11 also show that as the condensor seawater exit temperature decreases, the pumping power and area also decrease. However, in accordance with the equation,

$$Q = V_{H_2O} \rho_{H_2O} C(T_{c_{exit}} - T_{c_{in}}) \ldots\ldots (6)$$

as the inlet-outlet water temperature decreases, the water flow rate must increase to remove the same amount of energy. This is illustrated in Figures 12 and 13. This in fact places a lower limit on the condensor seawater exit temperature, and an upper limit on the evaporator exit temperature, as at some point, the pumping losses, other than those at the heat exchanges, (ie, pipes, water distribution etc.) will become excessive. Figure 14 shows another interesting effect of the lower seawater exit temperature, at the condensor. At the lower exit temperatures, the condensor must have a greater frontal area, to accomodate the greater water flow, and thus the condensor depth must decrease, to have the same volume. This results in a broad, shallow condensor, which due to other plant considerations may not be practical.

Although this discussion deals mostly with the condensor, using the previous assumption, this work can be applied equally well to the evaporator, with the slight correction that the area and pumping requirements will be about three percent higher for that operation.

CONCLUSIONS

1. The optimum condensing and evaporating temperatures, for the given site location, using the assumptions in this paper, is about 55°F and 65°F respectively. The actual optimum operating temperatures may be somewhat different from these, as they are restricted by the ammonia flow rate the plant can be economically designed for.

2. The optimum seawater velocity is a trade off between minimum heat exchanger size and minimum pumping power.

3. The seawater temperature change across the heat exchangers has a direct bearing on the physical dimensions of the heat exchangers, and is thus integrally related to the whole design of the plant.

4. The optimum heat exchanger design consists of balancing the various heat exchanger operating characteristics within the physical limitations of a Sea Solar Power Plant.

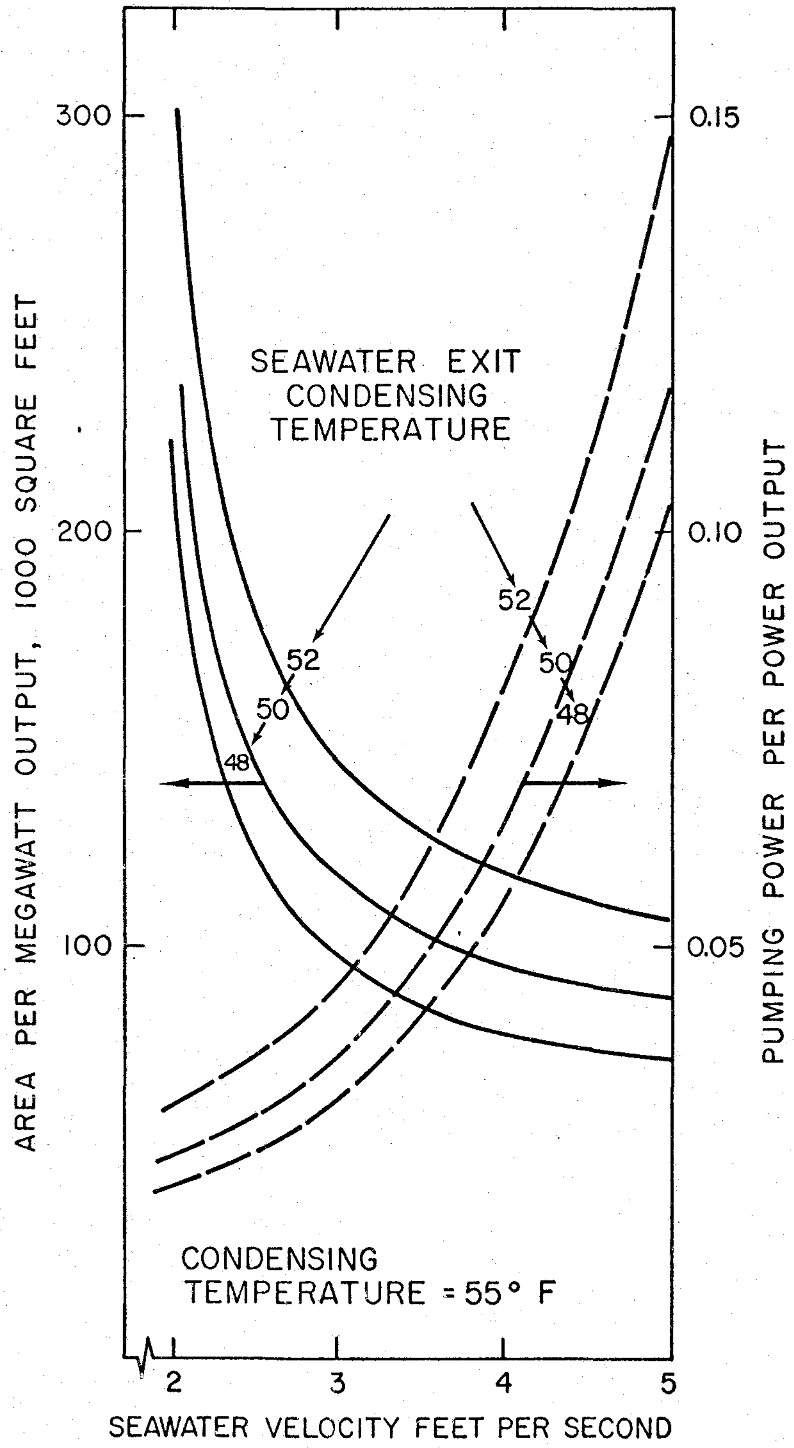


FIGURE 8. Affects of Increased Velocity on Pumping and Condensor Area Requirements

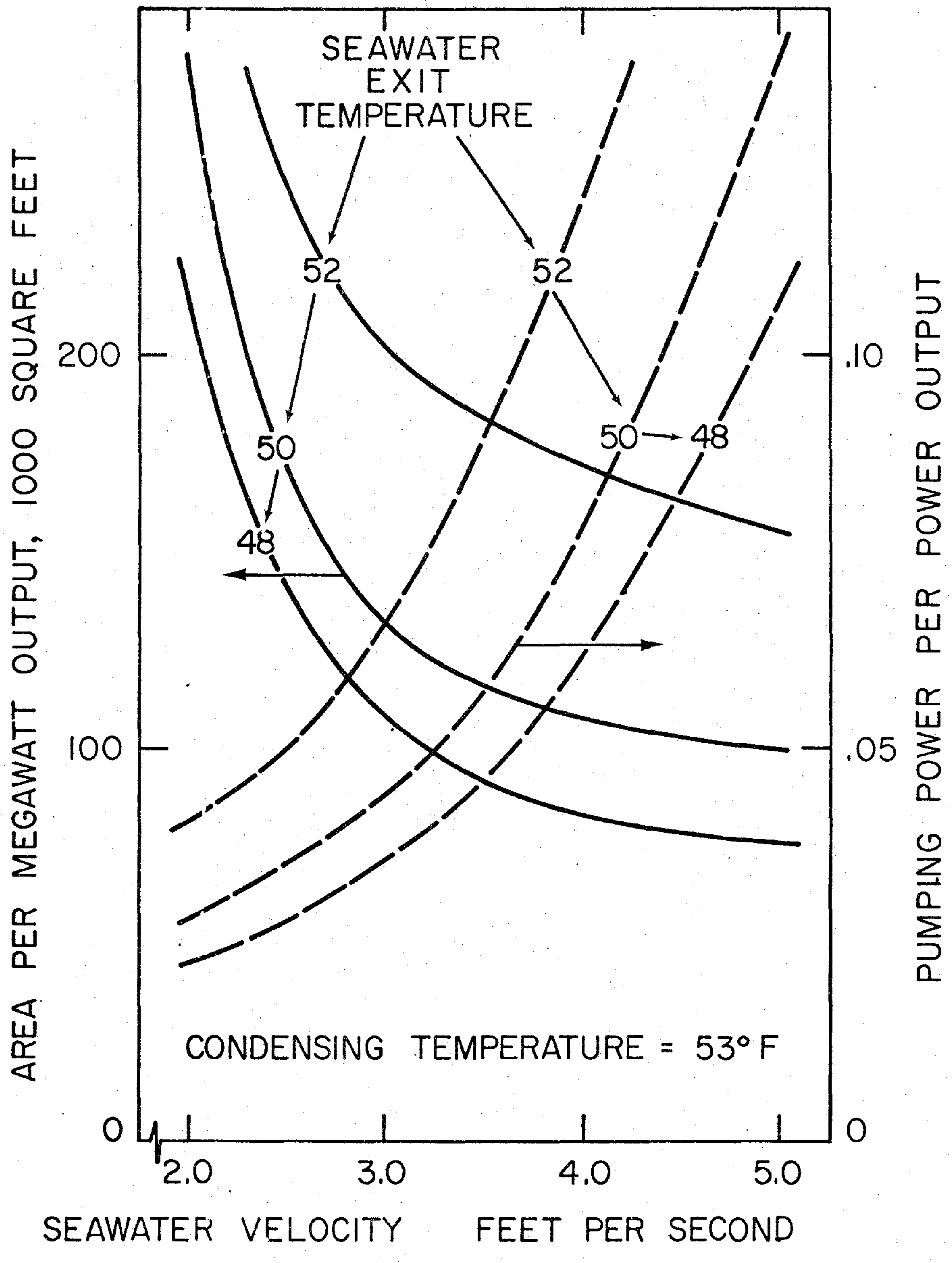


FIGURE 9. Affects of Increased Velocity on Pumping and Condensor Area Requirements

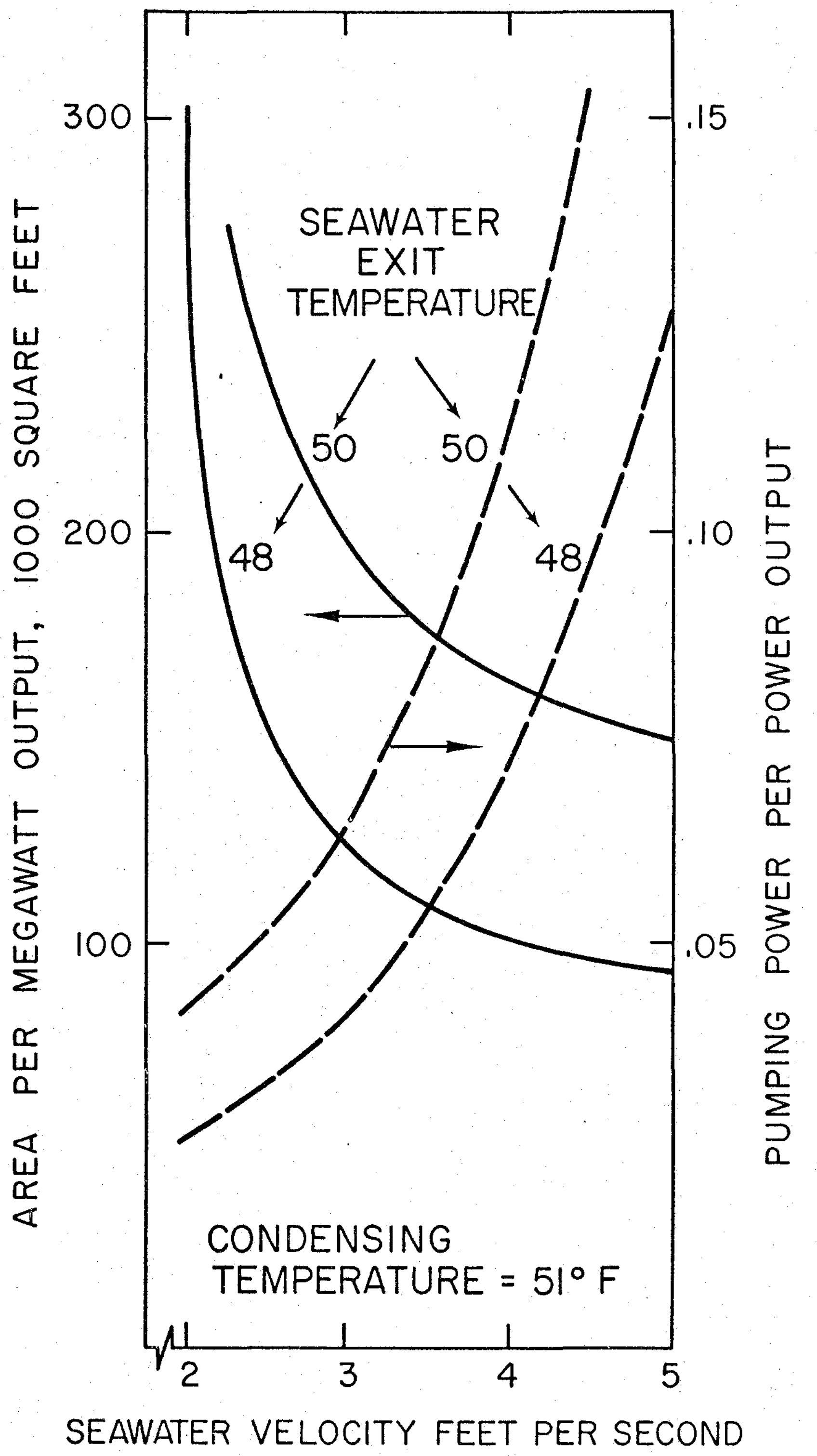


FIGURE 10. Affects of Increased Velocity on Pumping and Condensor Area Requirements

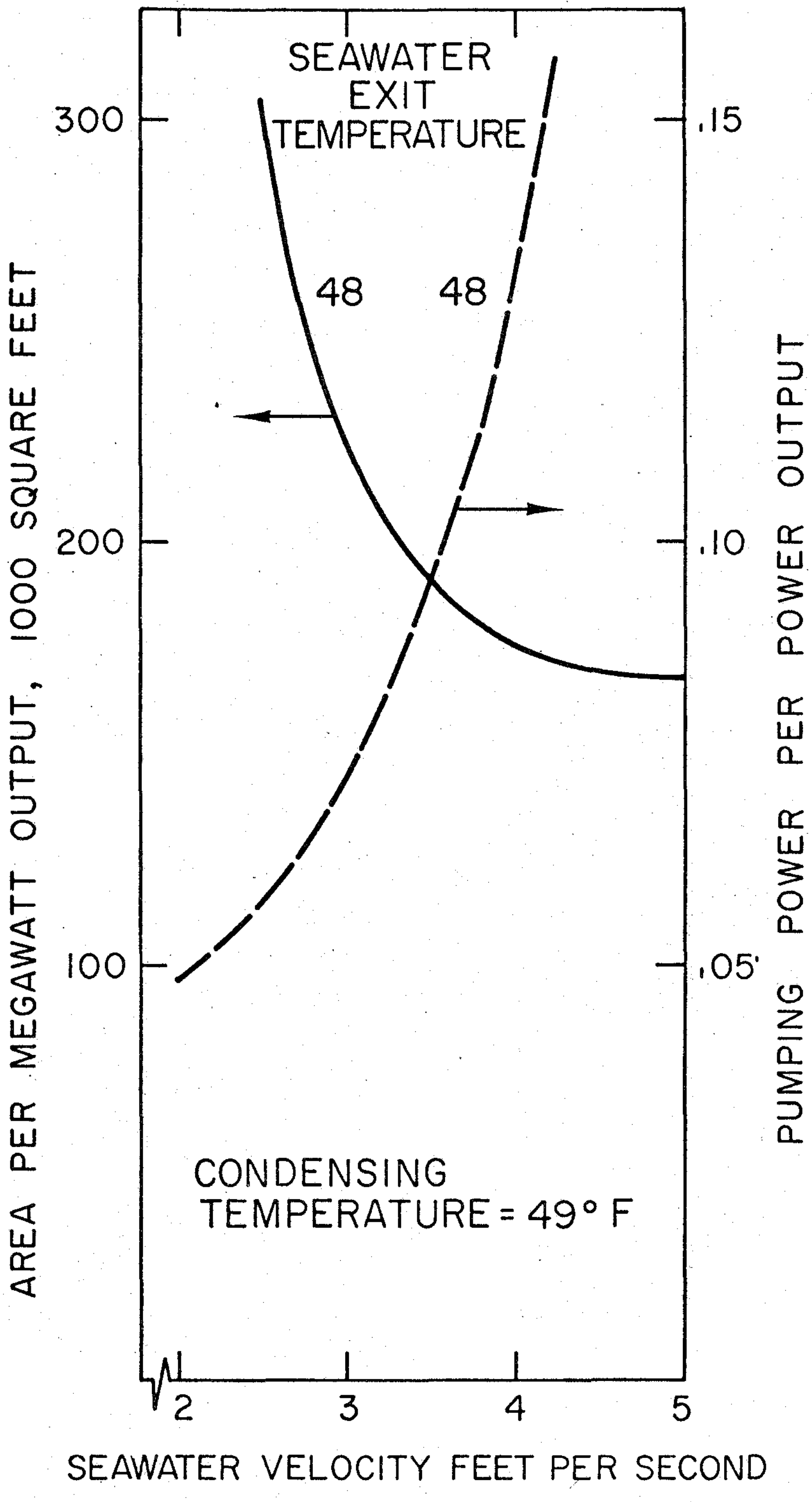


FIGURE 11. Affects of Increased Velocity on Pumping and Condensor Area Requirements

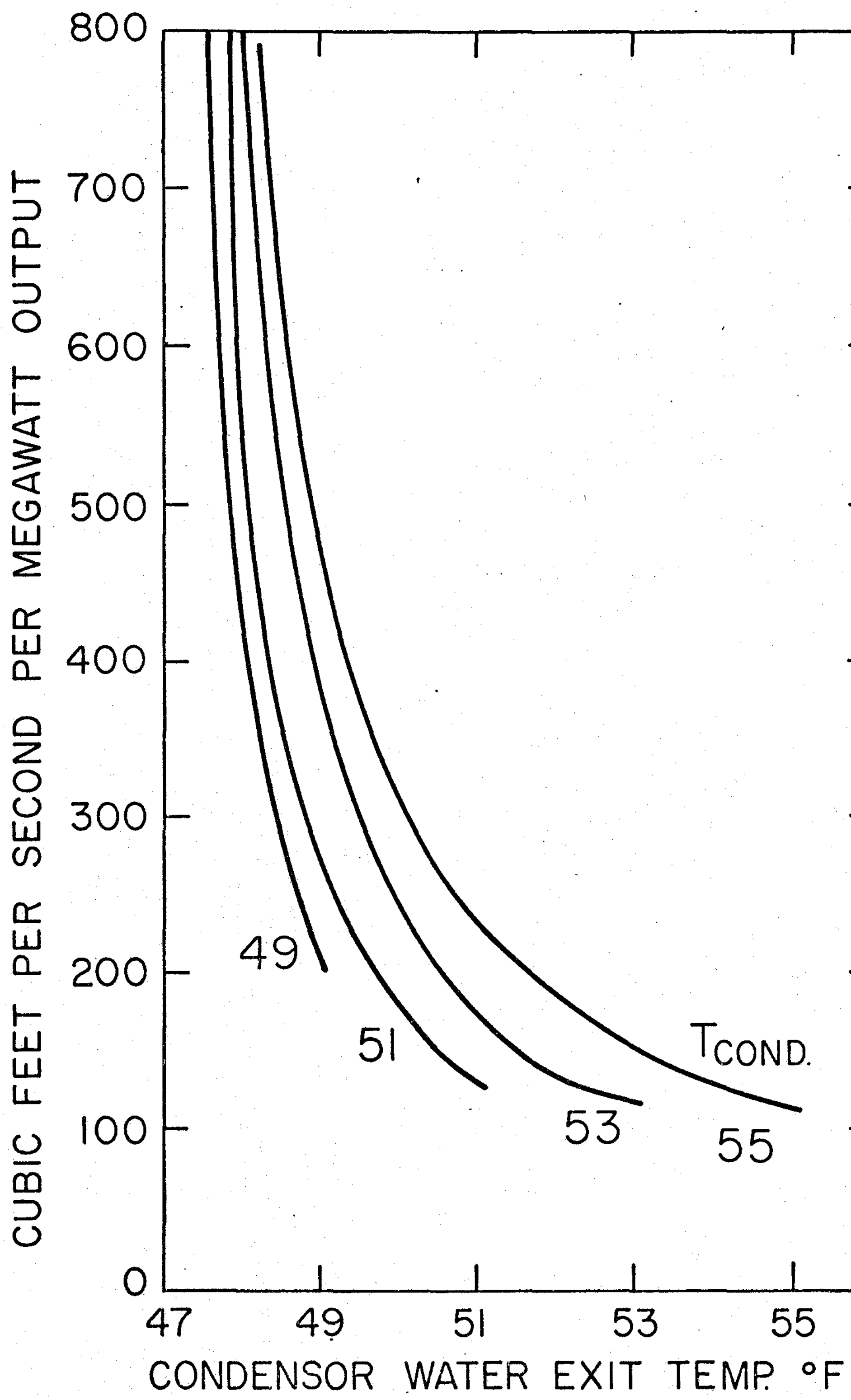


FIGURE 12. Seawater Flow Rate vs. Exit Temperature

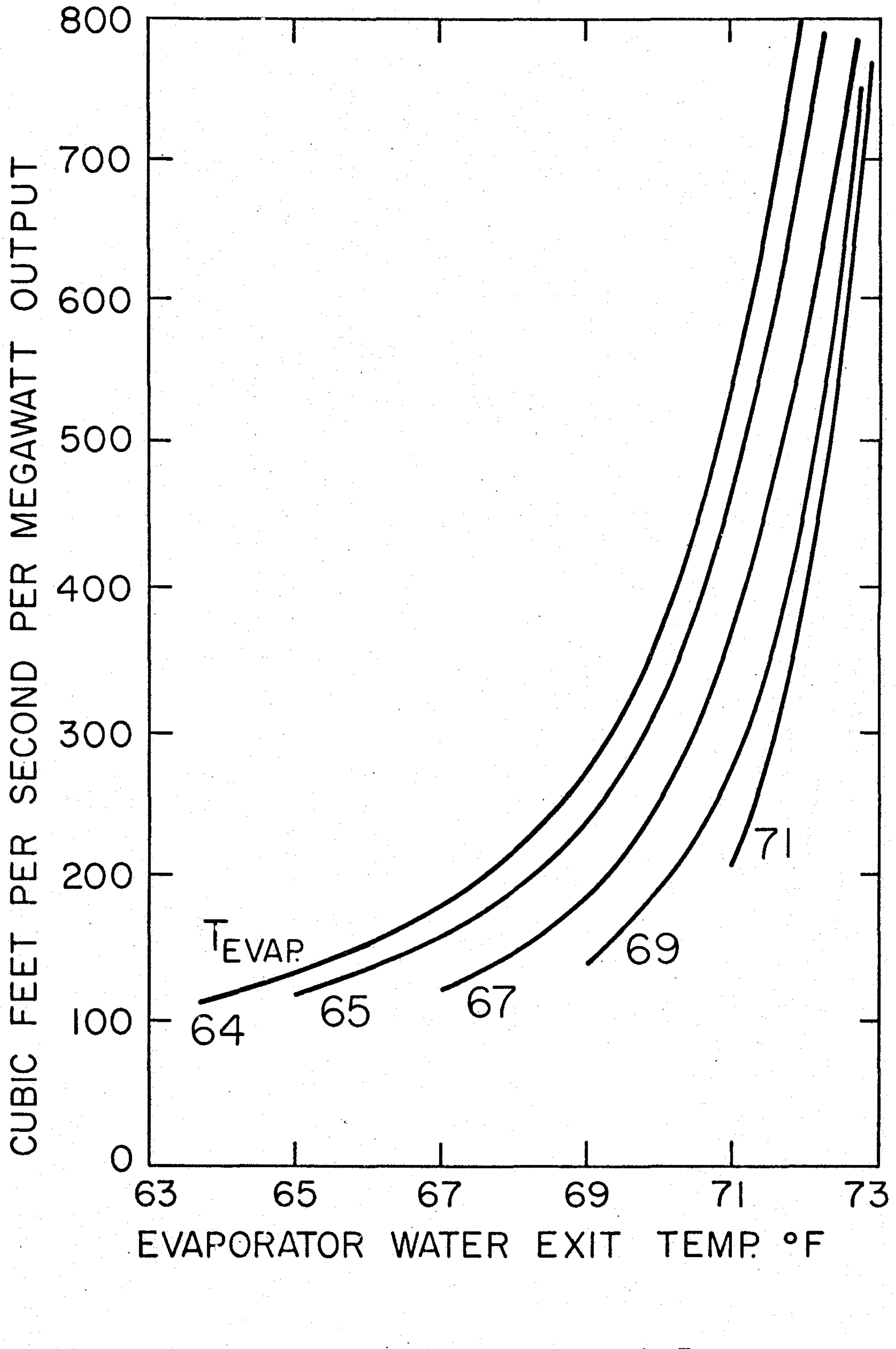


FIGURE 13. Seawater Flow Rate vs. Exit Temperature

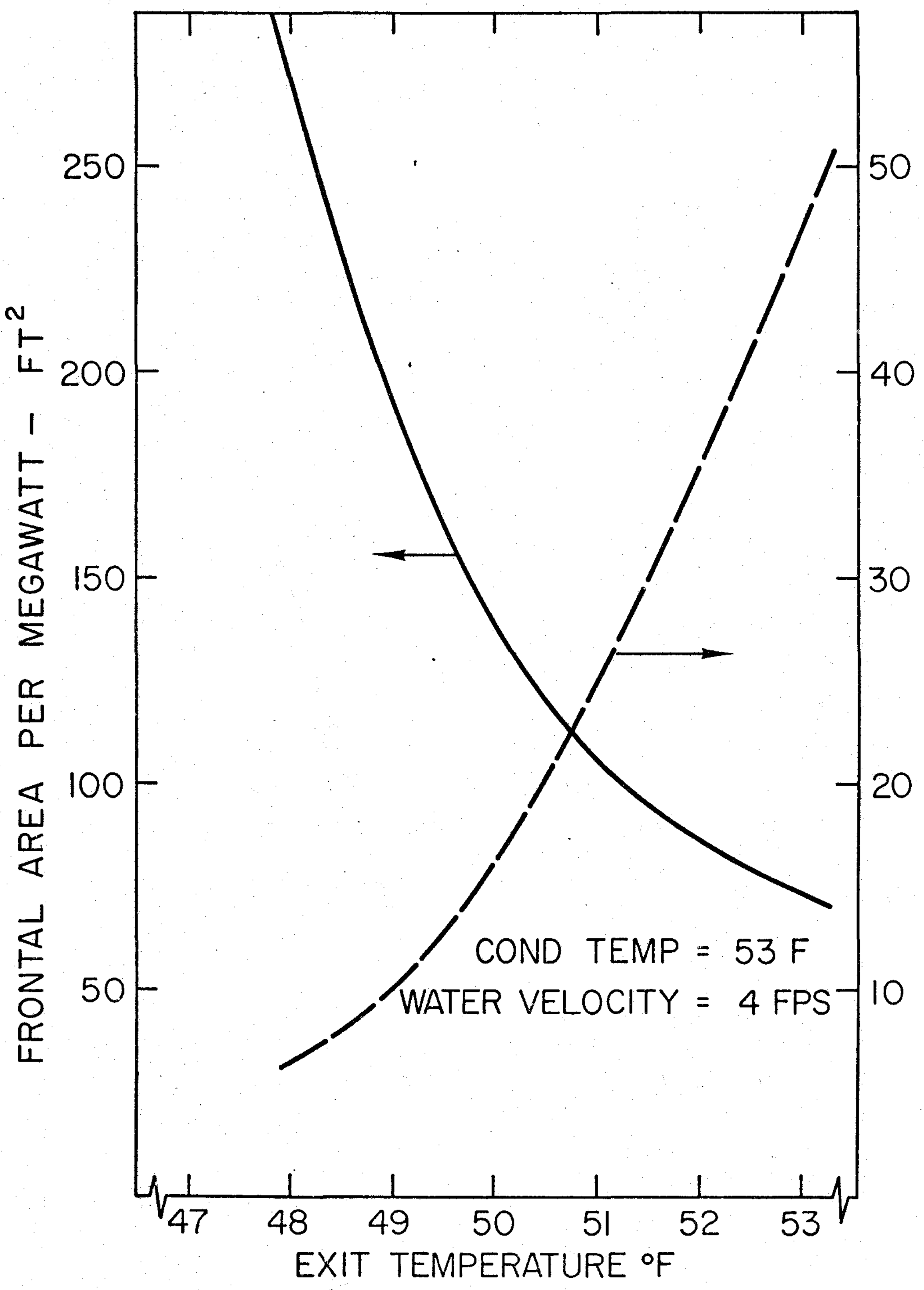


FIGURE 14. Condensor Frontal Area and Length for Varying Exit Temperatures

REFERENCES

Anderson, J. Hilbert. "Heat Exchangers for Sea Solar Power Plants." Technical Report NSF/RANN/SE/GI-34979/TR/73/15. Sept, 1973.

Anderson, J. Hilbert. "Turbines for Sea Solar Power Plants" Proceedings - Solar Sea Power Plant Conference and Workshop. Proceedings, Carnegie-Mellon University, Pittsburgh, Pennsylvania. June, 1973.

Godard, Jepson, Bothwell & Kane. The Corrosion of Light Metals. New York: Wiley & Sons, 1967.

Kays & London. Compact Heat Exchanger. McGraw Hill, 1965.

Olson, Pugger, Shippen & Avery. "Preliminary Considerations for the Selection of a Working Medium for the Solar Sea Power Plant." Solar Sea Power Plant Conference and Workshop. Proceedings, Carnegie - Mellon University, Pittsburgh, Pennsylvania. June 1973.

Van Wylen - Sonntag. Fundamentals of Classical Thermodynamics. New York: Wiley & Sons, 1965.

Welty, Wicks & Wilson. Fundamentals of Momentum Heat and Mass Transfer. New York: Wiley & Sons, 1969.

SEA SOLAR POWER PLANT - PLANT SIZING

by

Don Debok & Jim Leshuk

ABSTRACT

The size, weight, and internal power requirements of a 25 megawatt, gross, sea solar power plant were investigated. The design plant selected was an underwater plant to act as a shore side electric feeder. Equipment for plant operation was considered, and items of critical size and weight were selected. The evaporator and condenser were investigated as the primary space consideration. The cold water pipe and hull were the primary weight considerations. The investigation showed the plant to be fairly large in relation to its gross output, but within state-of-the-art. The resultant configuration is very adaptable to multiple units or mating with other devices. Plant power requirements for the various production and life support systems is investigated and compared to unit gross plant output.

SEA SOLAR POWER PLANT - PLANT SIZING

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Don Debok & Jim Leshuk

INTRODUCTION

Overall plant size of a Sea Solar Power Plant is one of the major questions still to be answered in the minds of technical people. Before they will consider the idea to be of merit they must believe that it can produce reasonable power from a size conceivable to them. To this end the size, weight and internal power requirements, of a 25 megawatt (Net) plant were evaluated.

To make this evaluation certain input decisions were required. This input was considered to be the result of a separate decision making process, and includes decisions for: a submerged plant, ammonia as a working fluid, an Electrical Feeder Plant, and 25 megawatts net power.

In evaluating the size of a submersible plant, it is important that overall weight also be determined. Plant volume must provide neutral buoyancy. Adjustable ballast must be sufficient to absorb weight variations in the plant.

Beyond the initial input data, decisions more directly relating to size and weight were made.t These included:

1. Intake pipes should be hinged at the plant to reduce the need for overly thick pipe wall.

2. The design calls for general operation by 12 men, with the ability to support 24 men for periods of repairs. Estimated stay time for each crew is 21 days, rations will be allowed for 42 days.

3. The capability for periodic surfacing for routine supply purposes must be provided at zero or reduced power output. This down time will be 12 hours maximum.

4. A majority of required maintenance will be performed on site. The minimum on site design time is five years. Routine maintenance must be allowed for while the plant is operational. Major repairs will be by subsystem modular replacement.

The identification of systems necessary to plant operation was the first step in actual sizing of the plant. These items were classified according to the manner in which they might effect the plant size and weight. Table 1 summarizes this classification and estimates the importance of each system within the classification (five being of greatest importance). The major efforts were undertaken in the sizing of the boiler, condenser and cold water pipe. A conceptual layout was made to estimate the weight of the plant hull and the ballast tanks. Other items were estimated from off the shelf equipment or from research information.

PLANT VOLUME ESTIMATION

The estimation of total plant volume is important to the overall cost and feasibility of a Sea Solar Power Plant. It is also important in determining plant layout and drag forces for mooring and anchoring. To determine overall size, two groups of items were considered. 1) Those whose size varied with plant operating parameters; and 2) Those of size that is relatively constant, being based on initial decisions.

Items whose size varies are the boiler, condenser and water intake pipe. The size of the boiler and condenser is a function of intake water temperature (T_{in}). Water exit temperature (T_{exit}), flow velocity (V), and ammonia condensing temperature (T_{cond}). T_{in} is established by site selection. In order to minimize total plant size a general layout was selected. This arrangement, shown in Figures 1 and 2, was chosen for its:

1. Streamlined shape
2. Adequate space for buoyancy
3. Optimum position of machinery space
4. Ease of condenser and pump maintenance
5. Space effective means of changing water direction

Calculations for total volume of the condenser and evaporator units were based on internal surface area per output megawatt (S.A.), required pumping power (MW_p/MW_g) and flow rate (Q) obtained from data calculated by Dan Ladd in a preceeding report. To account for pumping costs in both the evaporator and condenser, Q and S.A. must be increased by

$$2X\ MW_p/MW_g \text{ or,}$$

New Surface Area (NSA) =

$$(1 + 2\frac{MW_p}{MW_g})SA \ldots\ldots\ldots (1)$$

and Gross Flow Rate (Q_G) =

$$(1 + 2\frac{MW_p}{MW_g})Q. \ldots\ldots\ldots (2)$$

Then Total Heat Exchange Volume (THEV) may be expressed as,

$$THEV = NSA/48ft^3/1000ft^2SA \ldots . (3)$$

Since water occupies 44% of this volume, Total Waterside Volume (TVW) = (.44)(THEV). Also, condenser length (L) may be expressed as,

$$L = TWV/Q_G/V = TWV/PA_W$$

where PA_W = the project area of the waterside of the heat exchanger.

From this, Total Projected Area (TPA)

$$= THEV/L \ldots\ldots\ldots (4)$$

and Total Boiler Unit Volume (TV_B)

$$= TPA(L + 25). \ldots\ldots (5)$$

Similarly, the Condenser Unit Total Volume (TV_C) may be expressed as, TV_C

$$= (TPA + PA_W)(L + 20 + \frac{PA_W}{[(4\pi)(TPA)]^{\frac{1}{2}}}) (6)$$

Figures 3, 4, 5 and 6 illustrate the effects of varying these parameters. Selected final operating levels were:

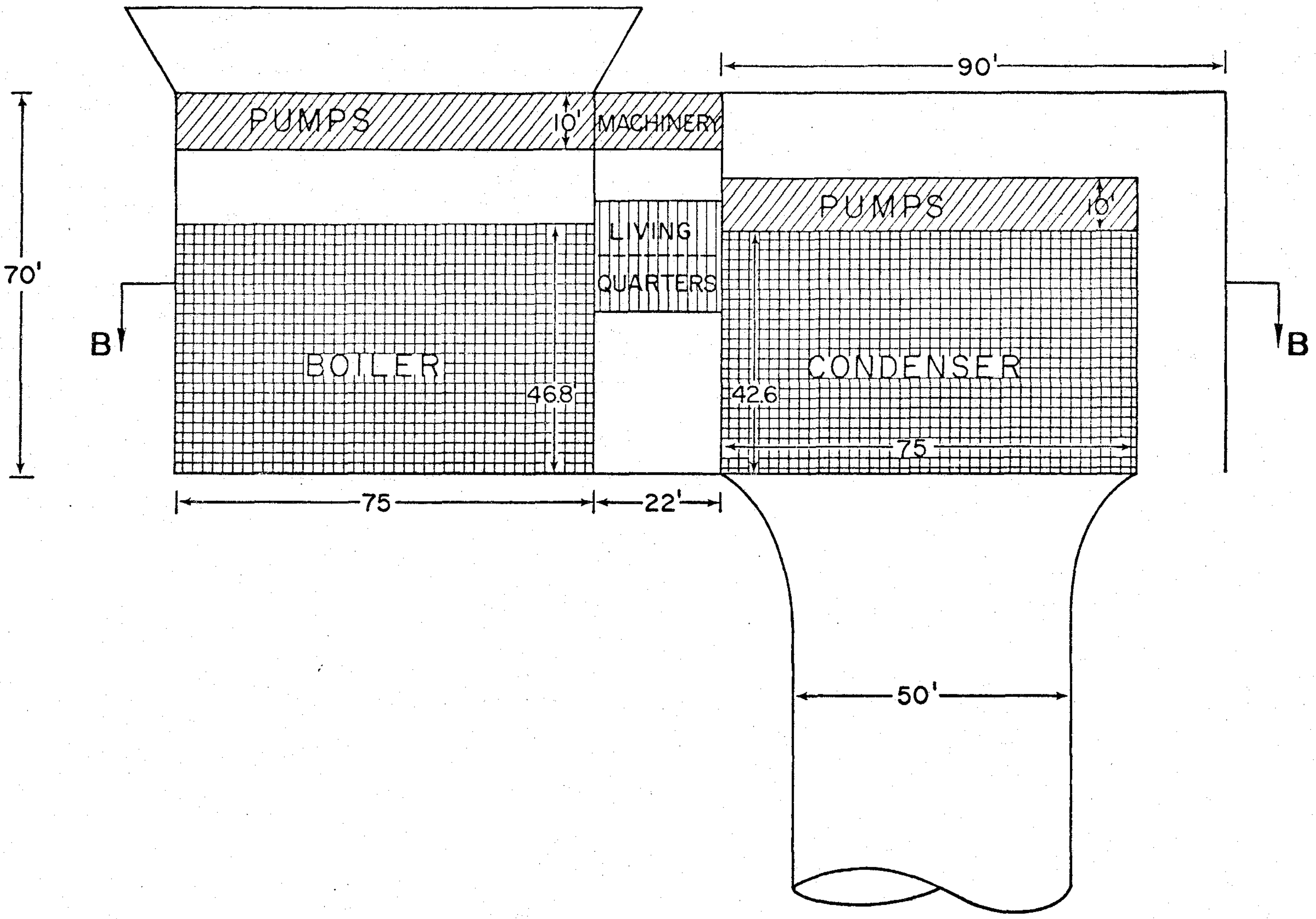

FIGURE 1. Profile Plant Design

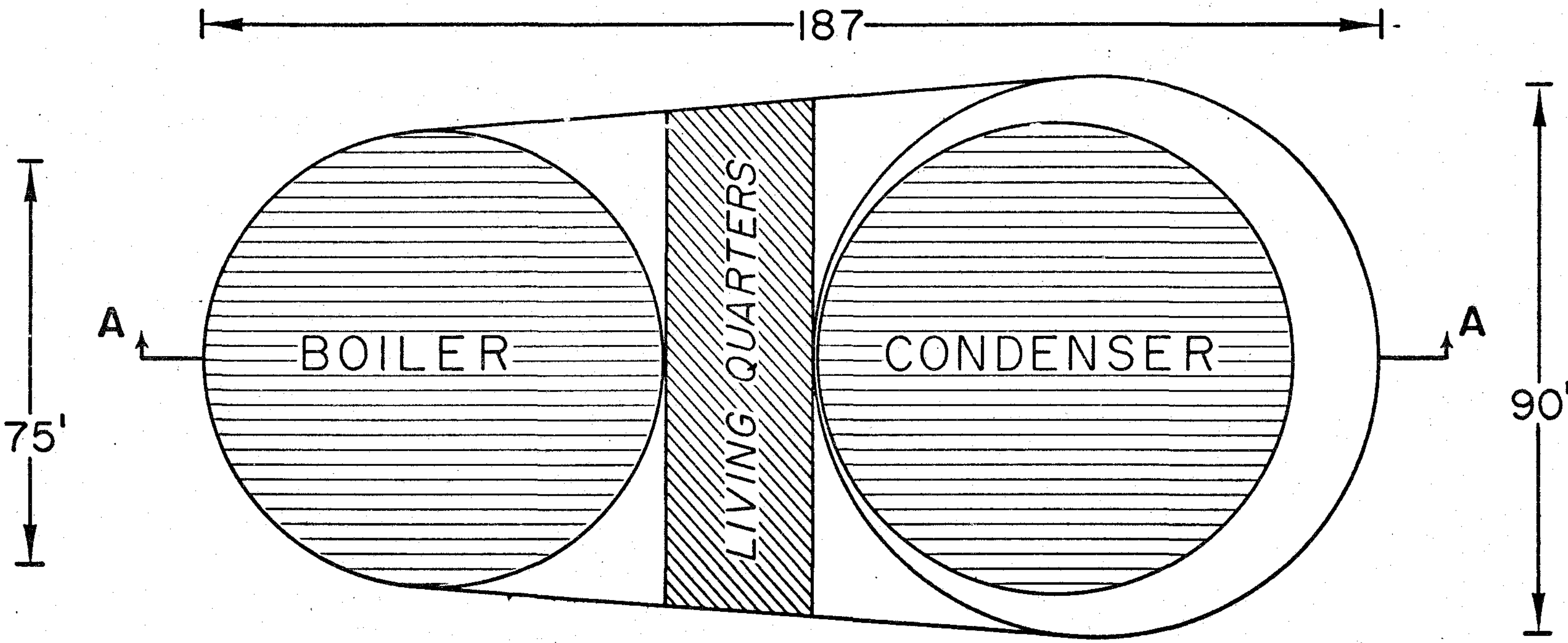

FIGURE 2. Plant Design - Plan View

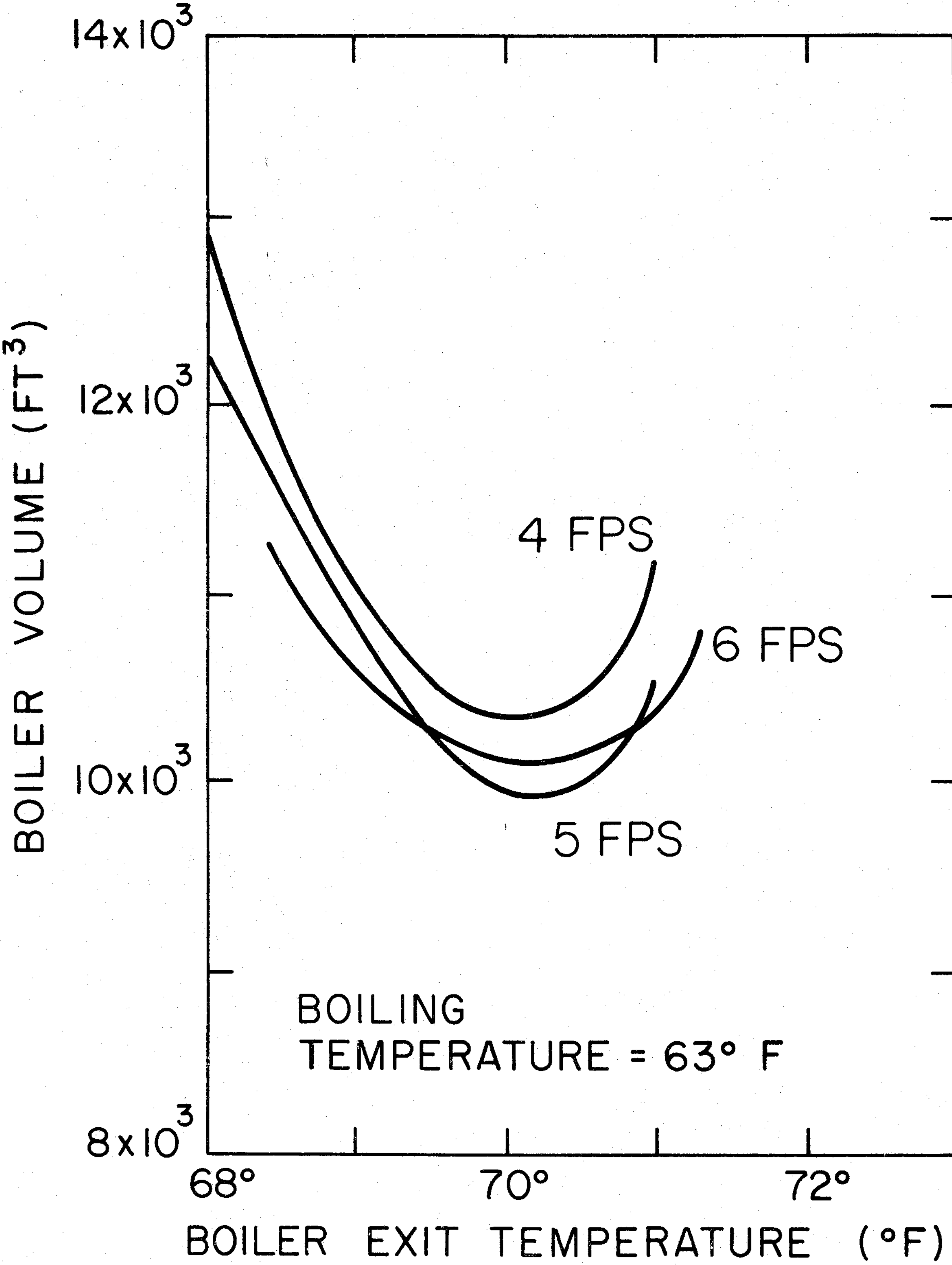

FIGURE 3. Boiler Volume vs Exit Temperature For 63° F Boiling Temperature

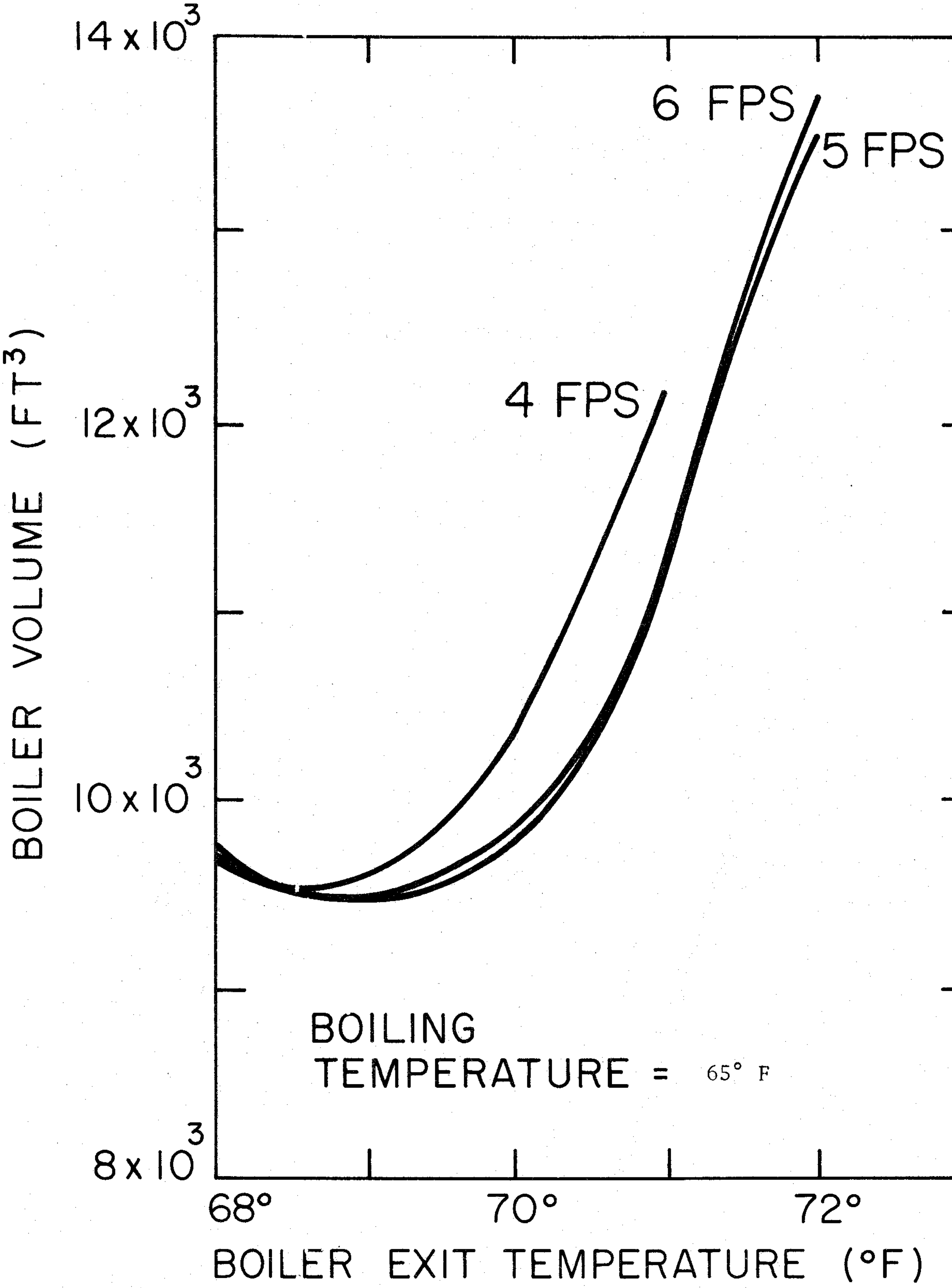

FIGURE 4. Boiler Volume vs Exit Temperature For 65° F Boiling Temperature

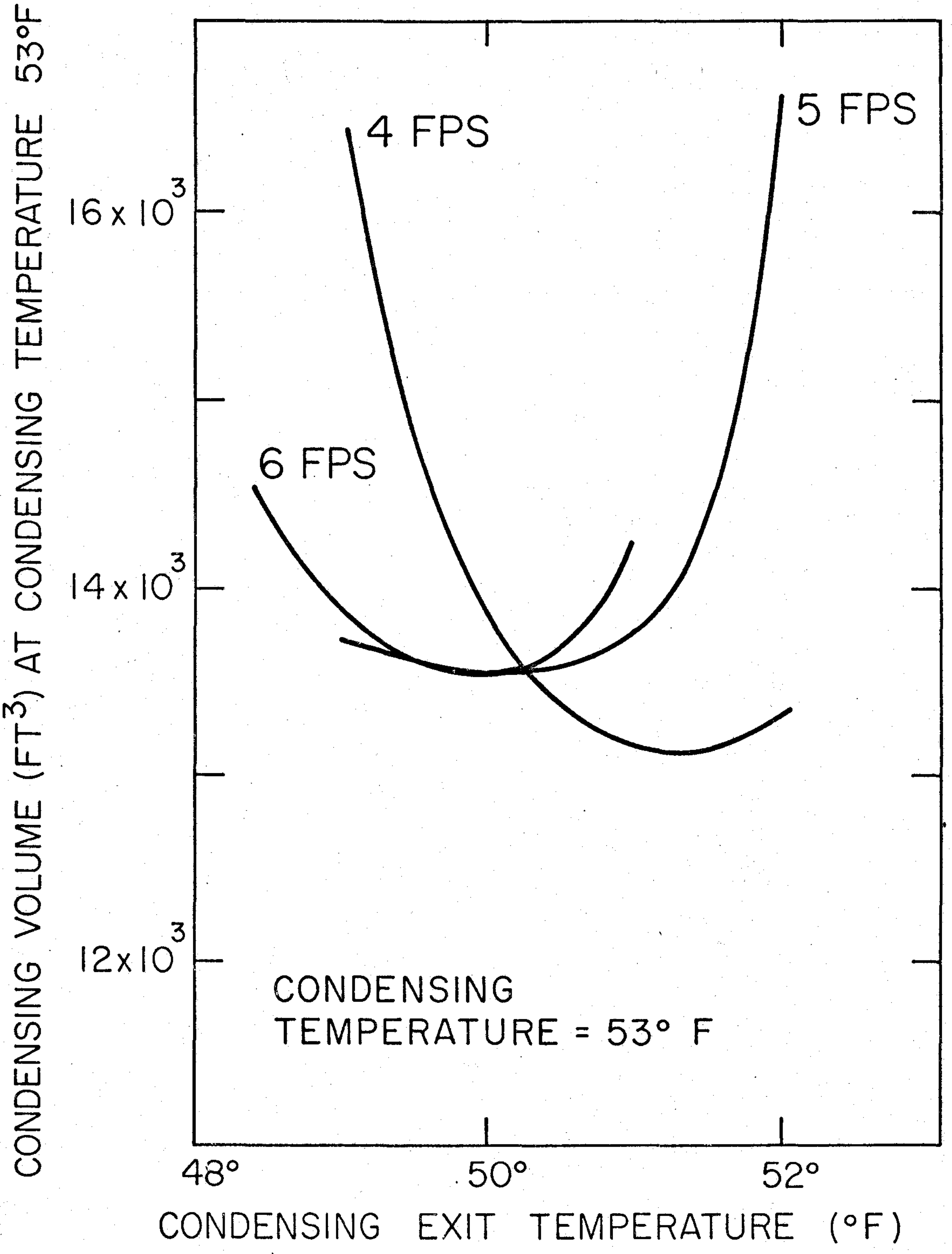

FIGURE 5. Condenser Volume vs Exit Temperature For 53° F Condensing Temperature

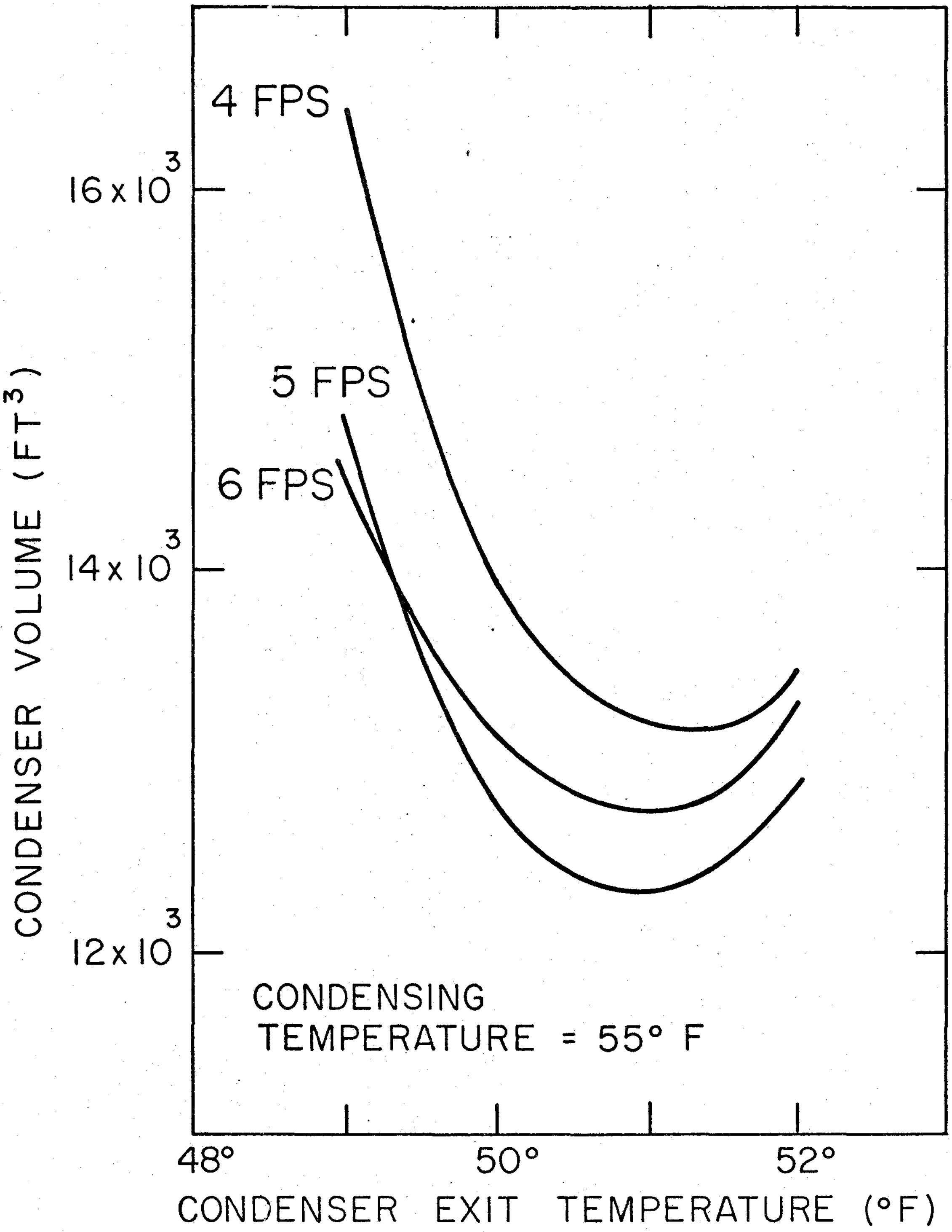

FIGURE 6. Condenser Volume vs Exit Temperature For 55° F Condensing Temperature

TABLE 1. PLANT SYSTEM CLASSIFICATION

Area of Interest / Item	Power Requirement	Added Volume Added Buoyancy	Added Volume Neutral Buoyancy	Added Volume Added Weight	Added Weight Only
Boiler			5		
Condenser			5		
Turbine				4	
Generator	2			4	
Transformers	1			1	
Air Compressor	1			1	
Ammonia Pumps	4			3	
Water Pumps	5			4	
Heating/Air Cond.	3			1	
Life Support	3			3	
Dewatering Equipment	1			1	
Communication Equip.	1			1	
Emergency Escape				1	
Fresh Water Production	3			3	
Fresh Water Storage				3	
Batteries	1			2	
Living Space	3	4			
Hot Water Tank	2			1	
Tools/Spare Parts				3	
Pressure Hull					4
Cold Water Pipe					5
Fire Fighting Equip.	1			2	
Control Station		3			
Ballast Tanks		4			
Anchor Hand. Equip.	2				
Air Storage		2			
Plant Hull					5
Ammonia Piping				4	
Lighting	1				

Boiling Temperature	65°F
Boiler Water Exit Temperature	68°F
Condensing Exit Temperature	55°F
Condensing Water Temperature	51°F
Flow Velocity	5fps

This selection results in a plant of the dimensions shown in Figures 1 and 2. Total volume of this configuration is 1.18×10^6 ft^3.

The sizing of the cold water pipe is dependent on the flow rate Q established by the operating parameters. The length is established by the site selection. The pipe diameter is dependent on the flow rate and flow velocity. The decision on size requires balancing the need for smaller diameter to reduce drag and construction cost against the need for lower pumping cost. As indicated later in this report, the pumping loss due to the density difference is the major pipe loss up to four feet per second. Beyond this point frictional losses become increasingly important.

The pipe may be sized to a minimum diameter by balancing the dollars saved in construction by reducing the pipe size with dollars spent increasing condenser size to provide additional required pumping power. This analysis requires a determination of the total power loss (P_L) for the established flow rate for several velocities.

$$P_L = \left(\frac{fL}{D} + \Sigma K + 1 \right) \frac{v^2 \gamma Q}{2g} \quad \ldots . (7)$$

where

f = friction factor
L = pipe length
D = pipe diameter
ΣK = sum of the minor losses
V = flow velocity
g = acceleration of gravity
γ = specific weight
Q = flow rate

To obtain the new condenser volume required (Vol_{gr}),

$$Vol_{gr} = \frac{P + P_L}{P} Vol_{cond} \quad \ldots \ldots (8)$$

where

P = Net power of the plant
Vol_{cond} = Original condenser volume

To obtain the new pipe diameter (D_{gr})

$$Q_{gr} = \frac{P + P_L}{P} Q$$

$$\text{and, } D = \left[\frac{4 \ Q_{gr}}{\pi V} \right]^{\frac{1}{2}} \quad \ldots \ldots \ldots \ldots (9)$$

To compare the costs, condenser volume was converted to weight per cubic foot of condenser (C.C.) and pipe diameter was converted to weight/foot diameter (P.C.). Since there is no fuel cost involved, the construction costs may be compared directly. Maintenance and anchoring costs are considered small per pound of material. A detailed analysis of cost per pound of material fabricated was not undertaken as it is intuitive that condenser fabrication is considerably more expensive. Figure 7 demonstrates the trend of the lowest cost (P.C. + C.C.) for various velocities and ratios of C.C./P.C.

Actual sizing of water pumps was not undertaken. A set of propeller pumps of this capability within the space allowed was considered within state-of-the-art capabilities.

Items fixed in size by initial conditions were living quarters and machinery spaces. Fraser (1968) gives examples and compilation of confined space situations. In general, the cases for confinement space for three weeks were space ship examples. Due to the repeated nature of the manning and desire for crews' comfort. Chamberlain's (in Fraser, 1968, p. 456) recommendation of 2000 ft^3/man was adopted. To allow additional space for maintenance personnel, approximately 30,000 ft^3 was allowed for living space. The author's (Debok) shipboard experience confirms that 30,000 ft^3 is adequate for up to 24 individuals. This

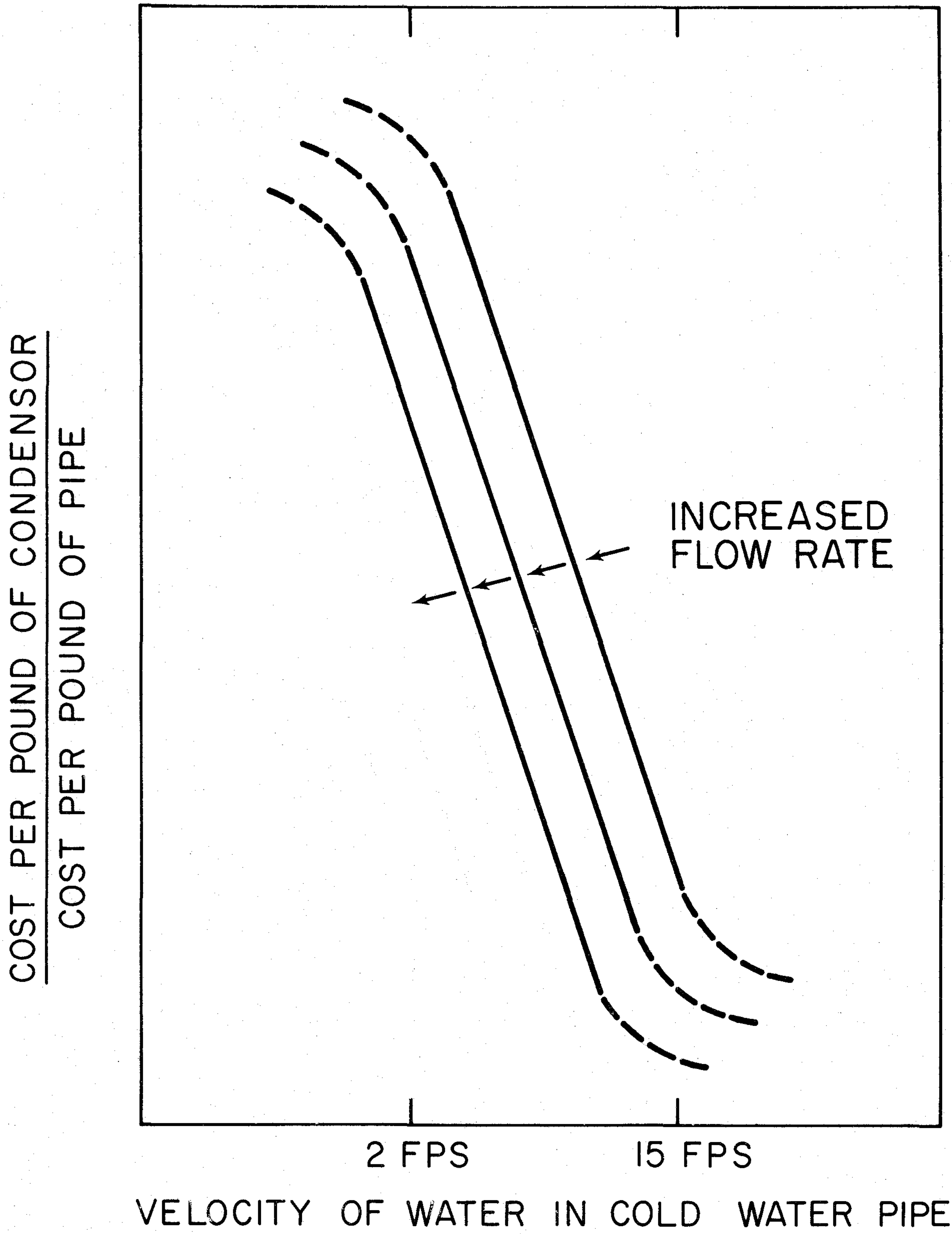

FIGURE 7. Construction Cost

space would include all berthing, messing, recreation, food storage and sanitary spaces.

In the initial layout the same space was allowed for living and machinery areas. Table 2 lists major machinery items and their volume. The allotment of 30,000 ft appears adequate but perhaps cramped. In general, added space would be free volume converting adjustable buoyancy to fixed buoyancy. The relationship of these two does not appear critical to the design at this time.

PLANT WEIGHT ANALYSIS

Accurate plant weight estimation is especially critical for the Sea Solar Power Plant due to the requirement for neutral buoyancy. The basic analysis approach here was similar to that for volume estimation. Major plant subsystems were identified as to their relative importance and state of refinement with the most analytical effort being spent on those potentially large systems not available off the shelf. Some iterative calculations were necessary, but the system was, in general, proven quite stable in this respect. Large changes in mass and volume affected only mooring and trim systems. These in turn, only impacted mass and volume estimates slightly. Ultimately, identification of mass, material, and configuration will be of great value in the capital cost analysis of the prototype design.

The constituent subsystems for which weight and displacement estimates and calculations were to be performed were: heat exchangers and protective shell, ammonia boiler, life support hull, serviceable machinery hull (which includes the bulk of prime movers, auxilary pumps, switching, transformers, and controls), cold water pipe (which can be considered independently by virtue of its design), external machinery, external storage, structure, exostructure, ballast systems, and miscellaneous. Miscellaneous items were those which it was felt contributed negligible amounts to the total weight and/or displacement. These included positioning controls, communications interfacial systems, and external maintenance systems.

HEAT EXCHANGERS

Using the aluminum heat exchanger design presented by Dan Ladd in a previous paper, a linear variation of ammonia quality between state points, six mil thick aluminum, a 1.34:1 corrugation ratio, and 1/4" spacing for both seawater and ammonia sides were assumed. On that basis, the weight of the condenser including ammonia was calculated at 23.79 lb_m/ft^3 of heat exchanger, the weight of the evaporator at 23.94 lbm/ft^3 of heat exchanger, and the displacement of both units at 0.560 ft^3/ft^3 of heat exchanger.

For our 25 mw net output plant, condenser heat exchanger volume was determined to be 186,000 ft^3 and the boiler volume to be 170,000 ft^3. Thus the total weight of the condenser is 4.43×10^6 lbs with a displacement of 1.04×10^5 ft^3. The evaporator weight is 4.07×10^6 lbs with a displacement of 9.52×10^4 ft^3.

PRESSURE HULL

While significant contributors to weight, such as prime movers and distribution and control systems, are conceived to be housed in a pressure hull, the most important contribution of the pressure hull to the weight analysis is in terms of displacement. This parameter in turn is influenced primarily by human factor considerations, which

TABLE 2. PLANT MACHINERY SYSTEM VOLUMES

Item	Volume	Source
Turbine	700 ft^3	Anderson
Generator	2000 ft^3	Standard Handbook for Mechanical Engineers estimate
Fresh Water Storage	432 ft^3	Volume of 4000 gals
Fresh Water Production	35 ft^3	Personal observation (DeBok)
Life Support Equipment	200 ft^3	Personal observation (Arneson)
Ammonia Pump	500 ft^3	Estimate of 50 HP pump
Air Compressor	300 ft^3	Estimate of 25 HP air compressor

were researched for the assumed 12 man crew. Beyond this, other major studies were conducted of power, gas, food, and water storage, generator and turbine sizing, life support equipment, and general warehousing. The studies are arbitrarily divided into two categories, those housed in the crew hull, and those housed in the machinery hull.

Crew Hull Interior: Major weights, packing factor, and volume requirements of items contained in the crew hull are shown in Table 3. These values were chosen primarily from ship and submarine outfitting references and experience.

The largest impact on displacement comes from living space considerations for the work crew. Greatest contributors to weight are water and food storage.

Total weight of the interior furnishings are estimated at 144,300 lbs. A 22 ft diameter hull was arbitrarily selected for the pressure hull. Allowing for the total internal volume of 28,658 ft^3, this yields a length of 75.4 ft. Assuming an inch thick HY-80 hull with three inch atends and 10% additional for bulkheads and decks (also serving as structural reinforcement), a total displacement of 29,187 ft^3 and weight of 424,200 lbs is achieved for the personnel hull.

Machinery Hull: The machinery hull contains all those components which need only occasional attendance and life support systems which might present a hazard or nuisance to the work crews. The primary contributors to weight were considered to be the generators, turbines, and lead-acid emergency power storage batteries.

Generator information was obtained from standard AC central power plant equipment. This resulted in a weight of 237,200 lbs, including the exciter. This weight is at least an order of magnitude greater than any other component except lead acid emergency power storage batteries. For a 1500 amp-hour reserve, 240,000 lbs of batteries are required.

The propane turbines are small in comparison as only an eight foot diameter turbine is required for 30 mw output. Thus if the weight of the batteries and generators are considered to constitute the bulk of the weight (approximately 90%), the weights of the turbines and all other components can safely be neglected. If the same size hull is picked for the machinery sphere as the personnel hull (for purposes of standardization and economy), the same displacement is achieved, 29,187 ft^3 but the new weight is 1,209,400 lbs as shown in Table 4.

COLD WATER PIPE

Weight analysis of the cold water pipe is perhaps the most important single calculation. The determination is simplified by the existence of precedents, particular hydro-power penstock design criterion. Nonetheless, some examination of the assumptions behind these criterion and some modification of them for seawater service was necessary.

Steel was assumed to be the best material for pipe construction from the standpoint of economy, corrosion, fabrication, strength and fouling. The top joint was assumed to be flexible, thus eliminating bending stresses. Fabrication was assumed to be an onshore process with the plant towed to site. Weld efficiency of 90% and a design stress of 15,000 psi was assumed.

Stress conditions examined were buckling under pressure differential, both for new and corroded pipe, and collapse from the structures own weight while being fabricated on land, calculated for the new pipe only, since removal from the water is not forseen.

TABLE 3. PERSONNEL HULL WEIGHT TABULATION

Component	Space Req't (ft^3)	Weight (lbs)
Freshwater Production	35	6,000
Freshwater Storage	432	24,000
Living Space & Personnel Furn.	26,000	4,800
Hot Water Production & Storage	40	2,500
Refrigeration	300	15,000
Contaminant Control	196	2,000
Emergency Breathing Equipment	40	1,200
Carbon Dioxide Scrubbing	20	200
Oxygen Supplies & Gen.	125	7,000
Food Storage	120	72,000
Food Preparation	600	6,000
Ventilation	300	400
Sanitary Facilities	400	2,000
Communications	10	200
Controls	40	1,000
Sub-total	28,658	144,300
Hull (Includes airlock & escape tube)	529	279,900
Total - Personnel Hull	29,187	424,200

TABLE 4. MACHINERY HULL WEIGHT TABULATION

Component	Volume (ft^3)	Weight (lbs)
Turbines (2, 40 kw each)	*	*
Generators (2, 40 kw each)	2,000	605,000 (2)
Auxiliary Pumps & Compressors	*	*
Switch Gear	*	*
Transformers (Local)	*	*
Cooling System	*	*
Piping	*	*
Cabling	*	*
Power Storage	750	240,000
Tools & Shop	*	*
Warehousing	*	*
Ballast Tank Compressor	*	*
Ammonia-Boil-Off Tanks	*	*
		x 1.1 [1]
Internal Totals	28,658 [2]	929,500
Hull (Includes Airlock & Escape Tube)	529	279,900
Total	29,187	1,209,400

[1] Weight of all minor items assumed to be 10% of the total weight source.

[2] Calculated hull internal volume, not the sum of component volumes.

The thickness dictated by the pressure differential criterion is quite small (on the order of 0.5 inches) since maximum pressure differential from suction head created by the pumps is only about three feet (approximately 1.5 psi). This value is somewhat dependent on pumping velocity.

Corrosion rates are about 10 mils per year (mpy) on the inside and four mpy on the outside due to differing local water velocities inside and outside the pipe.

The thickness dictated for land construction was calculated from methods discussed by Sarcaria (1958). This value was determined to be 1.55 inches. Thus, if the pipe were to corrode at 14 mpy, it would be 75 years before a pipe designed to this criterion would fail from the first criterion.

Assuming a thickness of 1.55 inches, an inside diameter of 50 feet, and a length of 737 feet with the last 50 feet radiused into a bell shape (using a 50 foot radius) to give the smooth entrance assumed by power loss calculations, the displacement of the pipe would be 17,667 ft^3 and the weight would be 8,640,000 lbs.

STRUCTURE AND EXOSTRUCTURE

A rather simplistic approach was taken to the structure and exostructure containing the plant, tieing together sub-systems, and providing a faired shape.

The total area of surface surrounding the heat exchangers and pressure hulls was determined from the general layout. One inch thick epoxy reinforced with 85% glass fibers was assumed to constitute the exostructure, along with 5% of this volume being aluminum structural members and fasteners. Using this formula, every square foot of external surface contributed .0876 ft^3 to the displacement and 13.1 lbs to the weight of the plant.

For the 45,120 ft^2 of area of surface computed (exclusive of heat exchanger armor), the structure and exostructure displaced 3760 ft^3 and weighed 562,000 lbs.

The heat exchanger armor was computed using the same formula, but the surface area is more sensitive to plant siting parameters. For a 75 foot diameter and 42.8 foot high condenser, the armor displacement is 840 ft^3 and weight is 126,000 lbs. For a 75 foot diameter and 46.8 foot high evaporator, the displacement is 919 ft^3 and weight is 137,000 lbs.

The total mass of structure and exostructure constitutes roughly 5% of the weight of the plant and 7% of the buoyancy. In light of this, if more refined design information became available, some additional time spent on improving this estimate would be justified.

EXTERNAL EQUIPMENT

Equipment mounted external to the pressure hulls was assumed to include pumps for moving the cool and warm water, storage of large spares not affected by seawater or pressure, and oil filled transformers.

Five pumps, similar to Allis-Chalmers units designed for the city of Ottumwa water system, low head, high volume with fixed blades with enclosed electric motors were located above the evaporator heat exchanger. Five similar units were located above the condenser heat exchanger. The weight of the units used was 28,000 lbs each, with a displacement of 220 ft^3 each.

External storage was assumed to occupy 5000 ft^3 and have an empty weight of 30,000 lbs.

Oil filled transformers were assumed to be placed in compensated pressure housings outside the plant. Without more complete data on power distribution, the transformers were assumed to displace 10,000 ft^3 and weigh 640,000 lbs. To minimize the impact of this large unknown on the balance of the plant, they were assumed to be constructed to be neutrally buoyant. This estimate could use considerable refinement if an electrical distribution design was included in the prototype report.

MISCELLANEOUS ITEMS

Those subsystems identified but not quantified for purposes of the weight estimate include items which are either of such small magnitude as to represent insignificant numbers in the total displacement and/or weight. These are listed in Table 5 along with their expected order of magnitude. A note on the mooring lines is in order. It was determined that the forces encountered could be handled with nylon line of practical diameters. Nylon line is nearly neutrally buoyant, thus it would not enter into an iterative process on weight and displacement calculations, despite large quantities needed in mooring.

WEIGHT SUMMARY

Totals for the weight analysis for the various components are shown in Table 6. The net weight of the plant is 11.27×10^6 lbs and that of the cold water pipe is 8.64×10^6 lbs for a total weight of the entire structure of 19.91×10^6 lbs. Displacement of the various systems is also shown. However, these figures ignore the seawater sides of all components. Thus the actual volume of the condenser is 22.7×10^4 ft^3 with 17.3×10^4 ft^3 being water filled and neutrally buoyant. This is also true for the evaporator and pumps. Further, this table does not include the volume occupied by the cold water discharge flow beyond the pumps, and free volume areas within the plant superstructure not associated with any particular component. Adding these neutral water volumes to the Table 6 displacement total yields 10.852×10^5 ft^3. Thus 2.7×10^5 ft^3 remains as ballast control area. The total buoyancy of this ballast area is 17.3×10^6 lbs which is more than adequate for plant buoyancy maintenance and for surfacing and freeboard requirements.

PARASITE POWER

An accurate determination of parasitic power consumption is essential to the sizing and feasibility study of an SSPP. It was excessive power consumption by auxiliary systems, particularly cold water pumping, that contributed in part to the failure of the Claude Cuban and Abidjan thermal power extraction plants.

The percentage of pumping power required to gross output is extremely dependent on temperature. A temperature difference drop of a few degrees will greatly increase the parasitic power. This makes quantification of these losses most important. Some of the other losses are relatively fixed, such as hotel load for a manned plant. Thus, while a plant can be designed to provide a certain gross output, regulation of the variable parasitic power losses is essential to determine the economic feasibility of such a plant. Figure 8 shows the justifiable investment per kilowatt (net output) in a fuel-free plant for various prices of fuel for fuel fired plants.

This section will identify and typify, if not quantify, these parasitic losses for the prototype SSPP design.

While water pumping is the most significant loss, other losses to be

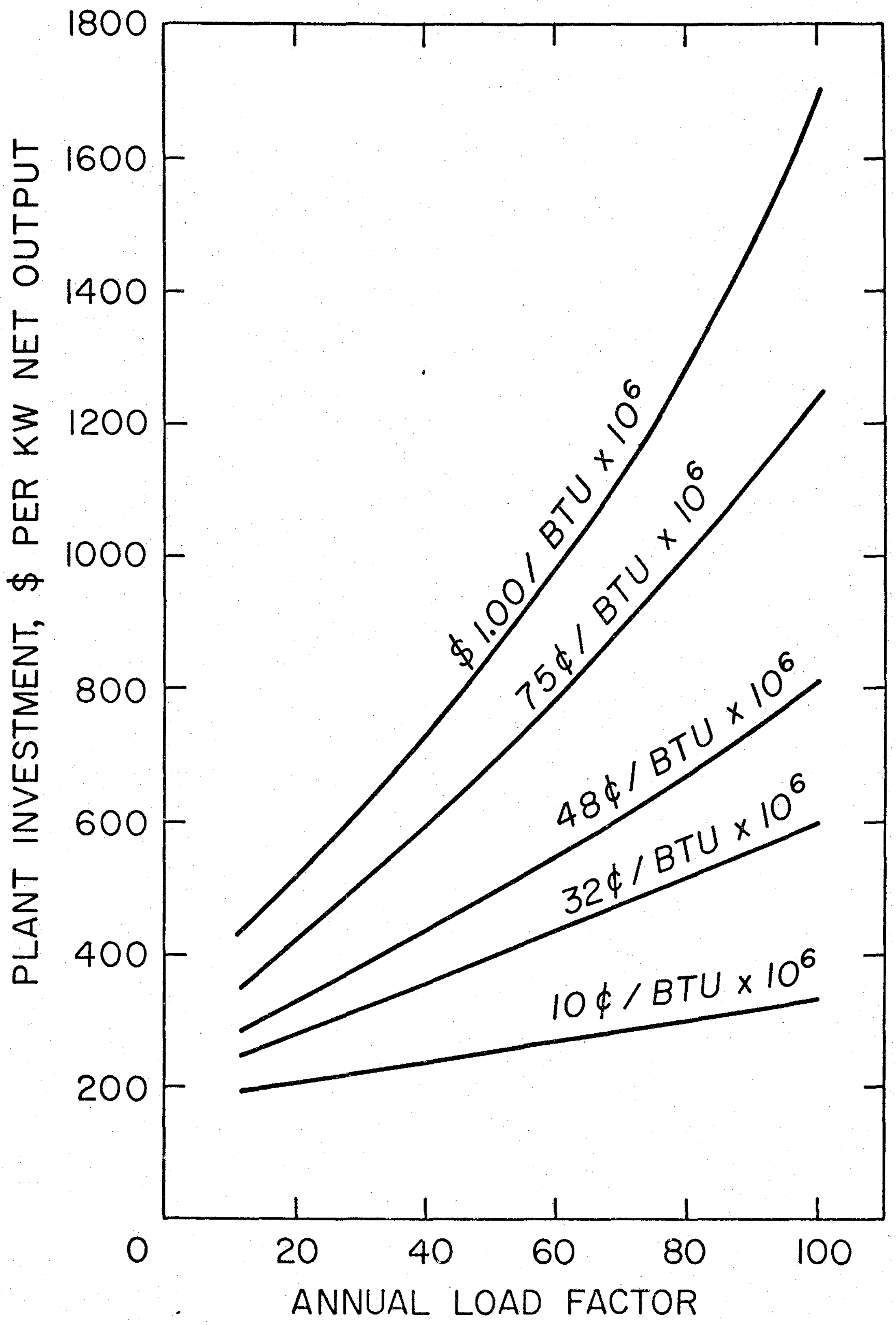

FIGURE 8. Investment in Fuel Free Plant Equated to Energy and Investment Costs in a 300 MW (Trojan = 1100 MW) Fuel Fired Plant for Various Load Factors and Energy Costs

TABLE 5. MISCELLANEOUS ITEM WEIGHT TABULATION

Item	Displacement (ft^3)	Weight (lbs)
Position Control	*	*
Thrusters	10^3	10^5
Lines	*	*
Communications Buoy	*	*
Diver Support	10^3	10^4

* Does not affect net buoyancy.

TABLE 6. WEIGHT SUMMARY

Component	Weight (lbs x 10^6)	Displacement (ft^3 x 10^4)	Buoyancy (lbs)	Net Buoyancy (lbs)
Condenser	4.43	10.40		
Condenser Armor	0.13	0.08		
Evaporator	4.07	9.52		
Evaporator Armor	0.14	0.09		
Personnel Hull	0.42	2.92		
Machinery Hull	1.21	2.92		
Evaporator Pumps	0.14	1.25		
Condenser Pumps	0.14	1.77		
External Storage	0.03	0.50		
Transformers	Neg	Neg		
Mooring Cables	Neg	Neg		
Structure & Exostructure	0.56	0.37		
Plant Subtotals	11.27	29.82	19.08 x 10^6	+7.81 x 10^6
Coldwater Pipe	8.64	1.77	1.13 x 10^6	-7.51 x 10^6
Totals	19.91	31.59	20.22 x 10^6	+0.31 x 10^6

1. Control surfaces and thrusters not included.
2. For buoyancy computation, ρ = 64 lbs/ft^3.

identified and investigated are those associated with bringing warm surface water to depth, circulating the working fluid, disposing of plant discharge, and cooling of the various pieces of machinery.

Other, minor loads that at least deserve order of magnitude consideration are those associated with life support system, hotel load, control of machinery, mooring, positioning, and communication.

Water Pumping Power: The preliminary concept locates the plant in a major current, thereby eliminating the need for pumping warm water to the boiler side heat exchanger, and discharge from both heat exchangers. However, the current decreases with increasing depth and the equivalent head from density differential necessitates the expenditure of pumping power to bring the cool benthic water to the condensers. While several pumping schemes are feasible, a variable pitch propellor type pump is the plan design pump as it is a low head, high volume, low maintenance pump. Multiple fixed pitch pumps would be a suitable alternate. These pumps will be located at the downstream end of the cold water flow loop. Cavitation was not seen as a problem with this atypical arrangement, since the relatively great depth of the plant provides a net positive suction head of approximately six atmospheres, far in excess of the total pressure head developed in such a pumping operation. For purposes of providing design temperatures to the heat exchanger, the cold water pipe was assumed to extend 730 ft below the plant to an absolute depth of 1,000 ft. It has been discussed that the pipe would hang nearly vertically, and that it would possess a well-rounded entrance, a trash rack to keep out debris, and possess one 90% bend, with no valving or control other than the ability to regulate pump speed and pitch.

The pumping load was further broken down into constituent parts to yield a general expression for pumping work:

$$W = Qg(\rho\Delta H + \Delta\rho H) \qquad (9)$$

where

W = Work (watts)
Q = Flow (L^3/T)
ρ = Density (M/L^3)
H = Headloss (L)

DENSITY DIFFERENCE LOSSES

Perhaps the least understood and most significant underwater transport loss is that encountered in overcoming density differences. In most engineering applications these losses are small (though the analagous case for air is the crux of draft and chimney calculations) and are neglected. However, at low velocities and heads, they can be significant. Not suprisingly, they have been neglected in some other studies of SSPP cold water pumping studies, thus casting dispersion on the credibility of those results.

The concept is probably most easily visualized if the pumping system is thought to act across two fluids of different density in a vertically stratified situation rather than over a density gradient (see Figure 9). This can be validly assumed since work is a line integral, dependent only on the end states, and independent of path. Thus, no matter what shape the pipe may take, or whatever density gradient might exist, the density difference pumping head remains constant for constant conditions of state at the inlet and outlet.

For the prototype SSPP design the conditions at the inlet and outlet are:

	Depth	Temp.	Salinity
Outlet	275 ft	73.0°F	36.45%
Inlet	1000 ft	47.0°F	34.98%

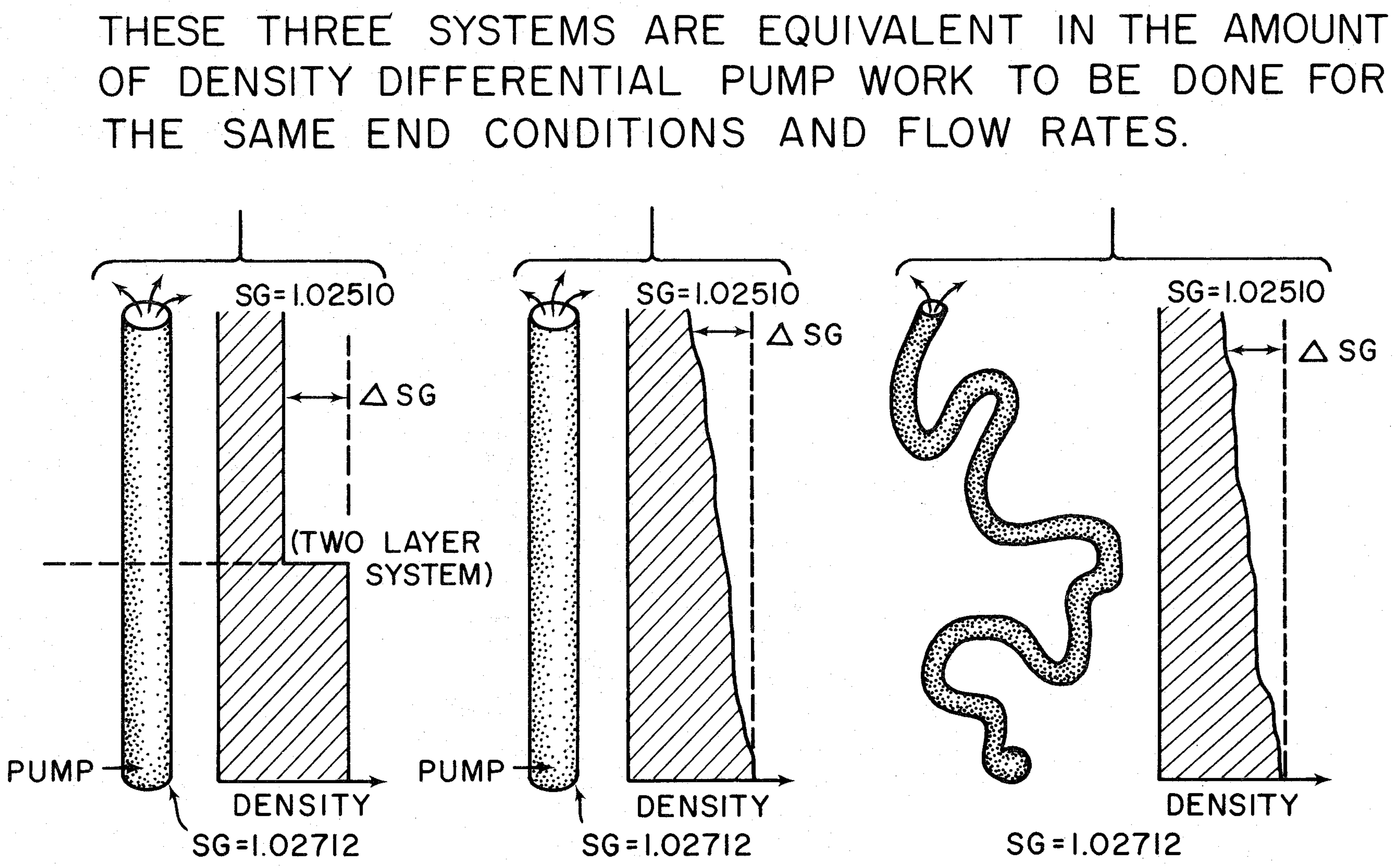

FIGURE 9. Concept-Denisty Differential Pumping Work

Using a nomograph published by the Navy Hydrographic Office (Figure 10), these values can be related to specific gravity by determining σ_t = (S.G.-1) x 1,000, and ultimately to density. The corresponding specific gravities at inlet and outlet are 1.02712 and 1.02510 respectively for a density difference of 0.000411 slug/ft^3.

As seen from the pump work equation, this density differential and vertical distance can be related to gravity head at constant density. This equivalent gravity head is 1.523 ft of seawater (S.G.= 1.0256).

Using this in the work equation, a specific loss of 142.4 watts/ft^3/sec, or 1.07 mw, results for a 7500 cfs cold water flow rate.

VELOCITY RELATED LOSSES IN PIPE

Head loss in pumping the cold water exclusive of gravity effects and heat exchanger losses can be expressed as follows.

$$H = \Sigma h = \underbrace{\frac{V^2}{2g}}_{\text{Velocity Head}} + \underbrace{\Sigma k \frac{V^2}{2g}}_{\text{Minor Loss}} + \underbrace{\frac{fL}{D}\frac{V^2}{2g}}_{\text{Friction Loss}} \quad . \; . \; (10)$$

where

V = Fluid velocity
g = Acceleration of gravity
k = Minor loss coefficient for various deviations from regular conveyance
f = Friction factor
L = Pipe length and,
D = Pipe diameter

Thus, a portion of the pumping load can be expressed as a series of terms which are a function of flow velocity and geometry, and which are proportional to the square of velocity for a given geometry. By establishing the desired flow and pipe geometry, these losses can be described as a function of pipe diameter.

This was done in Figure 11 for a geometry described by

Pipe Length = 730 feet
K_e, loss coefficient for entrance = .03[F]
K_s, loss coefficient for screen = 1.10[E]
K_b, loss coefficient for 90° vaned elbow = 0.60[D]

Pipe roughness was assumed to be constant over varying diameter. At three feet per second and below, fouling with barnacles 1/4" high was assumed. At velocities of three and four feet per second, friction factors were computed for a roughness corresponding to asphalt covered rolled steel plate.

The units of the abcissa of Figure 10 are expressed in watts per cubic foot per second, thus facilitating the power loss computation for any flow rate. The ordinate is pipe diameter in feet. If a flow rate and velocity is assumed, the diameter must be calculated by:

$$D = \sqrt{\frac{4Q}{n\pi V}} \quad . . . . \quad (11)$$

Similarly, for a given flow and diameter, velocity can be determined by:

$$V = \frac{4Q}{n\pi D^2} \quad . . . . \quad (12)$$

where n = number of equal diameter pipes used.

Several interesting trends can be surmised from Figure 11. It can be seen from the two plots for three feet per second velocity that surface roughness has a minor effect on pumping losses, particularly at the larger diameters. This is because wall friction is a very small contributor to total losses. By this same token, additional pipe length would increase losses only very slightly. For the range of velocities and diameters that are considered, in no case did velocity dependent losses approach the density differential losses by more than 60%.

Bearing in mind that optimizing a sub-system independently of the whole

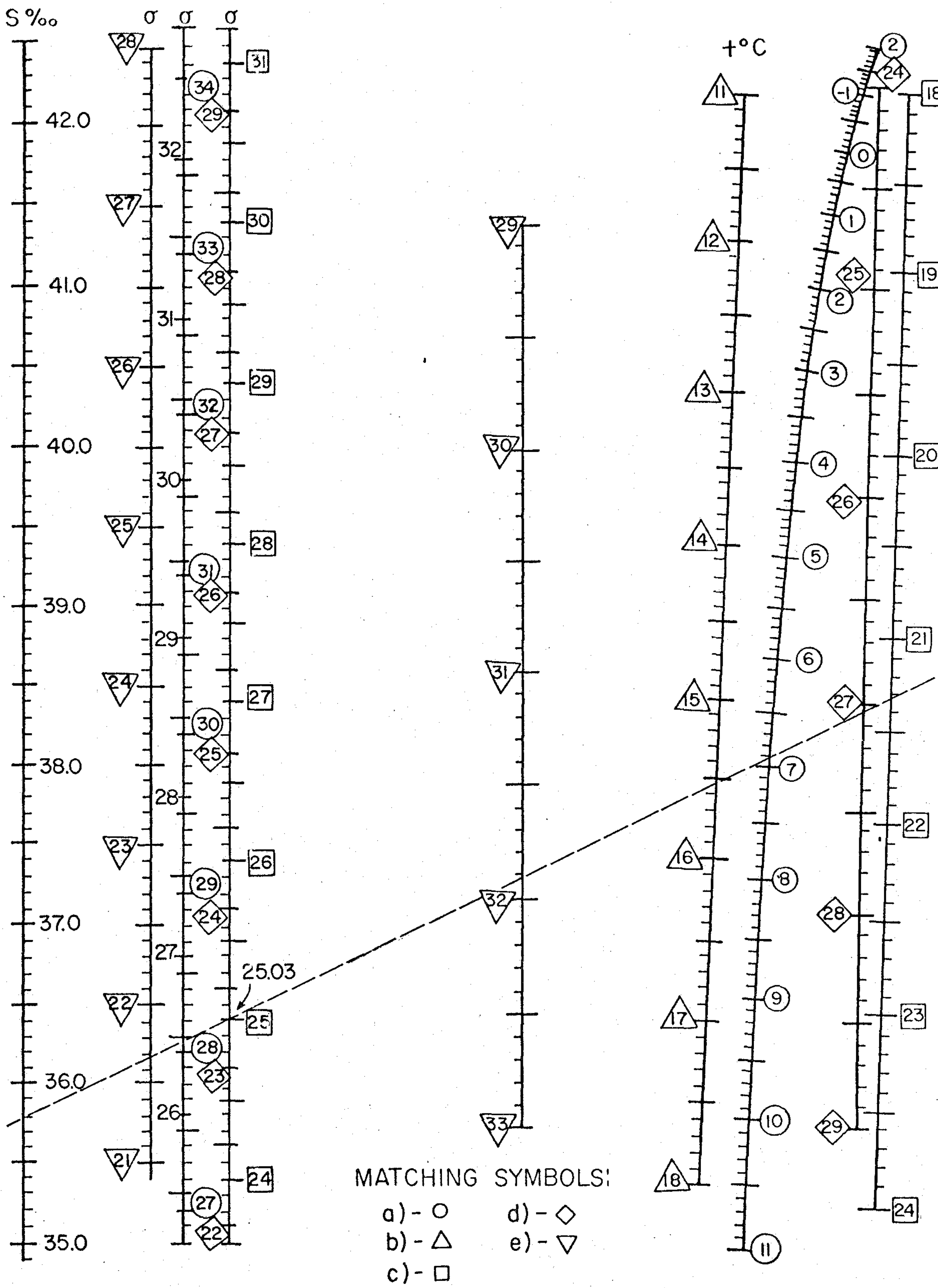

FIGURE 10. Nomograph for the Determination of Density from Salinity and Temperature of Sea Water (U.S. Navy Hydrographic Office, H.O. 007)

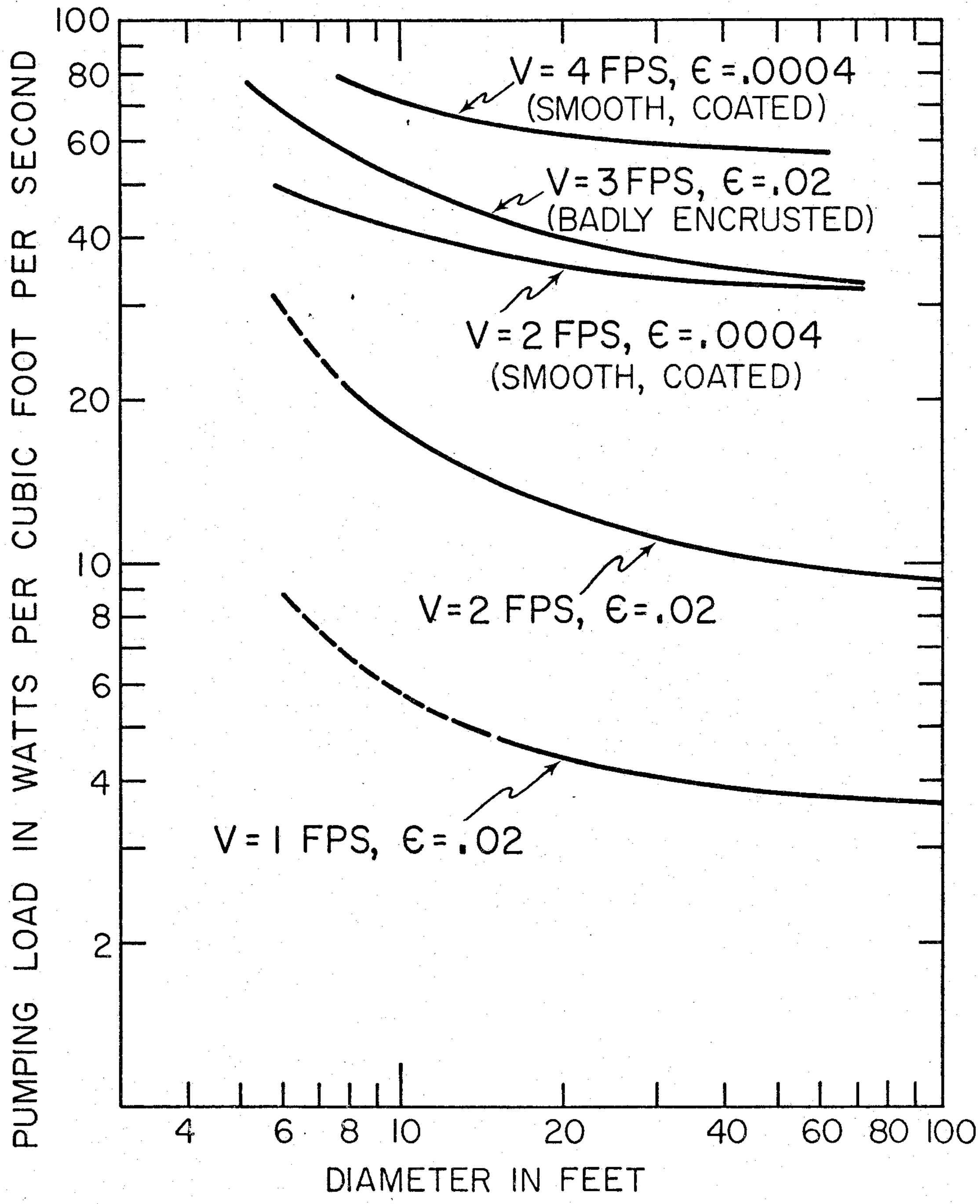

FIGURE 11. Pumping Power vs Pipe Diameter for Various Velocities & Friction Factors

can detract from the overall efficiency, an optimal diameter can be determined based on Penstock Design Economics (Sarkaria, 1958). This was computed for a flow rate of 7500 cfs to be 102' and a corresponding velocity of .92 feet per second, yielding a loss of 3.3 watts/cfs or 25 kw total velocity-related loss. The same flow divided over three pipes results in a diameter of 63.7', a velocity of .79 fps, and a loss of 1.5 watts/cfs.

Using a Bureau of Reclamation (1973) formula, the optimal velocity for a Penstock operating under the same conditions would be 1.22 fps with a corresponding diameter for a single pipe of 88.3 feet and a specific loss of 5.5 watts/cfs.

Using five fps and a 50' diameter, the friction losses are roughly 100 watts/cfs, or .75 mw for 7500 cfs.

Efficiencies in pumps of the size we are dealing with approach 100%. However, if one wanted to take a more conservative position, the above power ratings could be divided by .9 thus yielding a more cautious estimate.

HEAT EXCHANGER PUMPING LOSSES

In the design of the plant, it was determined that the ambient water temperature at the main plant was adequate to run the ammonia boiler, thus eliminating the need for warm water piping. Further assuming selection of a site to provide a steady current to move warm and discharged water to and from the heat exchangers, discharge piping is eliminated. All that remains is the loss occuring from moving the warm water across the heat exchangers at the desired flow rate. This same loss must be added to the cold water pumping loss.

Since the heat exchangers were determined to be the most critical element of the design from a functional standpoing, a constant loss of 13% of gross power was assumed in the design of both boiler and condenser side of the heat exchanger, this value having been determined from optimization of condenser sizing. Thus this same value will be used for the calculation of total parasitic power.

WORKING FLUID PUMPING LOAD

Working fluid pumping load is an intrinsic quantity determined by the thermodynamic cycle chosen for the plant. Only losses and pump inefficiencies will alter this figure upward from the minimum value determined by the cycle. In our prototype design, ammonia was chosen as the working fluid between temperature limits of 55°F and 65°F. The heat exchangers, deemed the most critical element of our design, were designed about these and other existing conditions.

The pumping load was computed in the report by Dan Ladd on cycle and working fluid considerations, and can be represented by the work in going from state three to state four on a temperature entropy diagram. This portion of the pumping power can also be determined by using the formula (Hawkins, 1967):

$$\frac{P}{\dot{M}} = V\Delta p \quad (13)$$

where

P = Power

V = Specific volume

Δp = Pressure at state four minus pressure at state three

$\dot{M}$ = Mass flow rate of working fluid

For the temperature limits of 65° and 55° respectively

$P4 = 117.8$ psi

$P3 = 98.1$

Therefore,

$$\Delta p = 19.7 \text{ psi}$$

and, $\bar{V} = 38.5$ ft /lb

Thus,

$$\frac{\omega}{\dot{M}} = 140.3 \text{ Btu/lb of working fluid.}$$

For a working fluid flow rate of 2000 lb/sec, this portion of the pumping work is .2806 mwatts.

Since the working fluid flow path is simple and large diameter conduits do not represent a significant heat loss problem because of the low temperature differential, it is assumed that friction losses in this system are negligible. A more conservative approach would be to assume a reasonable conduit diameters and calculate friction head at:

$$(5 \text{ to } 10) \times \frac{V^2}{2g} .$$

A pump efficiency of .90 in this flow rate and pressure range would be appropriate.

PRIME MOVER

Another area which falls into the category of parasitic power is that of prime mover efficiency. Any prime mover efficiency not only represents a loss of potential work from the thermodynamic cycle, but these losses usually manifest themselves as waste heat, for which pumping or blower power must at times be expended to remove, as in the case of turbine and generator bearings, and generator windings.

For purposes of this study, a standard 40 mw generator was selected (Mark, 1967). The efficiency of this unit is 98.7. Since a heat sink is readily available (the ocean), coolant pumping losses were considered to be small (5% of the heat load). Thus the parasitic load of the generator is given by:

$$PL = \frac{1.05*(1-\eta)*P}{\eta} \quad . . . . (14)$$

or, .55 mwatts for a 40 mw unit.

The turbine selection criteria used by Anderson (June, 1973) suggests an efficiency of .85 is possible for ammonia. Approaching this in the same manner as equation 14, the parasitic load of the turbine is $(1.05)(1-\eta)P/\eta$ or 7.41 mw for a 40 mw gross output.

HOTEL POWER LOAD

Providing power for life support and comfort of a manned crew represents a parasitic load on the plant output. These demands, along with an investigation of the need for air conditioning for crew and serviceable machinery areas are summarized below.

Relying heavily on data available for nuclear and research submarine life support and hotel loads, the breakdown shown in Table 7 was achieved. Total parasitic load for these functions is about .15 mw.

An investigation of the need for air conditioning/heating showed that because of the range of water temperatures nearby being at, or slightly below room temperature, the need for internal temperature control is minimized. The analysis concludes that the crew area with a three inch thick steel and 1/4" thick fibreglass will need heating of approximately 3300 Btu/hr, but that the machinery area will need substantially more cooling (434,260 Btu/hr) for the same insulation. By eliminating the machinery area insulation, this value can be reduced to a net loss. It is then practical to assume that an intermediate amount of insulation would give a slight net heat surplus from the machinery area, which could be used to heat the crew quarters. More detail on this budget would be needed for complete

TABLE 7. PARASITIC POWER ASSOCIATED WITH MANNED PLANT

Item	Load (kw)
Life Support	
0^2 Generator	2
CO_2 Scrubber	1
CO & H Burner	1
Gas Analyzer	
Compressor Pumps	10
Ventilation	8
Sub total	22 kw
Hotel	
Distillation, Personal Water	60
Hot Water Heating	16
Cooking & Food Preparation	4
Lighting	3
Communications Receivers	5
Refrigeration	10
Laundry & Dishwashing	20
Waste Water Pumping	1
Sub total	119 kw
Heating/Cooling	
Heat Pump	2
Total	143 kw

system design. A value of heat input pumping has been included in the hotel/ life support power load summary.

An actual need for air-conditioning in the crew area may occur while the plant is at or near the surface in warmer water. While this would not affect parasitic power inasmuch as the plant is not operable under these conditions, provision should be made for the installation of air-conditioning to be powered from shore or support ship. Further, if computer or automation equipment is included, the temperature and humidity must be more closely controlled. It is apparent that, in any event, some dehumidification scheme should be included.

MISCELLANEOUS PARASITIC POWER

The remaining items which were identified as contributing to parasitic power, but which were considered out of scope of the study by virtue of their apparent relative insignificance or of their dependence on arbitrary out-of-scope assumptions, are enumerated in Table 8. Also included are estimates of their order of magnitude and comments on the areas of investigation needed to refine the estimate, with the exception of transmission transformer losses, which depend heavily on the socio-economic-political decision. As to the form of electrical power produced and the distribution system, most of these losses can be proven minor and easily quantified once the subsystems are designed. The other losses are assumed to total to less than one percent of the gross output of the plant.

A summary of parasitic power requirements is shown in Table 9. It will be noted that approximately 45% of gross plant power is required for internal plant operation and only 55% is available for export power.

CONCLUSIONS

1. The 25mw plant considered in this paper is viewed purely as a prototype size plant. Final operating plant design would either be significantly larger, or be based on clusters of plant modules based on a 25mw to 50mw output for each module. This study has established that the physical size of a 25mw module is not excessive but is well within reasonable limits for the magnitude of physical facility required.

2. The selected plant design appears ideally suited for incorporation into the module concept. Further, the design pronotes on station stability cnd minimization of plant volume.

3. While the cold water pipe is negatively buoyant, the main plant structure possesses sufficient free space for conversion to ballast tanks to allow adequate buoyance for on station maintenance and for surfacing.

4. For a 25mw net output, approximately 45mw gross output will be required. 45% of this gross will be needed for plant internal operation, with 55% available as export power. While a direct comparison is not entirely appropriate as this system requires no purchased energy input, a very favorable comparison can be made between these figures and either fossile or nuclear power plants having plant efficiencies of approximately 33% and 40% respectively.

TABLE 8. MISCELLANEOUS PARASITIC POWER CONTRIBUTIONS

Component	Power Consumption (watts)	Comments
Communications		
Radio Transmitter/Telephone	10^2 - 10^4	Intermittent
Warning Lights/Sirens	10^3 - 10^4	Intermittent
Position Control		
Winches	0 - 10^5	Intermittent
Thrusters	0 - 10^6	See Mooring Section, Intermittent
Ballast Pumps	10^4 - 10^5	Intermittent
Control Surfaces	10^2 - 10^4	Intermittent
Plant Control		
Switch Gear	10^2 - 10^4	
Valves	10^3 - 10^5	
Computer	10^4 - 10^5	
Transformer Losses	10^2 - 10^6	Dictated by Desired Power Output Form

TABLE 9. PARASITIC POWER REQUIREMENTS SUMMARY

Component	Power Requirement (kw)
Cold Water Pumping (density)	984
For 6900 cfs @ 4fps (friction)	690
Evaporator and Condenser Pumping (26% of gross output)[1]	12,400
Working Fluid Pumping (for 2,000 lbs/sec)	281
Total Pumping Power Required	14,355
Pump Inefficiencies $(1-.9/.9)(1.01)P_p$	1,608
Generator Inefficiencies (1.3% of gross output)	685
Turbine Inefficiencies (10% of gross output)	4,500
Hotel Load	1,213
Total Parasitic Power Required	21,291

[1] For 25mw net output.

REFERENCES

Allis-Chalmers Co. Turbines and Auxilaries. (Advertising Brochure). Allis-Chalmers Co. 1950.

Ambs, L.L. and J. Marshall. "Ocean Thermal Difference Power Plant Turbine Design", Technical Report: NSF/RANIV/SE/GJ-3497/T12/73/17. University of Massachusetts, 1973.

Anderson, J.H. "Turbine Design: Sea Solar Plants", Technical Report. University of Massachusetts, June, 1973.

Charlier, R.H. "Harnessing the Energies of the Ocean", Marine Technology Society Journal. 3(2): 13-32, May, 1969.

Fraser, T.M. Confinement and Free Space Requirement. NASA, 1966.

Hawkins, G.A., and J.B. Jones Jr., Engineering Thermodynamics. New York: Wiley and Sons, 1967.

Hodgson, C.W. "Modern Thermal Station Design", Construction Engineering. September, 1963.

King, E.F. and H.W. King. Hardbook of Hydraulics. New York: McGraw Hill Book Co. 1963.

Ley, W. Engineer's Dreams. New York: Wiley and Sons, 1954.

Mark, L.S., ed. Mechnical Engineers Handbook. New York: McGraw Hill Book Co. 1967.

Sarkaria, G.S. "Economical Diameter For Penstocks", Water Power. 10(9): 352-356, September, 1958.

Streeter, V.L. Fluid Mechanics. New York: McGraw Hill Book Co. 1966.

U.S. Bureau of Reclamation. Design of Small Dams. Washington D.C.: U.S. Government Printing Office, 1973.

SEA SOLAR POWER PLANT - MOORING AND ANCHORING SYSTEM DESIGN

by

Jim Wright, Blaine Hafen, & Byungho Choi

ABSTRACT

General state-of-the-art design criterion and methods of approach in selecting a mooring and anchoring system for the Sea Solar Power Plant are discussed. Static and dynamic systems are compared and evaluated. For this evaluation a complete description of load and force source and type from the ocean environment and the plants dynamic response to these forces is required. Wide scatter still exists in such fundamental aspects as the values of C_D and C_I as used in the Morrison equation. Thus mooring system design is still an experimental science and a conservative design can only be obtained for any given conditions. Finally, potential anchoring systems are discussed and evaluated.

SEA SOLAR POWER PLANT - MOORING AND ANCHORING SYSTEM DESIGN

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INTRODUCTION

While a Sea Solar Power Plant cannot function without the successful operation of all subsystems, certainly no single subsystem is any more critical than the mooring and anchoring system. This criticality of the mooring and anchoring system arises from three sources, with each greatly magnifying the effects of the other two. These are:

1. The great diversity of source and types of loads to be absorbed by the system. Wind, wave, and current forces including steady state and transitory elements and the plants' dynamic response to these forces will form a very complex and diverse integrated system of loads for the mooring and anchoring system to counter.

2. The magnitude of these forces will be far greater than those encountered by existing mooring and anchoring systems. This magnitude is due to the large size of the plant, the depth of the water on site, and to the current velocity in the Florida current.

3. The current state-of-the-art of mooring and anchoring system design is such that definite solution of the conditions described in 1 and 2 for a specific system stress requirement is not possible. Such basic parameters as C_D and C_I for the Morrison equation have not yet been assigned definite values so that wide scatter still exists over their potential ranges.

This paper will define and analyze the expected load forces on the plant and discuss potential mooring and anchoring systems for use with the plant. Certainly more research is required before the mooring and anchoring system requirements can be fully defined.

LOAD SOURCES AND CONDITIONS

Currents: Surface currents will affect the drift and stabilization of the Sea Solar Power Plant (SSPP) when in a surfaced condition. Subsurface currents affect the stabilization of the structure as well as the anchorage. Currents result in the following load conditions: 1) a drag force which is proportional to the square of the velocity, and 2) an interference with mooring lines through the action of vertical and horizontal shear.

Two major types of currents must be considered, near-sea-floor turbidity currents and major oceanic currents. Near-sea-floor turbidity currents caused by submarine landslides are intermittent at best but have caused the fastest currents know with velocities of almost 50 knots on the sea floor. Their infrequent occurrence obviates their use in design criteria, although an on-site survey should include an analysis of this type of disturbance. There is little known about how far above the sea floor these currents act; however, the interaction between the cold water intake pipe and this current is considered improbable as over 600 ft. separates the cold water intake and the ocean floor.

Major ocean currents related to density distribution can reach speeds of greater than four knots (Ippen, 1966). A velocity profile for the Florida Current is shown in Figure 1. Open-ocean streams do not tend to flow along constant paths unless confined, as in the case of the Florida Current. Thus the complexity of load calculation is somewhat simplified due to the basically steady state nature of the Florida Current. Average net transport for this current is 26×10^6 m^3 per second.

Currents may here be defined as fluid velocities which may be treated analytically as steady-state phenomena. Therefore, the design profile used in the subsequent determination of forces was generated from averaging seasonal values at depths and approximating these by polygons. Figure 1 shows that the velocity of the current has dropped to approximately 2.10 feet per second at a depth of 200 feet. Therefore, in a submerged condition the SSPP will still have considerable current forces to overcome. The forces due to currents are mainly those caused by drag, and a drag coefficient and design current must be selected. It should also be noted that the vector sum of the steady current and wave particle velocity should be considered in a swift current such as the Florida Current.

Winds: Much of the information on wind measurements has been based on data gathered over land areas. Thus data for wind description over water is generally lacking and the design wind speed must be selected on the basis of a statistical evaluation of area records. Methods such as the Beard Method (Beard, 1952) and Gumbel's Standard Skewed Distribution (Gumbel, 1958) can be applied to the problem of obtaining design wind speed. There are several methods available to proceed further and correlate wind speed and fetch with wave heights. Bretschneider (1959) was one of the first to apply a correlation method for design purposes. His dimensional analysis method provided the basis for some of the more complicated and restrictive methods also used today. Other methods are discussed in Ippen (1966). Falvey (1974) recently has devised a correlation method and compared his results with standard wind casting curves. His method predicts values of significant wave heights accurately by replacing complex mathematical formulations with empirical formulas.

Wind loads however will not be considered as critical for the SSPP as the plant is submerged approximately 98% of the time. It is assumed that when surfaced, the structure will exhibit 20

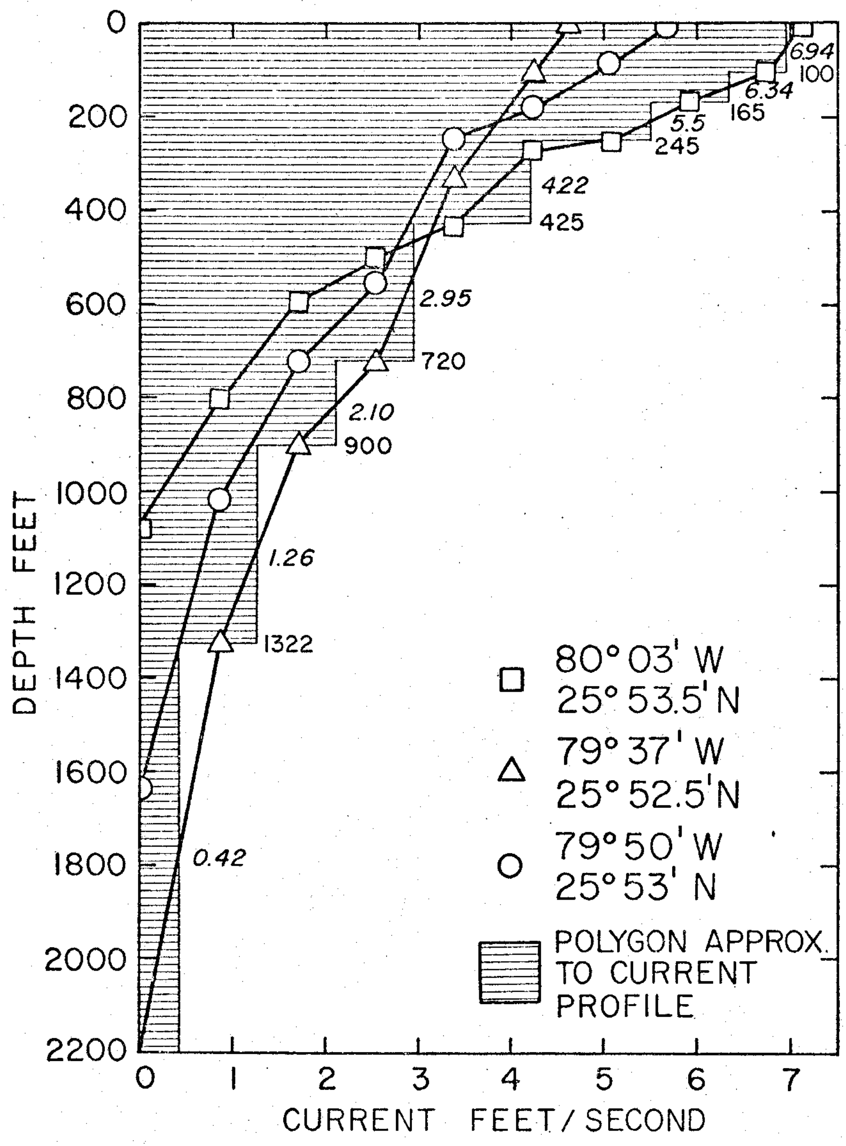


FIGURE 1. Florida Current Velocity Profile

feet of freeboard and experience a maximum wind of 20 knots since the surfacing capability allows optimization of wind and wave conditions. Myers (1969) indicates procedures used in calculating wind forces, where the most difficult determinations as far as engineering judgment is concerned, revolves about the values of C_L and C_D, coefficients of lift and drag. Occasionally dynamic effects must be considered. Turbulent eddies or gusts of wind having frequencies near the natural frequency of the structure can excite resonance.

Wave: In simplified form, the forces on a structure in the marine environment can be classified as forces due to waves and forces due to currents. In determining wave forces on a structure, the following steps are noted: 1) selection of a design wave height and period; 2) selection of an appropriate wave theory to compute water particle velocities and acceleration; and 3) selection of drag and inertia coefficients from experimentation or experience which may be applied to a suitable wave-force equation.

The determination of a design wave is greatly dependent on a statistical approach as is the determination of a useful wave theory. Beginning with Gerstner's first wave theory in 1802, many approaches have since been proposed (Bascom, 1964). As a first approximation to the wave forces on the SSPP, the Airy wave theory found directly from the linearized hydrodynamic equations will be used. Here the amplitude is considered half the wave height. Hence,

$$\phi = \frac{ag}{\sigma}\frac{\cosh k(h+z)}{\cosh kh}\cos(kx-\sigma t) \qquad (1)$$

$$u = -\frac{\partial\phi}{\partial x} = \frac{agk}{\sigma}\frac{\cosh k(h+z)}{\cosh kh}\sin(kx-\sigma t) \qquad (2)$$

$$\frac{\partial u}{\partial t} = -agk\frac{\cosh k(h+z)}{\cosh kh}\cos(kx-\sigma t) \qquad (3)$$

The range of usefulness of all wave theories cannot be precisely determined. One suggested criterion would be to use the breaking index curve (h/T^2 vs. H_B/T^2) as by Weigly (1971), or by Myers (1969) who includes a comparison of sinusoidal, cnoidal, and solitary wave theories. The decision to use first-order or higher order will depend on the problem to be studied and the magnitude of the design wave height. Higher order theories generally predict higher velocities with very little change in acceleration (Ippen, 1966). Thus a higher order theory should be chosen where the drag force is important. Prior to describing the forces on a solid body it is assumed that the structural dimension is small with respect to the wave length so that its presence is not detectable in terms of wave and current motion a short distance away. A large structure will significantly modify the wave and current fields relatively far from the structure.

There exist two types of forces in the direction of the wave. One is a drag force which is due to the relative velocities between the structure and the fluid. It requires real fluid effects (friction, vorticity, shear, turbulence) and produces a net shear and pressure force on the structure. The other is an inertia force which is due to relative accelerations between the structure and the fluid. It can exist in an ideal as well as a real fluid and potential flow theory is satisfactory for its computation. It produces a net pressure force on a body and is associated with unsteady flows relative to the body or fluid. The drag force may be divided further into skin drag (due to flow parallel to the surface) given by:

$$F_{D_s} = C_f \rho A \frac{U|U|}{2} \qquad (4)$$

where

C_f = skin friction coefficient (derived from boundary layer theory)

A = area parallel to flow,
and, U = approach velocity.
and into form drag (due to flow perpendicular to surface) given by:

$$F_D = C_D A\rho \frac{U|U|}{2} \quad \text{.................(5)}$$

where

C_D = drag coefficient which is a function of the shape of the body,
and, A = frontal area.

Since C_f is approximately 0.01, the contribution of skin drag to total drag is small and will be neglected in SSPP calculations.

For a fixed structure in an accelerating fluid, the inertial force equals the displaced fluid mass times the acceleration of the fluid mass plus the added mass of adjacent fluid particles times that acceleration (Ippen, 1966). Thus the total forces on a body under wave action is:

$$F_T = F_D + F_I = C_D A \frac{U|U|}{2} + C_I \rho \frac{\partial U}{\partial t} \quad \text{..(6)}$$

where

C_D is determined experimentally, and
C_I is determined from potential flow theory.

Note that in Weigle (1971) and Yamamoto (1963) there is considerable scatter as to values of C_I,C_D in an oscillating wave situation. It has been assumed for the SSPP, that $C_I = 2.0$ and $C_D = 1.0$. However, as has been indicated, these values can only be assigned true value through experiment. Yamamoto (1973) indicates where F_D and F_I are important to consider by comparing h/L to H/D.

Another force which may occur on a structure is due to asymmetries in the velocity field about the stagnation stream line. This results in forces at right angles to the flow field and is called lift. Asymmetries exist in velocity fields because of vortex shedding, circulation (Kutta-Jonkowski) or the effect of a solid boundary or air-sea interface. As an example lift becomes important for a cylinder where $e/D < 1.5$. In other words, within certain limits, as the distance between an object and a solid boundary decreases the coefficient of lift, C_L, in

$$F_L = \rho C_L D \frac{U|U|}{2} \quad \text{................(7)}$$

increases (Yamamoto, 1973). This may also be compared with work by Ippen (1966).

In summary, the following parameters will be used in the determination of forces on the structure:

Surface wind	= 20 knots
Freeboard	= 20 feet
Design wave	H = 15 ft
(Airy Theory)	T = 10 sec
	L = 512 ft

As calculated later in this paper, this gives a total maximum drag of 1.2×10^6 pounds due to wind, wave, and current forces.

RESULTANT FORCES DUE TO LOADS

Currents: Ocean currents vary not only with time but with position; therefore, current is a dynamic loading condition, and will interact with the structure dynamically. In the time domain the current will have a spectral density, and if the dominating frequency does not correspond to the first model frequency of the structure, the loading due to current can be considered as static. The maximum loading condition should be used.

In a uniform velocity the following forces apply:

$$F_T = F_f + F_D \quad \text{..................(8)}$$

F_f = friction force

F_D = drag force

$F_f = C_f V^2/2 \; BL$

$F_D = C_D \rho A V^2/2$

where

C_D = drag coefficient
ρ = density--(varies with depth, temp., and salinity)
B = transverse width
L = length of surface parallel to flow
A = cross-sectional area
C_f = friction coefficient

For laminar flow, $C_f = \frac{1.328}{R}$;

for transition flow ($3 \times 10^5 \leq R \leq 5 \times 10^5$),

$$C_f = \frac{0.455}{(\log R)^{2.58}} - \frac{1.700}{R} \text{ ; and}$$

for turbulent flow ($R > 5 \times 10^5$),

$$C_f = \frac{0.455}{(\log R)^{2.58}}$$

As the Reynolds number increases vortex shedding occurs in the lee side of the structure. The frequency of this shedding can be characterized by the Strouhal number, i.e., $S = fD/V$ where f = frequency of shedding. As indicated in Figure 2, the Strouhal number for $10^2 < R < 6 \times 10^5$ is $\simeq .2$. Due to this shedding, a lift force is created which is perpendicular to the direction of flow. This force alternates from side to side. Therefore, the loading is a dynamic loading and the frequency of shedding needs to be compared with the model frequency of the structure. The lift force can be calculated as follows:

$F_L = C_L \rho A V^2/2$ ...(OSU Engr. Exp. Sta., 1971)...(9)

The lift force for $R \approx 6 \times 10^3$ is approximately 25% of the longitudinal force and when the wake is fully turbulent, the maximum lift force is approximately 40% of the longitudinal force.

Thus, if the loading due to the current can be considered static, the forces due to current can be calculated based on equations (8) and (9).

Wind: Loading due to wind can be considered as three separate components: wind-induced water current, static load due to mean wind velocity and dynamic loading due to turbulence in the air. A further breakdown of these loads includes (Plate and Nath, 1969):

1. Wind-induced water current:

$F = C_D \rho_W \, AV^2/2$ ....(10)

where

V is the surface velocity proportional to shear velocity, and

$V = \sqrt{\frac{\tau_{WS}}{\rho_A}}$ (τ_{WS} = water surface shear stress exerted by wind).

Not a lot is known about this force, however, an approximation of 3% of wind speed has been used.

2. Static load due to mean wind velocity:

$F = C_{DA} \rho_A \, (\overline{U}_A^2)_{avg} \, DL/2$ ....(11)

C_{DA} = drag coefficient
$\overline{U}_A$ = local temporal mean velocity (avg. with respect to height)
L = some length parameter of structure
A = refers to air motion
ρ_A = mass density of air (should contain sea spray)

Note: Review of Bishop and Hassen (1964) indicates that the oscillation of a cylinder in fluid flow increases the drag force as much as fourfold. More research is needed in this area to determine the actual effect. Since it can be assumed that any motion of the structure will influence the forces, the dynamic response of the structure must be examined for all loading conditions (i.e., comparison of model frequency to loading frequency) before a static condition can be assumed.

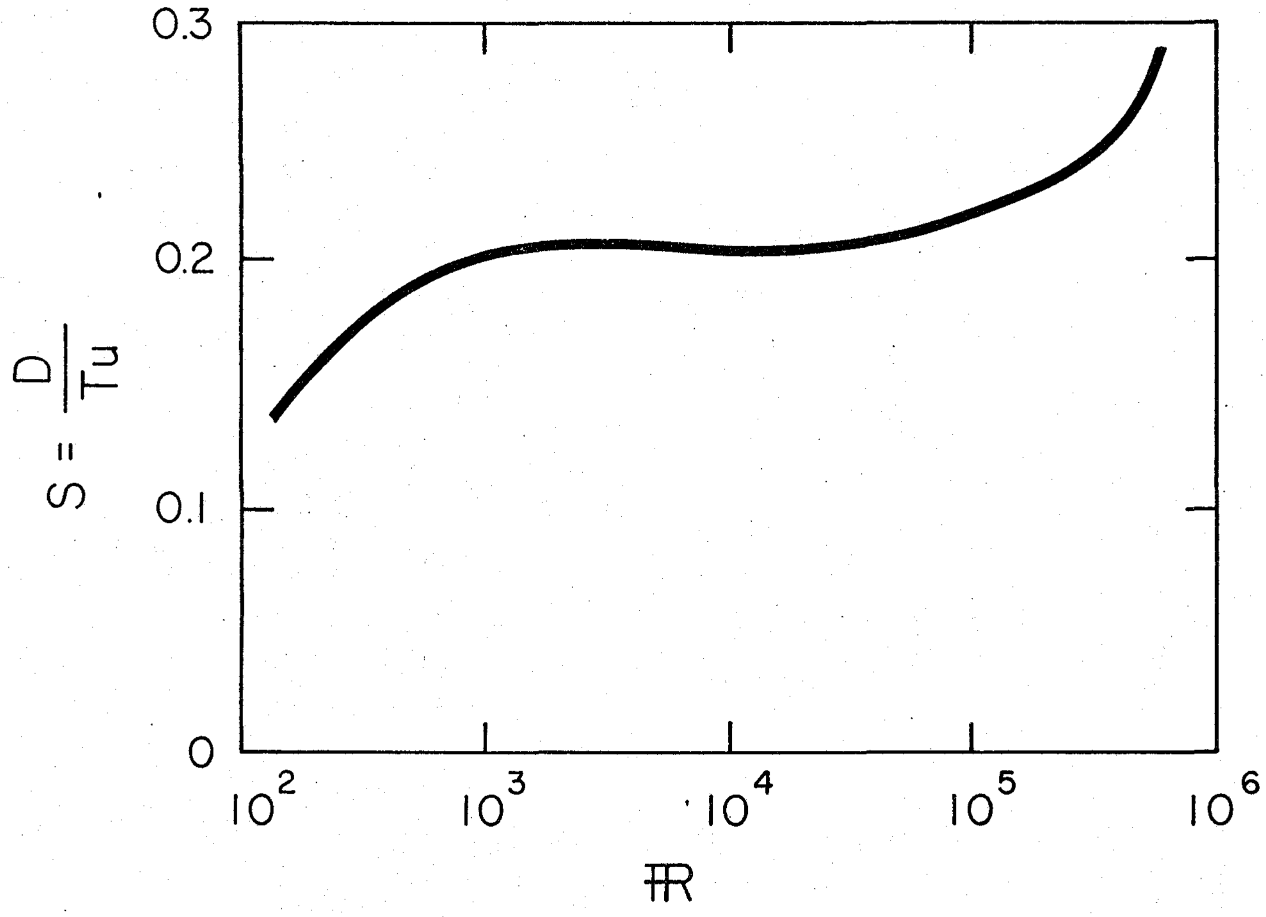


FIGURE 2. Variation of Strouhal with Reynolds Numbers, Circular Cylinders (Ippen, 1966)

3. Dynamic loading: Dynamic loading is of much greater consequence because it may add to dynamic loading of waves. Forces due to air turbulence are generally small; however, air turbulence spectra have their peaks at larger frequencies than wave spectra and thus closer to frequency of structure; therefore, their effect may be appreciable.

Waves: As indicated for currents, the dynamic response due to waves must first be investigated with reference to a sea-surface spectral density vs. frequency graph to determine if static condition can be approximated.

For a force on cylindrical pile due to a passing wave (5) the Morrison equation can be applied which consists of drag and inertia components as follows (Nath and Harleman, 1970):

$$\phi(z,t) = C_D A\rho\, u|u|_{rel} + B\rho\dot{u} + C_I B\rho\dot{u}_{rel} \quad \ldots\ldots..(12)$$

where

- B = volume of cylinder
- ρ = mean density of water
- C_I = inertia coefficient
- u_{rel} = relative horizontal bed of fluid particle with respect to structure
- $\dot{u}$ = horizontal acceleration of fluid particle
- $\dot{u}_{rel}$ = relative horizontal acceleration of fluid particle with respect to structure

u, and $\dot{u}$ can be calculated from the appropriate wave theory (see Figures 3 & 4). However, for design considerations the Airy wave theory can be used as a first approximation. Since both u and $\dot{u}$ vary with depth, the force due to waves in an integral form (Ippen, 1966) are:

$$F_T = \int_0^{h+\eta} \frac{C_D\rho}{2} Da^2\sigma^2 \frac{\cosh^2 ks}{\sinh^2 kh} \cos\sigma t\,|\cos\sigma t|\,ds - \int_0^{h+\eta} C_m \frac{\rho\eta D^2}{4} a\sigma^2 \frac{\cosh ks}{\sinh kh} \sin\sigma t\, ds \quad \ldots.(13)$$

where

C_m = coefficient of added mass = $\dot{C}$ + 1 (2.0 for cylinder, potential flow)

and u and $\dot{u}$ are obtained from Airy wave theory. For deep water conditions, $h/L = 1/2$, F_T can be approximated by:

$$F_T = \gamma \frac{C_D D}{4} a^2 \cos\sigma t\,|\cos\sigma t| - \gamma C_m \frac{\pi D^2}{4} a \sin\sigma t \quad \ldots\ldots(14)$$

and total moment by:

$$M_T = \gamma \frac{C_D D}{4} a^2 h \cos\sigma t\,|\cos\sigma t| \left[\frac{kh-\frac{1}{2}}{kh}\right] - \gamma \frac{C_m \pi D^2}{4} ah \sin\sigma t \left[\frac{kh-1}{kh}\right] \quad \ldots\ldots..(15)$$

In general:

$$F_T = \frac{C_D\rho D}{2}\int_0^{h+\eta} u(s)|u(s)|\,ds + C_m\rho \frac{\pi D^2}{4}\int_0^{h+\eta} \dot{u}(s)ds \quad \ldots..(16)$$

Tangential Forces: As has been indicated the SSPP has an operating depth of 200 feet to take maximum advantage of the existing thermal gradient while being beyond the effective depth of surface wave forces. Thus the maximum uplift forces will occur when the plant is surfaced for crew rotation or minor repairs. Assuming again a 20 foot freeboard surfaced condition with deep water wave conditions of H = 15 feet, L = 512 feet, T = 10 sec.:

$$\text{Tangential drag force } (f_t) = C_f AW^2/2 \quad ..(17)$$

where

W = vertical velocity $= \frac{agk}{\sigma}\frac{\sinh k(h+Z)}{\cosh kh}\cos(kx-\sigma t)$

A = total surface area parallel to W $\simeq 8.8 \times 10^5\ ft^2$ for SSPP

$C_f \simeq 0.01$

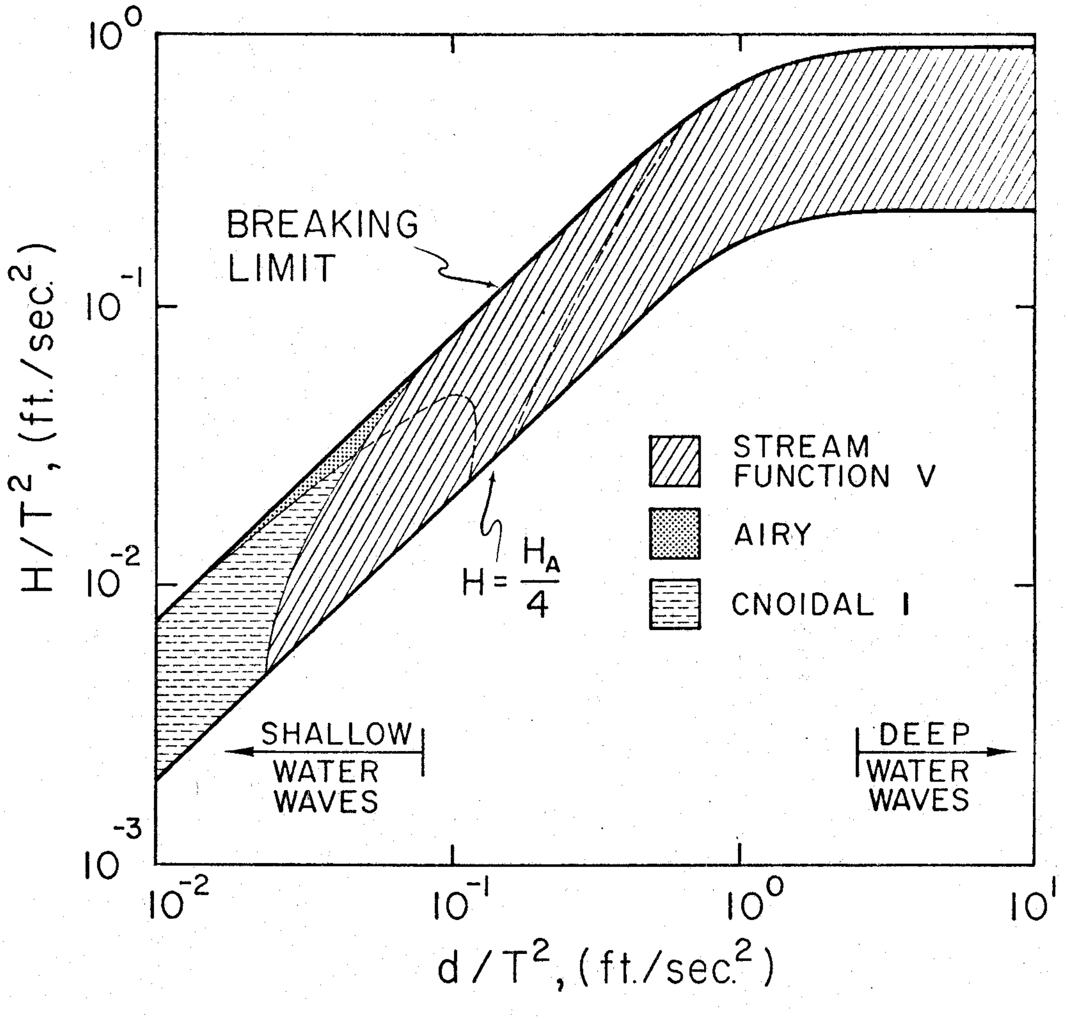


FIGURE 3. Periodic Wave Theories Providing Best Fit to Dynamic Free Surface Boundary Condition (Analytical and Stream Function V Theories)

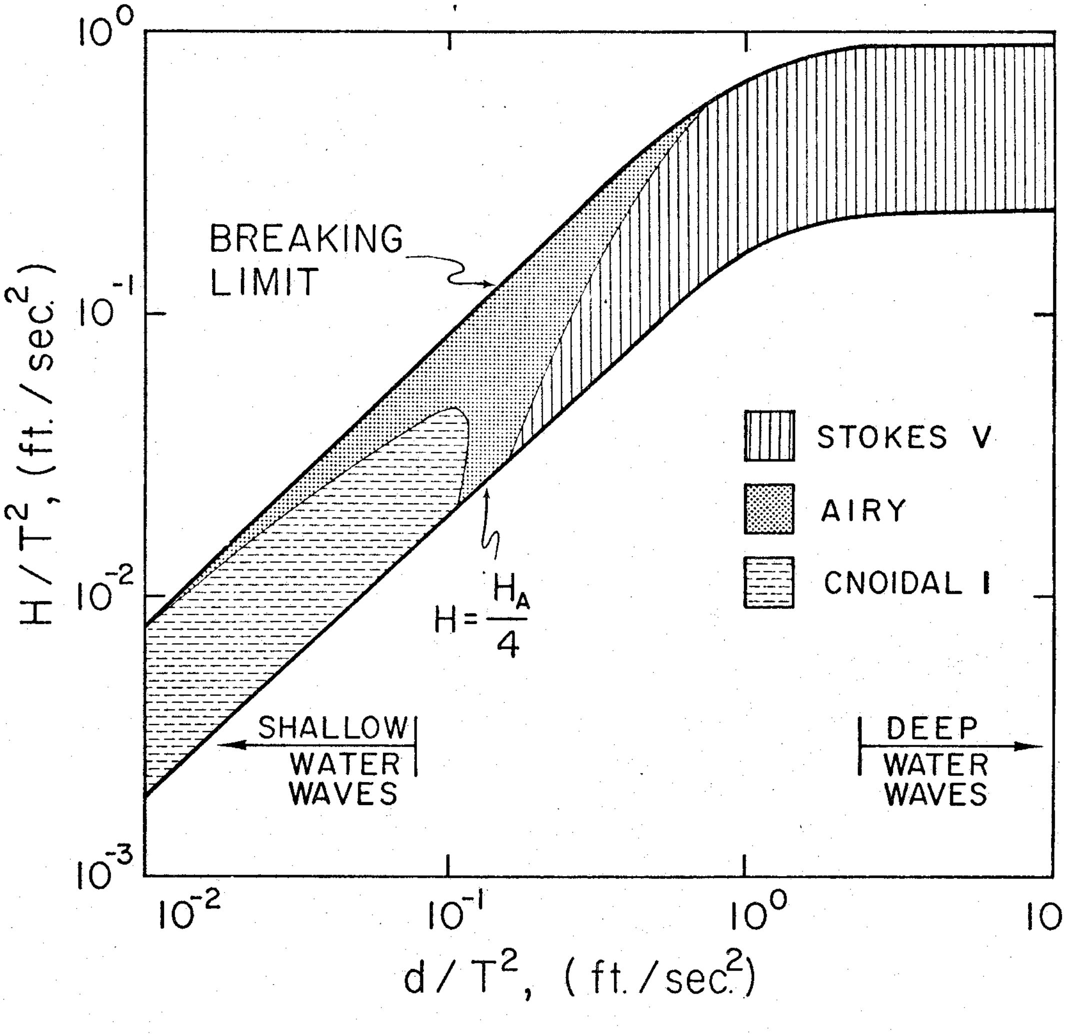


FIGURE 4. Periodic Wave Theories Providing Best Fit to Dynamic Free Surface Boundary Condition (Analytical Theories Only)

Therefore maximum $W = \frac{\pi H}{T} e^{KZ}$

and total drag = $\int_0^Z f_t \, dZ \approx 2000$ lbs

As mentioned before, this force is fairly negligible compared to form drag forces.

Other Forces:

1. Wave induced vibrations: In all the wave force treatments of the SSPP prior to this a static loading condition is assumed. In actuality the structure will respond to the action of the waves and currents in such a way that the equations of motion of the structure must be satisfied. Experimental tests, beyond the scope of this report, are necessary to determine the dynamic responses of the structure. Ippen (1966) has some experimental data relating to rigid moored structures. In order to accurately determine the response of the structure, both the energy spectrum (i.e. the energy per unit frequency) and the frequency of occurrence should be determined at the proposed location of the structure. These are discussed by Kinsman (1965). The structure should be designed such that its natural frequency is well above the surface wave spectrum in order to avoid a resonance condition between the wave and structural frequencies. Most semi-submerged, pile supported, offshore structures have been designed with a minimum vibrational mode period of one second or less. Static assumptions have been justified in these cases since sea spectra do not normally contain significant energy content at this low frequency.

A secondary cause of structural vibration is the vortex shedding in the wave induced flow past a structure. Even if the natural frequencies of the structure are above the wave spectrum, these frequencies may still be in the range of those due to vortex shedding. As a wave passes a structure, lift forces which are quasi-periodic in nature will occur due to the shedding of vortices. The nature of the shedding is not well known in oscillatory flows (Ippen, 1966). The vortices shed in the wake of a circular pile for instance occur at a frequency which is a function of both the free stream velocity and the diameter of the pile. The frequency data are usually presented through the use of the Strouhal number.

$$S = fD/U \quad \text{.................} (18)$$

The relationship between the Strouhal number and the Reynolds number has been shown in Figure 2. The lift forces on the piling as an example is defined as:

$$dF_L = C_L \rho D \, U^2/s \, dS \quad \text{............} (19)$$

where C_L is the lift coefficient. If the vortex shedding period for a flexible cylinder, $1/f$, is near the natural period of the cylinder, lift forces could occur which were over four times the drag force based on a uniform steady flow at the same velocity (Ippen, 1966). Thus it seems evident that wave induced vibrations must be considered in the final design of the SSPP. It has been shown that in some experiments that the ratio of maximum lift to maximum longitudinal force is about 40% (Weigle, 1971).

2. Wave-making drag: Another force encountered in designing the SSPP position stabilization system, especially in a swift current such as the Florida Current, is wave-making drag. This is the drag resulting from the creation of surface waves by the structure in a surfaced position. The presence of the structure causes both local disturbances called bow waves and divergent waves and also transverse or free waves. All of these waves remain with the structure depending on the free stream velocities. Theoretical and experimental data on wave resistance are usually presented in the form of a drag coefficient (C_R) as a function of the Fronde number (U/gh) which in turn is based on body length. The values of C_R must be experimentally determined for the SSPP and the drag

will add to the total force in a surfaced condition.

3. Coupled heaving and pitching: In a surfaced condition the SSPP will experience rectilinear motions of heave, surge, and sway and the angular motions of pitch, roll, and yaw. Due to the 730 ft. cold water intake pipe and the position stabilization system (dynamic or static) some of these six degrees of freedom can be neglected. Thus only the motions of heave, and pitch will be discussed. McCormick (1973) offers a solution to these problems through the strip theory method. Other examples are given by Myers (1969) with a moored structure, the cable tension at the point of attachment to the structure is an additional constraint on the equations of motion. It is also indicated that a symmetric structure with a fairly constant cross section and a characteristic length equal to an integer multiple of the general wavelength of the design sea state will experience no net heaving force. The structure will experience only a pitching movement induced by the waves. Also the heaving and pitching motions of a submerged structure decrease exponentially with depth. However from several experimental results (McCormick, 1973) a semi-submerged structure such as the SSPP will experience little heaving or pitching motion for all but extremely long waves. Quinn (1972) relates that the heaving motion from the mean position of an unmoored 800,000 DWT tanker in a beam sea with 15 foot oscillatory waves is less than one-half the wave height in the vertical direction.

Coefficients: All the forces listed above have in some way depended on a drag or lift coefficient; therefore, it is appropriate to discuss the limitations of their use and the relative range of their values for actual conditions to be expected at sea.

The values of C_D, C_L, and C_m are determined experimentally, and for the steady state condition the value of C_D is as depicted in Figure 5 and Table 1, and a value of 2.0 for C_m has been calculated from potential flow; however, the fact that the motion at sea is unsteady and oscillatory has precluded the direct application of steady state drag coefficients. Investigation has its limitations due to the inter-relation of C_D and C_m (Myers, 1969). Most of the work to determine drag and mass coefficient in accelerating flows is listed in Table 2. A significant reason for the scatter previously shown in Figure 1 can be attributed to the relative roughness of the cylinder. At sea this condition will be accentuated by bio-fouling and corrosion; therefore, a correction to the drag and mass coefficient must be accounted for in their use. Figure 6 illustrates the net effect of a roughened cylinder; however, no concrete values for application have been established and a conservative estimate should be used.

LOAD CASE COMPARISON

Submerged Loading: It has been shown that in a submerged operating mode, the SSPP will experience loading primarily due to current forces and will respond dynamically and statically to these forces. The flow is assumed turbulent in the region of the cold water pipe and also in the region of the main plant. This is based on the overall first design approximation and the critical Reynolds numbers. As a first approximation to the dynamic response of the plant with regard to vortex shedding, the factor of 40% of the maximum longitudinal force was assumed (Weigle, 1971). Thus including minor wave forces and marine growth, a first estimate of the total force on the submerged structure is 6.0×10^5 pounds.

Note: Drag and inertia coefficients are not known for wire or synthetic rope; therefore, they must be estimated from shapes such as those shown in Figure 5. This applies as well to the longitudinal drag coefficient. (An analytical approach has been proposed by Nath, Smith and Yamamoto, 1974).

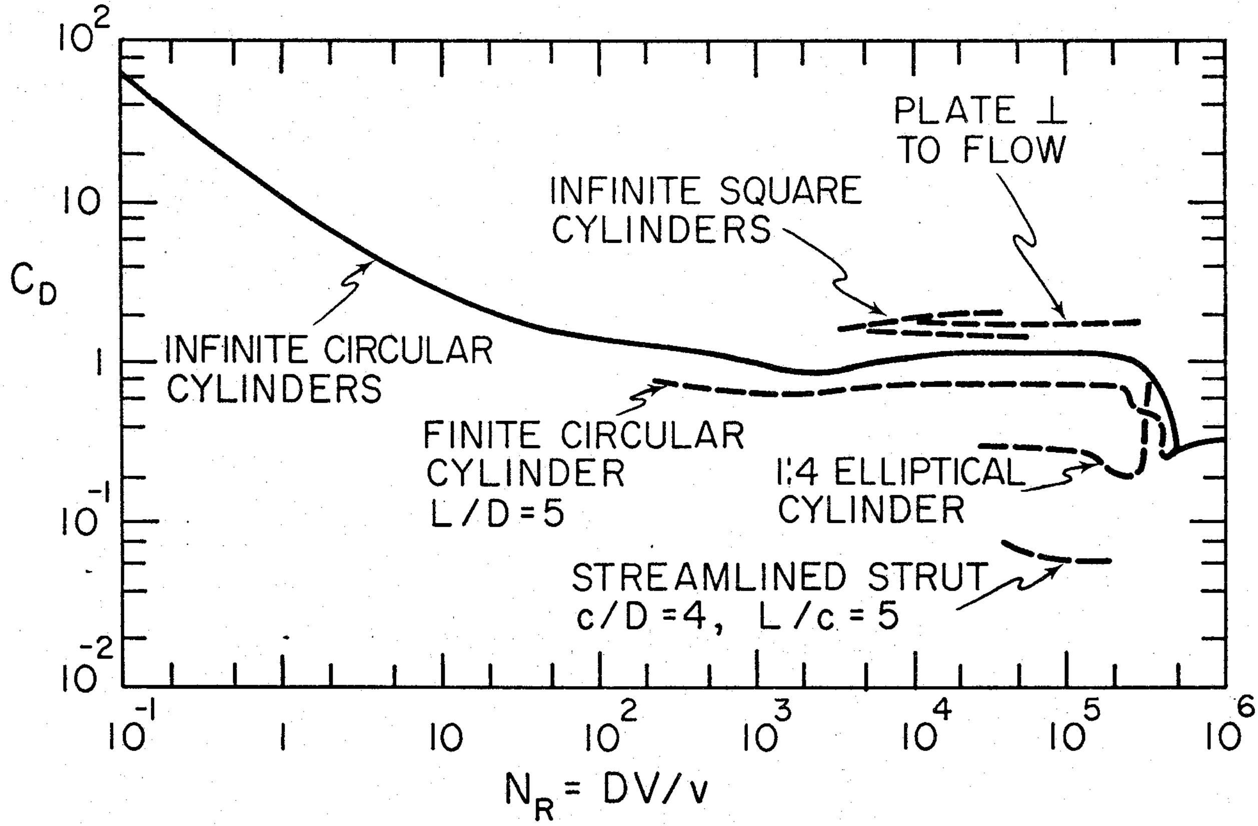


FIGURE 5. Drag Coefficient for Two-Dimensional Bodies

TABLE 1. APPROXIMATE VALUES OF DRAG COEFFICIENT AND DISK RATIO FOR VARIOUS BODY FORMS

Form of Body	L/D	R	C_D	Disk Ratio
Circular disk		$>10^3$	1.12	
Tandem disks (L = spacing)	0	$>10^3$	1.12	1.0
	1		0.93	0.83
	2		1.04	0.93
	3		1.54	1.37
Rectangular plate (L = length)	1	$>10^3$	1.16	
	5		1.20	
	20		1.50	
	∞		1.95	
Circular cylinder (axis parallel to flow)	0	$>10^3$	1.12	1.0
	1		0.91	0.81
	2		0.85	0.76
	4		0.87	0.78
	7		0.99	0.88
Circular cylinder (axis perpendicular to flow)	1	$>10^5$	0.63	
	5		0.74	
	20		0.90	
	∞		1.20	
	5	$>5\times10^5$	0.35	
	∞		0.33	
Streamlined foil (1:3 airplane strut)	∞	$>5\times10^4$	0.10	
Hemisphere: Hollow upstream		$>10^3$	1.33	1.19
Hollow downstream			0.34	0.30
Sphere		$>10^5$	0.47	0.42
		$>3\times10^5$	0.20	0.18
Ellipsoid (1:3, major axis parallel to flow)		$>2\times10^5$	0.06	0.054
Airship hull (model)		$>2\times10^5$	0.042	0.038

TABLE 2. DRAG AND INERTIAL COEFFICIENT VALUES FOR CIRCULAR CYLINDERS IN ACCELERATING FLOWS (FROM [6])

Authority and Date (1)	Nature of Experiments (2)	Cylinder Diameter (inches) (3)	Coefficient Value C_D (4)	Coefficient Value C_M (5)	Type of Flow (remarks) (6)
Crooke, 1955	Model	2, 1, 1/2	1.60	2.30	Oscillatory
Keulegan and Carpenter, 1956	Model	3, 2 1/2, 2	1.34	1.46	Oscillatory (av. of 29 tabulated values)
		1 1/2, 1 1/4	1.52	1.51	(av. of 57 tabulated values)
Keim, 1956	Model	1, 1/2	1.00	0.93	Accelerated, monoscillatory
Dean, 1956	Model	3	1.10	1.46	Accelerated, monoscillatory
Wiegel et al., 1956	Prototype	24	1.00	0.95	Ocean waves, west coast (based on their Fig. 15)
Reid, 1956	Prototype	8 5/8	0.53	1.47	Ocean waves, Gulf of Mexico
Bretschneider, 1957	Prototype	16	0.40	1.10	Ocean waves, Gulf of Mexico
Wilson, 1957	Prototype	30	1.00	1.45	Ocean waves, Gulf of Mexico

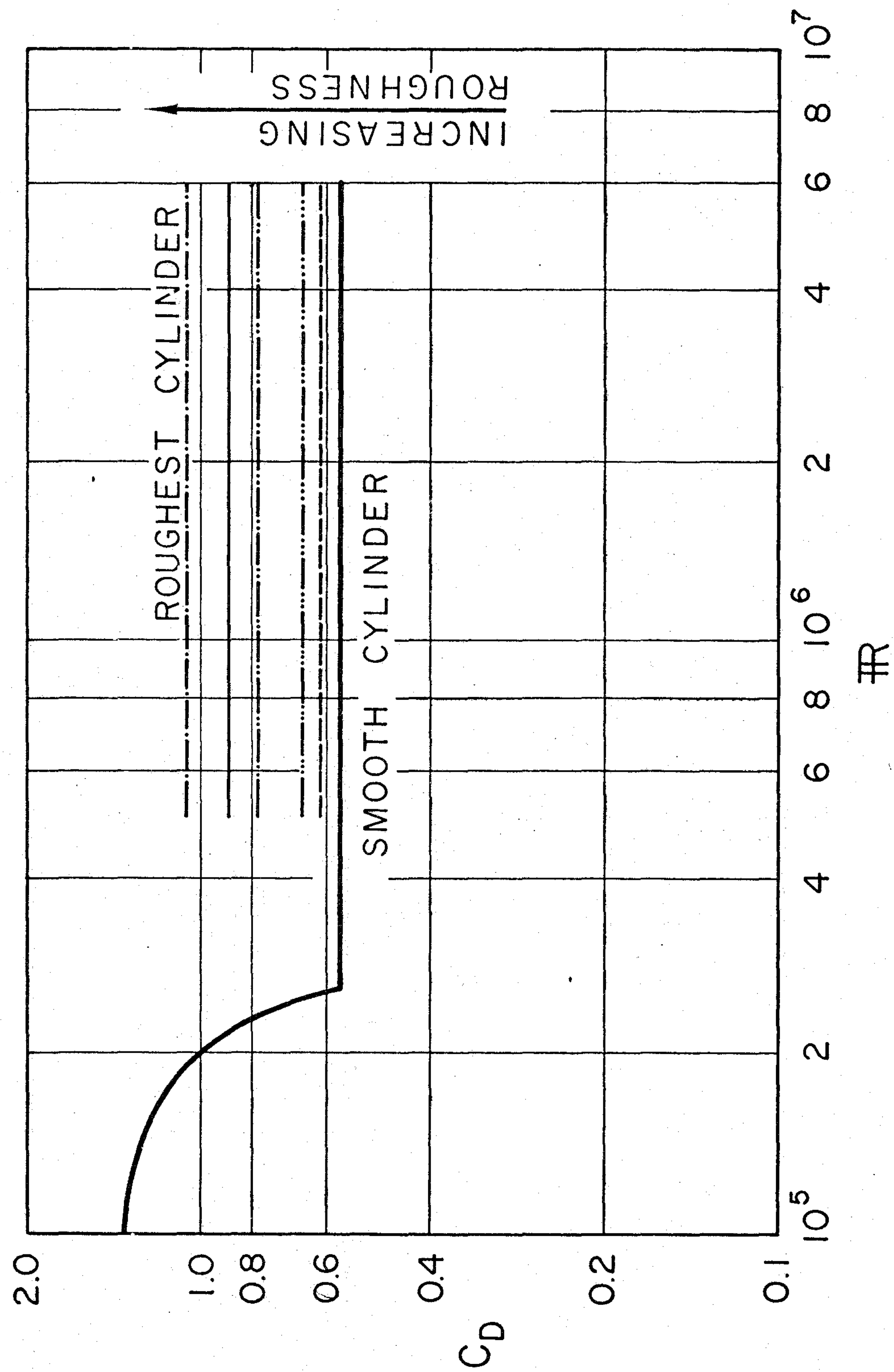


FIGURE 6. Effect of Roughness of Drag Coefficient in Supercritical Region

Surfaced Loading: The response of the structure in a surfaced condition is indeed the most critical in the determination of stability calculations. Aside from current, wind, and wave forces one must consider the forces due to structural vibrations caused by waves and shedding vortices and also forces due to wave-making drag and coupled heaving and pitching. It is noted that a small design wave and relatively calm sea state has been assumed in the surface loading condition. It is emphasized again that the sea state into which the plant will be surfaced can be chosen at an optimum time. As shown before, the total force due to winds, waves, and current including lift forces from flow field asymmetries is on the order of 1.2×10^6 pounds. The vibrations due to waves can increase the frequency of vortex shedding which in turn will increase lift forces. The added forces due to wave-making drag and heave will be determined by experimentation.

MOORING SYSTEM COMPARISON

For the purpose of a comparison analysis of the two types of moorings (dynamic and static), order of magnitude forces will be calculated based on the following assumptions previously discussed in this paper:

1. Dynamic response of the structure to waves, current, and wind is such that a static analysis can be made.

2. Current profile at site is as shown in Figure 1.

3. Plant physical configuration is as shown in Figure 7.

4. Surfaced condition will exhibit 20 feet of freeboard.

5. In surfaced condition plant will be subjected to:

   a. 20 knot wind ($\overline{U}_A$)
   b. 15 foot waves, 10 sec period, 512 ft long
   c. h = 2000'

6. Appropriate coefficients are:

   a. $C_D = 1.0$ cylinder
   b. $C_m = 2.0$ cylinder and rope

7. Airy wave theory may be used to predict u and $\dot{u}$ (at $z = -L/6$ for surfaced condition).

8. Plant will remain stationary in a wave (like spar buoy).

9. ρ = const = 1.98 slugs/ft^3

FORCES CALCULATION (SUBMERGED CONDITION)

Current for the turbulent condition:

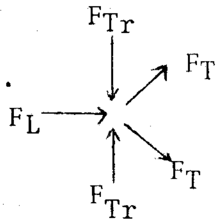
$$R_{pipe} \approx \frac{(2.5)(50)}{1.07 \times 10^{-5}} = 1.2 \times 10^7,$$

$$R_{plant} \approx \frac{(5.5)(90)}{1.07 \times 10^{-5}} = 4.6 \times 10^7,$$

$$C_{f_{pipe}} = \frac{0.455}{(\log R)^{2.58}} = .0029, \text{ and}$$

$$C_{f_{plant}} = \frac{0.455}{(\log R)^{2.58}} = .0024$$

For these turbulent conditions, Table 3 summarizes the submerged loading forces. From those calculations, therefore,



Total F_{long} = 498,341 lbs.

With F_{long} = 498,341 lbs,

$$F_{Trans} = (.4)(F_{long}) = 199,336 \text{ lbs}$$

$$F_T = (F_L{}^2 + F_{Tr}{}^2)^{\frac{1}{2}} = 536,620 \text{ lbs.}$$

Wave: Wave forces would be felt for a portion of the plant length since L/2 = 256'; however, their effect would be minor compared to the current forces. Therefore, in this analysis, a force of $\approx 6 \times 10^5$ lb will be assumed in the submerged condition (wave effect will influence the tension in the mooring line since the plant will not follow the wave).

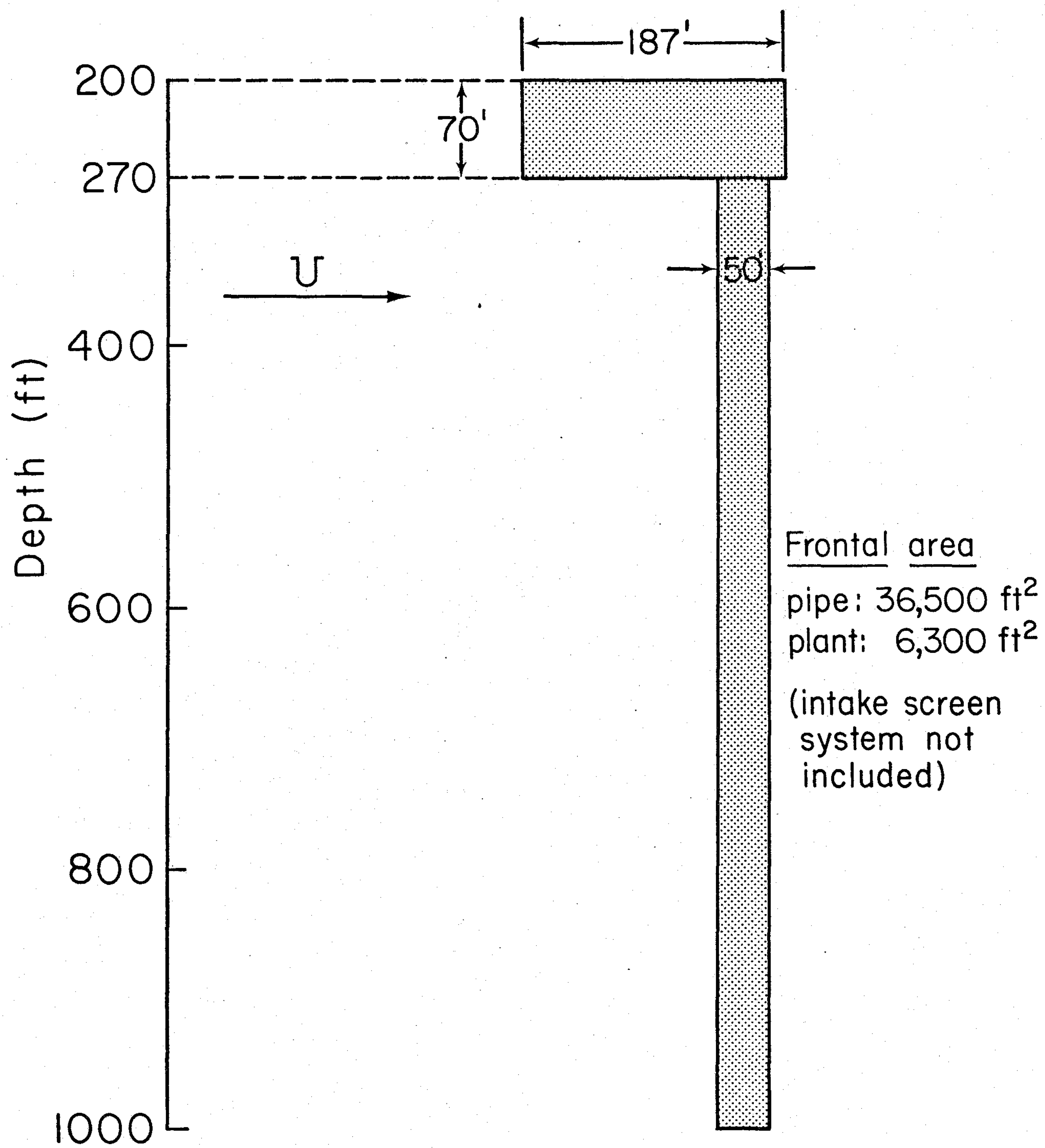


FIGURE 7. Elevation View of Plant

TABLE 3. PLANT SUBMERGED CURRENT LOADING

Section	Area	C_D	C_f	Velocity H/sec	F_f lb	F_D lb
Vertical pipe	5,000	1.0	0.0029	1.26	9.0	7,859
	9,000	1.0	0.0029	2.10	26.0	39,293
	14,750	1.0	0.0029	2.95	51.0	127,078
	7,000	1.0	0.0029	4.22	125.0	121,525
Plant	2,250	1.0	0.0024	4.22	212.0	39,400
	5,400	1.0	0.0024	5.50	947.0	161,716
				Totals	1,370.0	496,971

FORCES CALCULATION (SURFACED CONDITION)

Current: F_f is minor and will be ignored. Table 4 summarizes F_D for this condition.

TABLE 4. PLANT SURFACED CURRENT LOADING

Section	Area	Vel.	F_D
Plant	4,500	6.94	214,350
Pipe	2,500	6.94	119,205
	3,250	6.34	129,329
	4,000	5.50	119,790
	9,000	4.22	148,673
	14,750	2.95	127,078
	2,500	2.10	10,915
		Total	879,340

Therefore, $F_{long} = 879{,}340$ lbs

and, $F_{trans} = 351{,}736$ lbs

Wind: Maximum loading will be calculated for wind load acting in the same direction as current loading.

Static load:

$$F_{long} = C_{DA}\, P_A\, (\overline{U}_A)^2_{avg}\, DL/2 \quad \text{........(20)}$$

$$=[(1.0)(2.34 \times 10^{-3})(20 \times 1.69)^2] [(90)(185)/2]$$

$$= 22{,}350 \text{ lbs}$$

$$V = 33.8 \text{ ft/sec}$$

$$R = \frac{(33.8)(90)}{1.7 \times 10^{-4}} = 1.8 \times 10^7 \text{ (turbulent)}$$

$$F_{trans} = 8{,}940 \text{ lbs}$$

Current:
$V = (0.03)(33.8) = 1.01$ ft/sec - dissipates over 50', therefore 1.01 ft/sec is used for 25'.

$$F_{long} = C_D \frac{PAV^2}{2} = 4{,}545 \text{ lbs} \quad \text{..........(21)}$$

$$R = \frac{(1.01)(90)}{1.07 \times 10^{-5}} = 8.4 \times 10^6 \text{ (turbulent)}$$

$$F_{trans} = 1{,}818 \text{ lbs.}$$

Waves: Again, maximum loading will be calculated for wind load acting in the same direction as current loading.

$$u = \frac{agk}{\sigma} \frac{\cos h\ k(h+z)}{\cos h\ kh} \quad \text{..............(22)}$$

$$\dot{u} = -agk \frac{\cos h\ k(h+z)}{\cos h\ kh} \quad \text{............(23)}$$

where

$a = 7.5$
$k = 2\ /L = 0.0123$
$kh = 24.54$
$z = -60'$ for the plant, and
$z = -145'$ for the pipe; and,

$$F = C_D\, \rho A \frac{V|V|}{2} + C_m \rho B \dot{v} \quad \text{..........(24)}$$

Plant:

$$u = [(7.5)(32.2)(0.0123)\ /(2\pi/10)] [(\cos h\ 23.86)/(\cos h\ 24.54)]$$

$$= 2.395 \text{ ft/sec}$$

$$\dot{u} = [(7.5)(32.2)(0.0123)] [(\cos h\ 23.86)/(\cos h\ 24.54)]$$

$$= 1.505 \text{ ft/sec}^2$$

u and $\dot{u}$ are 90° out of phase with max "u" at the crest of the wave. Therefore forces are taken at that point for $\dot{u} = 0$.

$$F_{long} = \frac{(1.0)(1.98)(90)(90)(2.395)^2}{2}$$

$$= 45{,}997 \text{ lbs}$$

$$R = \frac{(2.4)(90)}{1.07 \times 10^{-5}} = 2.0 \times 10^7 \text{ turbulent}$$

$$F_{trans} = 18{,}399 \text{ lbs.}$$

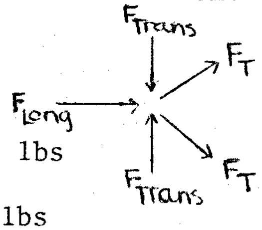
Pipe:

$$u = [(7.5)(32.2)(0.0123)/(2\pi/10)]$$
$$[(\cos h\ 22.82)/(\cos h\ 24.54)]$$
$$= .85 \text{ ft/sec}$$

$$F_{long} = \frac{(1.98)(50)(175)(.85)^2}{2} = 6{,}259 \text{ lbs}$$

$$R = \frac{(.85)(50)}{1.07 \times 10^{-5}} = 4.8 \times 10^6 \text{ turbulent}$$

$$F_{trans} = 2{,}504 \text{ lbs.}$$



Total: $F_{long} = 1{,}035{,}000$ lbs

$F_{trans} = 414{,}000$ lbs

$F_T = (F_L{}^2 + F_{TR}{}^2) = 1{,}114{,}729$

use 1.12×10^6 lbs as an order of magnitude

DYNAMIC POWER REQUIREMENTS

If dynamic mooring is used, then a thrust of approximately 1.2×10^6 lbs must be generated by the plant to stabilize the plant at the surface. As this thrust must come from the power plant, its computation is necessary.

Knowing $F = \rho Q(V_1 - 7)$
(at surface $V \simeq 7$ ft/sec)

$$\text{or,}\quad Q = \frac{F}{\rho(V-7)} \quad \text{.................}(25)$$

$$\text{and,}\quad HP = \frac{\gamma Q(V^2/2g)}{550} \quad \text{..............}(26)$$

required horsepower and required megawatts (Mw) may be calculated as summarized in Table 5.

TABLE 5. MW AND HP TABULATION FOR VARYING VELOCITY

V ft/sec	Q ft^3/sec	HP	Mw
8	606,060	70,085	52.3
10	202,020	36,502	27.2
12	121,212	31,538	23.5
15	75,758	30,799	23.0
17	60,606	31,647	23.6
16	67,340	31,149	23.2

Also,

$$Q = \rho AV = \rho \frac{\pi D^2}{4} V$$

$$\text{or,}\quad D^2 = \frac{4Q}{\rho \pi V} \quad \text{..................}(27)$$

Therefore, for V = 15 ft/sec,

$$D = \left[\frac{4Q}{\rho \pi V}\right]^{1/2} = 56 \text{ ft or its equivalent}$$

Therefore for dynamic positioning a minimum of 23.00 Mw must be expended to maintain plant on station. If a location which minimizes current is found for the plant then the magnitude of the power needed will be reduced since current is the predominant force. In this case it would be impractical to use dynamic positioning as 23 Mw of the 25 Mw plant output would be required for positioning. If, however, the same profile is used as in Figure 8 but with the currents normalized such that surface velocity is 1 ft/sec, then current forces would be as shown in Table 6.

From Table 6 then,

$$F_{long} = 18{,}186 \text{ lbs, and}$$

$$F_{trans} = 7{,}274 \text{ lbs.}$$

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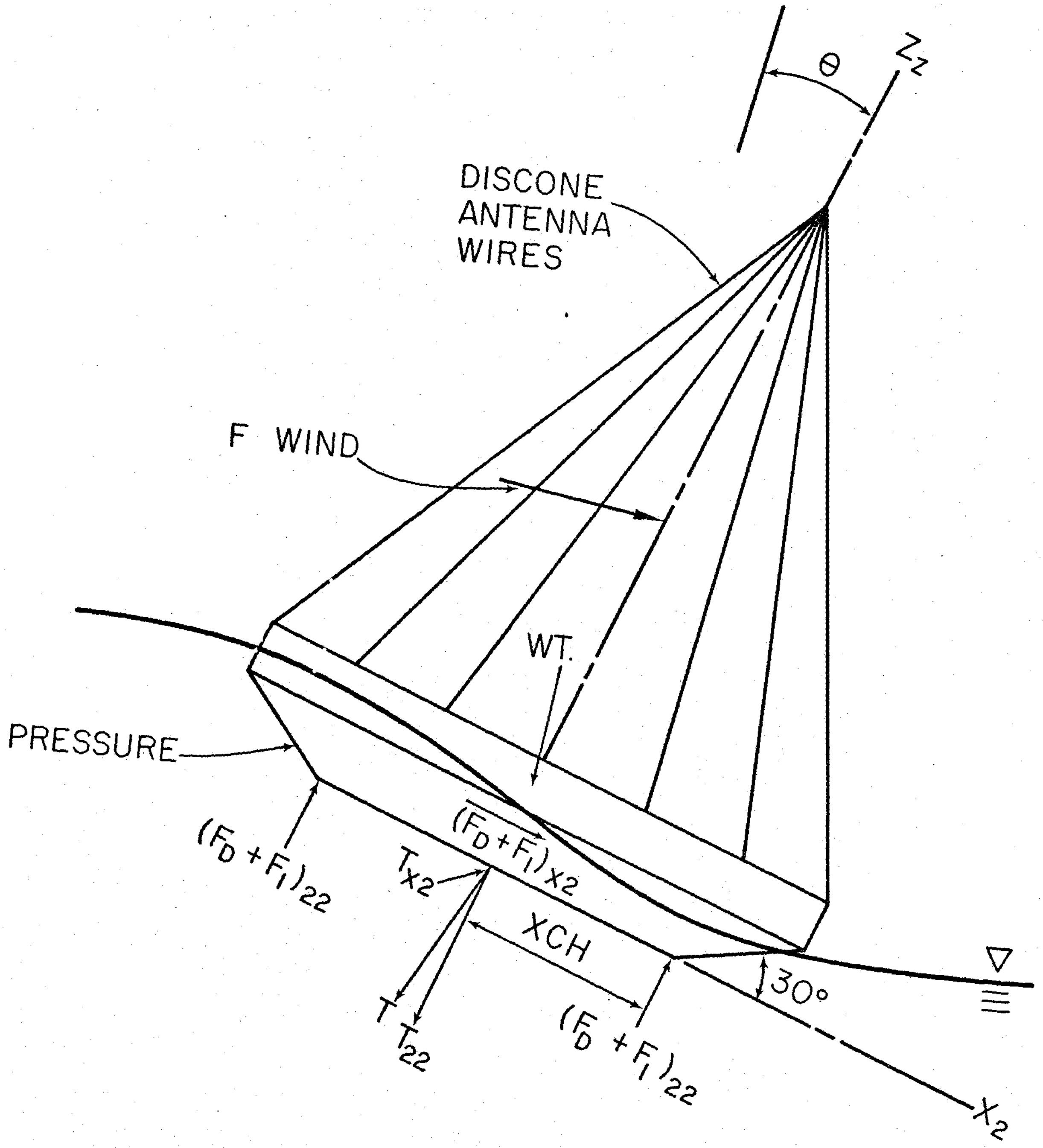


FIGURE 8. Free-Body Diagram of Buoy with All Forces Shown Schematically

Thus by changing only current forces and leaving other contributing loads unchanged we have,

F_{long} = 98,915 lbs,

F_{trans} = 39,567 lbs, and

F_T = 106,535 ≃ 1.1 x 10^5 lbs.

TABLE 6. NORMALIZED PLANT DRAG FORCES

Section	Area (ft^3)	Vel (fps)	F_D (lbs)
Plant	4,500	1.0	4,461
Pipe	2,500	1.0	2,475
	3,250	.91	2,664
	4,000	.79	2,472
	9,000	.61	3,315
	14,750	.42	2,576
	2,500	.30	223
		Total	18,186

Dynamic mooring: From equation (25), with a normalized velocity of 1 fps we have,

$$Q = F/\rho(V-1)$$

Thus, Hp and Mw values may again be tabulated as shown in Table 7.

TABLE 7. NORMALIZED MW AND HP CALCULATIONS FOR VARYING VELOCITIES

V ft/sec	Q ft^3/sec	HP	Mw
2	55,556	401	.3
4	18,518	535	.4
6	11,111	722	.54
8	7,937	917	.7
10	6,173	1,115	.83
12	5,050	1,314	1.0
14	4,273	1,513	1.1
20	2,778	2,007	1.5
25	2,222	2,509	1.9

Since, from equation (26) $D = [(4Q)/(\rho\pi V)]^{\frac{1}{2}}$, it follows that, at V = 20ft/sec, D = 9 ft., and at V = 6 ft/sec, D = 34 ft.

In this situation dynamic positioning is possible and since the plant is to be manned, the system would be reliable such as that proposed for Hawaii model city (Seidl, 1973).

STATIC MOORING - MOORING LINE ANALYSIS

For the analysis of the total tension in the mooring line of a static mooring system and hence the sizing of the line and anchor system, drag forces similar to those for the structure must be considered. This analysis would only consider static forces due to gravity, steady wind and current. Secondary considerations are the oscillating forces due to waves and gusty wind. Another oscillating force results from the steady drag on the body which gives rise to vortex shedding and thus oscillating forces previously mentioned which are perpendicular to the direction of the current, waves,,and wind (strumming). Longitudinal drag will effect the frequency response of the line (Nath, Yamamoto and Smith, 1974).

Another varying force which will effect the position of the buoy is current. This effect is discussed by Fofonoff and Garrett (1968), as shown in Figures 9 and 10, who plot the excursions of a single point moving surface buoy.

To account for the dynamic forces from waves and gusty winds a numerical procedure was developed by Nath and Felix (1970). The position and condition of the buoy and line due to steady forces were determined as the initial condition for the dynamic case. The resulting force equations which are keyed to Figure 8 are as follows:

$$(F_D + F_I)_{z_2} = C_{DAXL} AT/24\ V_{z_2} + C_{IAXL}\ P_T\ \overline{V}\ A_{z_2} \quad \ldots\ldots\ldots\ldots\ldots (29)$$

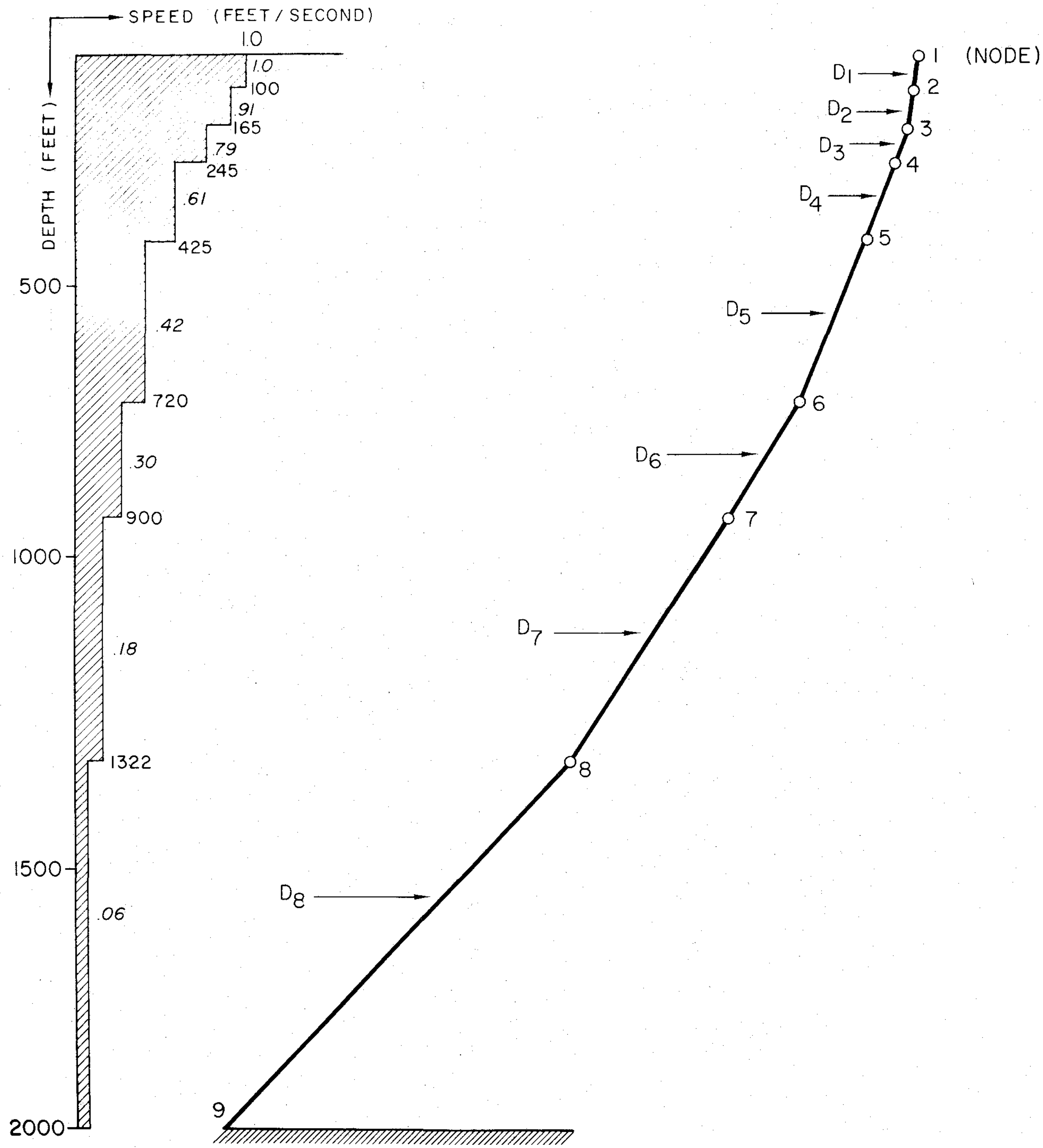


FIGURE 9. Polygonal Approximation

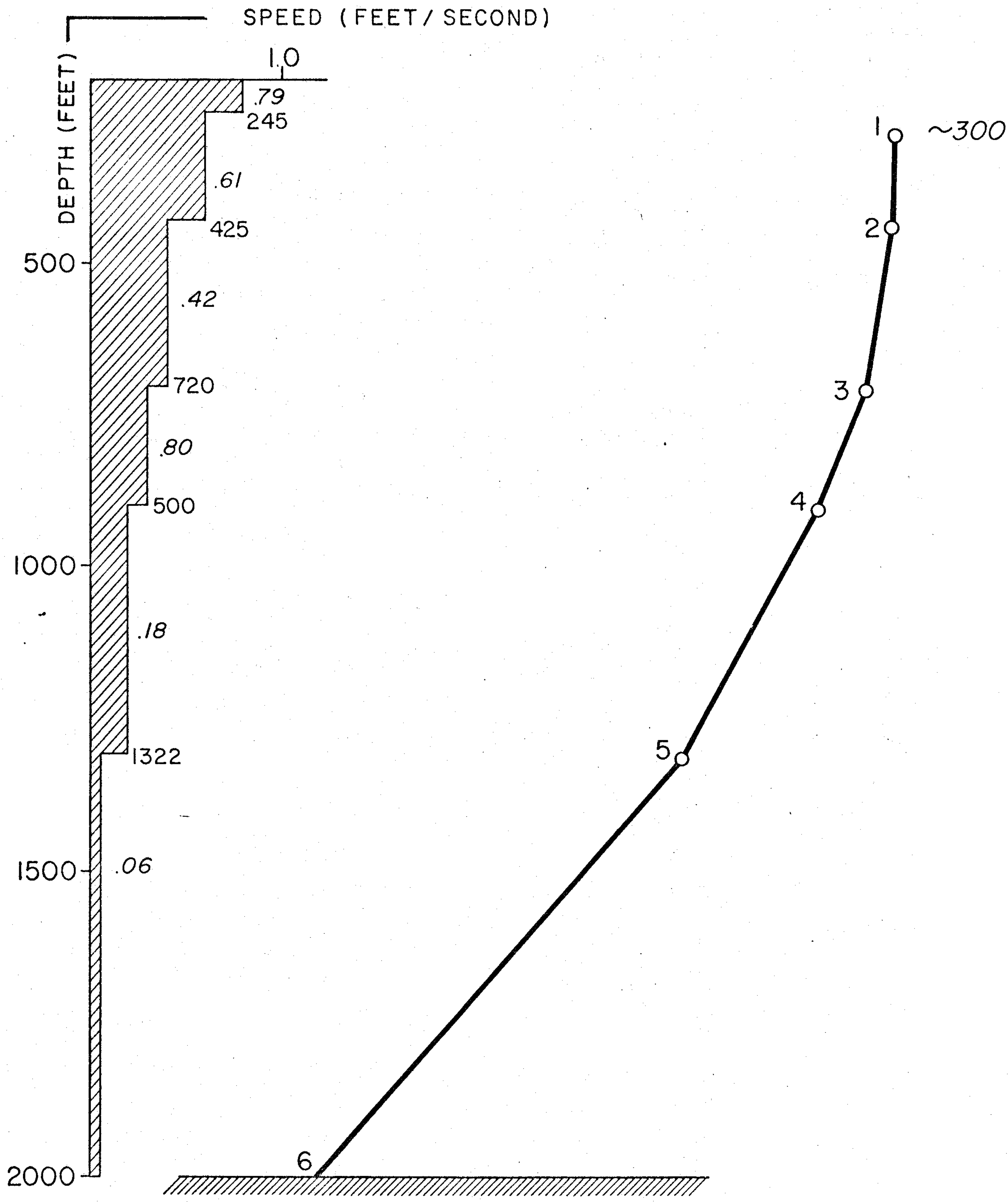


FIGURE 10. Polygonal Approximation

where

C_{DAXL} = drag coefficient in axial direction
C_{IAXL} = inertial coefficient in axial direction
A = vertical surface area
T = static line tension
V_{z_2} = relative velocity between water and buoy at lower chine
P_T = mass density of water psi cubic ft.
A_{z_2} = relative acceleration at lower chine
$\overline{V}$ = submerged volume

$$(F_D + F_I)_{x_2} = C_{DRAD}\ A\ P_T\ V_{x_2}|V_{x_2}| + C_{IRAD}\ P_T\ \overline{V}_T\ A_{x_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots (30)$$

where

C_{DRAD} = drag coefficient in x_2 direction
C_{IRAD} = inertia coefficient in x_2 direction
V_{x_2} = relative velocity between water and buoy
$\overline{V}_T$ = total submerged volume of buoy
A_{x_2} = relative acceleration between water and buoy

As summarized by Nath and Felix (1970), "the motions and position of all points on the line are determined from the forces acting on it and from the line state of the previous increment of time. The tension at the top of the line is determined from the strain at the top, which is influenced by the buoy position..."

A determination of the relative magnitude of the lift and drag forces at the surface can be approximated with the use of the following equation from Berteaux and with Figure 9.

$$T_i = \sqrt{(B_B - \sum_{n=0}^{n=i-1} W_n)^2 + (D_B + \sum_{m=0}^{m=i} D_M)^2} \quad \ldots (31)$$

where

B_B = product of total polygonal length by line density W
W_n = h(n) W/sinϕ(n) with h(n) = depth of uniform current region "n"
$W_0 = 0$
D_B = drag on the buoy
D_M = drag on segment m $D_0 = 0$
$= \frac{1}{2}\rho C d V_m^2 h_m$
with d = diameter of line
ρ = water density
h_m = width of layer "m"

and,

$$\phi_i = \tan^{-1} [(B_B - \sum_{n=0}^{n=i-1} W_n) / (D_B + \sum_{m=0}^{m=i-1} D_m)] \quad \ldots . (32)$$

where

T_i and ϕ_i are the tension and angle of inclination of line at node "i" respectively.

For a normalized velocity profile given in Figure 9, and using a 1.75" wire rope (3 x 46),

D_B = 1.1 x 10 lbs,
W = 5.12 lb/ft,
B_B = (20,000 ft)(5.12) = 102,400 lbs,
d = .146 ft, and
C = 1.0.

These calculations are summarized by node in Table 8.

It is obvious from the angle of inclination of the line that tow under will probably occur due to the inclusion of dynamic forces. To avoid this condition a multi-point mooring should be utilized to reduce the amount of drag experienced by any one line. This would reduce the value of D_B and thus increase the scope thereby increasing the angle of inclination.

TABLE 8. MOORING LINE FORCES BY NODE

Node	ΣW_n (lb)	ΣD_m (lb)	T_i (lb)	ϕ_i °
0	0	0		
1	915.6	99	149,736	43.0°
2	1,407.3	152.3	149,442	42.6°
3	2,013.6	201.7		42.5°
4	3,383.0	231.2		42.3°
5	5,643.2	462.3		41.9°
6	7,041.9	488.4		41.2°
7	10,348.6	501.9		40.8°
8	15,772.2	503.9	140,412	39.8°

If the scope = 12 for a single point mooring,

W = 30 lb/ft,

$\phi_i = \tan^{-1} 6.55 = 81.3°$, and

T_i = 728,354 lbs.

However, one is now outside the present state-of-the-art for anchorage systems which will hold this large a force.

However, if a multi-point is used, the system will loose the capability to surface. Since this is a primary prerequisite of the design, a multi-point mooring cannot be recommended.

COMBINED SYSTEM

If the plant is stabilized by both dynamic and static mooring, i.e., dynamic when surfaced and static when submerged, then the drag force will have an order of magnitude of 1.0×10^4 lb when conditions have been normalized (1 knot at surface). For this case:

Strouhal number = fD/V where the diameter = 1" wire rope (3x46),

R = range $6.6 \times 10^3 - 1.5 \times 10^3$

s = .2

$f = .2V/D$

range = 1.46 to .14 HZ

Equations for modal frequency for transverse vibrations (Nath, et al, 1973) is:

$$f_i = (T/u)/(2L) \quad \text{.................} (33)$$

where

f_i = modal frequency in HZ,

T = average line tension,

u = effective mass per foot, and

L = line length.

Then,

$B_B \simeq (\text{length})(W_{w_r})$ + buoyancy of buoy (reserve)

where

$W_{w_r} \equiv$ lb/ft for 1" wire rope (3 x 19)

or W = 5.12 lbs/ft.

Thus, $B_B \simeq$ 18,000 lbs,

W = 5.12 lbs/ft,

$T \simeq$ 20,000 lb,

u = .16 slugs/ft

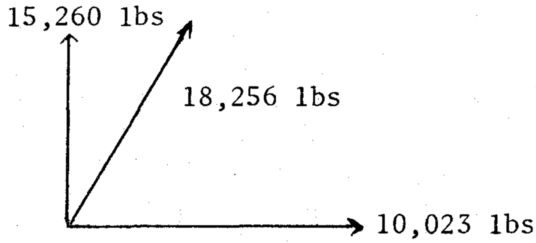
L = 17,000 ft, and

f = 7.35 HZ. Therefore, no reasonance is to be expected and a static approach can be taken. Using Figure 10, Table 9 is generated.

TABLE 9. COMBINED SYSTEM MOORING LINE TENSION

Node	ΣW_n lb	ΣD_m lb	T lb	ϕ
1	223	7.0		60.9°
2	751	14.5		60.6°
3	1,076	17.0		59.9°
4	1,841	19.0		59.4°
5	2,740	19.4	18,256	58.2°
6				56.7°

length of line 2,000 ft.,
free body at anchor



Note: Wire rope has been used because 1) failure of other types of lines from fish bites; 2) buoy will not be surface-following, so elasticity is not needed; and 3) required accurate position against large loads.

Thus when plant is surfaced the drag force will be overcome by dynamic positioning and the mooring line will support only itself and the reserve buoyancy of the plant. Rechecking frequency of shedding,

f = .14 to 2.4 cycles/sec,
frequency of wave of T = 10,
f = .63 cycles/sec.

Therefore, there will be no resonance and the system can still be considered static.

MOORING SYSTEM SUMMARY

The analysis of a mooring system essentially is composed of three parts: 1) investigation of resonance frequency of system versus oscillating forces; 2) static analysis of system to design conditions; and 3) dynamic interaction of system with forces. Presently many areas remain under investigation, such as 1) the values of C_D, C_I and C_L to use and under what conditions the values can be used; 2) the dynamic interaction between forces and system; 3) mooring motion and response of mooring lines, etc. In practice the designer relies heavily on Morrison's equation, Airy wave theory, a and C_D and C_I values derived from steady, uniform flow conditions. Many unknowns exist as to the validity of this method; therefore, normally large safety factors are used and the system is over designed so that it will last. The state-of-the-art is not at the point where a most economical design can be used; therefore, in this approach to the problem of static vs dynamic mooring, an order of magnitude study was used to present a logical approach to the problem and point out areas where refinements can be made.

CONCLUSIONS

1. The results of this study indicate that submerged plant operation has considerable advantage over surfaced operation. Surface wave and wind forces are essentially eliminated and current forces substantially reduced.

2. Dynamic mooring is impossible for the 25 Mw plant in a high velocity current as more power is required for that mooring than is produced. It appears that, while dynamic mooring may become feasible for larger plants, the excessive amount of power required will make it undesirable.

3. Based on this study, it can be concluded that the forces generated in a high velocity current are such that dynamic positioning is not feasible and static mooring reliability at best only about 54%, based on the presentation by Walden and Panicker (1973). In fact, current is one of the main sources of failure experienced in that study where a high current was considered approximately two ft/sec. Therefore, an area with a low current profile is desirable for mooring, and in this situation the best mooring for a "surfacing" plant might be a combination static-dynamic system, i.e., static submerged, dynamic surfaced.

4. Multiple point mooring shows significant advantages for on-site mooring. However, the requirement for periodic plant surfacing negates their use.

5. Extensive additional research is needed in the general area of mooring line analysis. Current state-of-the-art limitations must be pushed back a considerable distance in order to fully specify the requirements of a Sea Solar Power Plant mooring and anchoring system.

REFERENCES

Bascom, Willard, Waves and Beaches. New York: Doubleday & Company, Inc. 1964.

Beard, R.L., Statistical Methods in Hydrology. U.S. Army Office of Chief of Engineers, July, 1952.

Berteaux. "Introduction to the Status of Single Point Moored Buoy Systems", Woods Hole Oceanographic Institute, Technical Memo No. 11-68. 1968.

Bishop and Hassan, "The Lift and Drag Forces on a Circular Cylinder Oscillating in a Flowing Fluid", Proceedings of the Royal Society (London). Series A, Vol. 277: p. 51-75. 1964.

Bretschneider, C.L., "Wave Variability Spectra for Wind Spectra for Wind-Generated Gravity Waves." Tech Memo No. 118, Beach Erosion Board. U.S. Army Corps of Engineers. 1959.

Dantz, P.A. "The Padlock Anchor - A Fixed Anchor System", Naval Civil Engineering Laboratory Technical Report No. R 577. U.S. NCEL: Port Heuneme, California. 1968.

Daugherty and Fromzini. Fluid Mechanics with Engineering Applications. New York: McGraw-Hill. 1965.

Falvey, Henry T. "Prediction of Wind Wave Height", Journal of the Waterways, Harbors, and Coastal Engineering Division. ASCE IDD(WWI): 1-12. February, 1974.

Fofonoff and Garrett, "Mooring Motion", Woods Hole Oceanographic Institute Technical Bulletin. No 58-31, 1968.

Fluid Mechanics for Hydraulic Engineers. Dover: Hunter House, 1961.

Gumbel, E.J. Statistics of Extremes. New York: Columbia University Press, 1958.

Ippen, Arthur T. Estuary and Coastline Hydrodynamics. New York: McGraw-Hill Book Co. 1966.

Kinsman, B. Wind Waves. New Jersey: Prentice-Hall. 1965.

McCormick, Michael E. Ocean Engineering Wave Mechanics. New York: John Wiley & Sons. 1973.

Myers, John J. Handbook of Ocean and Underwater Engineering. New York: McGraw-Hill Book Company. 1969.

Nath, John. "Some Considerations for Buoys and Their Moorings." Oregon State University Circular No. 41. 1971.

Nath, et. al. Offshore Research Tower. Oregon State University Department of Oceanography. December, 1973.

Nath and Harleman, "Response of Vertical Cylinders to Random Waves", Journal of Waterways and Harbor Division. ASCE. May, 1970.

Nath, Yamamoto and Smith, "Longitudinal Motion of Taut Moorings", Journal of Waterways and Harbors Division ASCE. February, 1974.

Oregon State University Engineering Experimental Station. Seminar Proceedings: Topics in Ocean Engineering. OSU Engineering Experimental Station Circular No. 41, 1971.

Plate and Nath, "Modeling of Structures Subjected to Wind Waves", Journal of Waterways and Harbors Division, ASCE. November, 1969.

Quinn, Alonza DeF. Design and Construction of Ports and Marine Structures. New York: McGraw-Hill Book Co. 1972.

Seidl, "Hawaii Floating City Development Program", NSF Technical Report No. 2, May, 1973.

Smith, J.E. "Investigation of Free-Fall Embedment Anchor For Deep Ocean Application", Naval Civil Engineering Laboratory, Technical Note No. 805, Port Hueneme, California. 1966.

Smith, J.E., R.M. Beard, and R.J. Taylor. "Specialized Anchors for the Deep Sea" Naval Civil Engineering Laboratory Technical Note No. 1133, Port Hueneme, California. 1970.

Taylor, R.J. and R.M. Beard, "Propellant-Actuated Deep Water Anchor: Interim Report", Naval Civil Engineering Laboratory Technical Note No. 1282, Port Hueneme, California. 1973.

Taylor, R.J. and H.J. Lee, "Direct Embedment Holding Capacity", Naval Civil Engineering Laboratory Technical Note No. 1245, Port Hueneme, California. 1972.

Tudor, W.J. "Mooring and Anchoring of Deep Ocean Platforms". 1967 ASCE Conference, Proceedings, San Francisco, California. 1967.

U.S. Navy, Naval Facilities Engineering Command. Design Manual - Harbor and Coastal Facilities. NAVFAC DM-26. July, 1968.

U.S. Naval Oceanographic Office. Oceanography Atlas of the North Atlantic Ocean. Pub.No. 700, Sec. IV. 1963.

Walden and Panicker. "Performance Analysis of Woods Hole Taut Moorings", Woods Hole Oceanographic Institute Technical Report. No. 73-31. June, 1973.

Weigle, R.L. "Waves and Their Effects on Pile Supported Structures", Seminar Proceedings: Topics in Ocean Engineering. OSU Sea Grant Circular No. 41. 1971.

Yamamoto, Nath and Slotta. "Yet Another Report on Cylinder Drag or Wave Forces on Horizontal Submerged Cylinders", OSU Bulletin No. 47. OSU, 1973.

ANNEX 1.

ANCHORING SYSTEM EVALUATION

CONVENTIONAL ANCHORS

Other than in the case of "dynamic positioning", an anchorage system must rely on its attachment to the ocean floor to obtain fixity at one location. Securing to the ocean floor may best be achieved with implements embedded deeply and firmly into the bottom material. Design of conventional anchorages generally utilizes the standard drag-type anchors.

The following characteristics of conventional anchors for adequate holding power and reliability are summarized from the Naval Facilities Engineering Command design manual (NAVFAC DM-26, 1968).

1. Correct angle between flukes and shank for digging in (35 degrees for sand and 50 degrees for mud)

2. Large fluke area for anchor weight.

3. Large tripping palms for use in mud.

4. Sharply pointed, thin flukes.

5. A stabilizer or stock to prevent cockscrewing or rotating.

6. Ability to bury itself quickly without excess drag.

7. Good efficiency measured by the ratio of holding power to weight.

8. Strength and ruggedness to withstand maximum holding powers, vertical breakouts, and lateral pulls.

Several serious disadvantages of conventional anchors are:

1. Considerable horizontal distance is required to adequately set this type of anchor, necessitating large amounts of rope and gear and a massive mooring complex to make the plant.

2. Resistance to uplift is small and minor uplift loads can seriously reduce normal holding capacity.

3. Since the anchor offers primarily horizontal resistance, even a group of anchors placed at 120 degrees to each other may not provide the desired restraint for a majority of contemplated structures which will impose large uplift forces.

Schematic diagrams and details of conventional anchor design are discussed by P.A. Dantz (1968).

DIRECT EMBEDMENT ANCHORS

At present, the primary means of securing structures to bottom in depths 500 feet and greater is by using deadweights. Many shapes, materials and constructions are used, but all gravity anchors have low efficiency developing only their own weight in resistance to horizontal forces. Also, they are unstable and unreliable on sloping bottoms.

Conventional drag-types are unsatisfactory for use in deep water. They are designed to be embedded into the bottom by dragging to develop horizontal pull resistance and once embedded they lose holding capacity if uplift forces are applied. Generally, a dragging placement procedure may be impractical in water more than 500 feet deep. Also in great depths, it is difficult to prevent uplift forces from being applied. To overcome dragging and the uplift force problem, conventional anchors sometimes are used in tandem on the anchor line with another anchor or a deadweight. Though slight advantage gained by this procedure, handling and placement problems are increased.

The direct embedment anchor best satisfies the requirements of a deep ocean anchorage. Two advantages of this type anchor are the capability to embed directly into the bottom without the necessity of dragging and the capability to resist significant uplift loads. Direct embedment shortens the lowering and placement time, enhances the precision of placement, and reduces the quantities and sizes of accessory gear. Uplift resistance provides for both reduced amounts and variety of connective gear and greater flexibility of application of the anchor. Of the direct embedment anchor types, eg. vibratory, propellant-actuated, hydrostatic, free-fall; the propellant-actuated anchor proved most simple, reliable and economically feasible. Characteristics of one propellant-actuated anchoring system are shown in Table A-1.

Although additional tests will be required at various depths and into different types of seafloors to fully characterize this type of anchoring system, initial trials indicate that the explosive type of propellant activated anchor is: 1) most workable; 2) achieves acceptable holding capacities in silt seafloors; 3) assemble quickly and easily; and 4) revolutionary fluke keys rotate quickly and show no distress from penetration or pullout (Taylor and Beard, 1973).

Detailed diagrams and design specifications of direct embedment anchors are discussed by Taylor and Beard (1973).

HOLDING CAPACITY

Techniques for predicting the maximum uplift forces which may be applied to direct embedment anchors without causing the anchor pullout are provided by Talor and Lee (1972).

$$F_T = A(C\overline{N}c + \gamma_b D\overline{N}q) \quad \ldots\ldots\ldots\ldots\ldots (1)$$

where

F_T = holding capacity (lbs)

A = fluke area (ft^2)

C = cohesion of soil (psf)

γ_b = buoyant unit weight of soil (psf)

D = fluke embedment depth (fl)

$\overline{N}c, \overline{N}q$ = holding capacity factors

For rectangular flukes,

$$F_T = A(C\overline{N}c + \gamma_b D\overline{N}q)(0.84 + .16\ B/L) \quad \ldots\ldots\ldots\ldots (2)$$

where

B = fluke diameter or width

L = fluke length

Strength of soils by the Mohr-Coulomb equation,

$$\tau_f = C + N \tan\phi \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

where

τ_f = shear strength

C = cohesion

N = normal force on failure plane

ϕ = initial friction angle

For cohesive soils, equation (2) reduces to:

$$F_T = A(C\overline{N}c + \gamma_b D)(0.84 + .16\ B/L) \ldots (4)$$

TABLE A-1. A PROPELLANT-ACTUATED DIRECT EMBEDMENT ANCHOR (Taylor and Lee, 1972)

Characteristic

Functional water depth -------------- 100-20,000 ft.

Capacity-----Minimum long term holding capacity - 20,000 lbs.

	Sand fluke	Clay fluke
Length (in)	38	63
Width (in)	18	30
Plan area (ft)	4.5	12.5
Fluke weight (lbs)	147	337
Piston weight (lbs)	116	116
Piston extension weight (lbs)	24	24
Connective gear weight (lbs)	13	13

Gun barrel	37 inch long, 26 inch projectile stroke, 11 inch cartridge
Short start pressure	3,000 psi
Propellant	Navy Pyrotechnic
Projectile weight	300 lbs and 490 lbs
Launch vehicle weight	1,540 lbs
Projectile muzzle velocity goals	225 fps, 490 lb projectile 275 fps, 300 lb projectile
Maximum allowable gun barrel pressure	35,000 psi

$$\bar{N}c = 3.8(D/B)(.7/C + 0.3) \ldots\ldots\ldots(5)$$

or, $\bar{N}c = 9$, whichever is smaller, for $0.75 \text{ psi} \leq C \leq 4 \text{ psi}$.

For cohesionless soil, equation (2) reduces to:

$$F_T = A\gamma_b D\ \bar{N}q(0.84 + .16\ B/L) \ldots\ldots(6)$$

INSTALLATION

Having characterized the several types of anchoring systems, a brief review of installation techniques should be made. Three installation techniques will be covered: free-fall, dragging (both used for conventional anchoring systems), and propellant-activated (used for the direct embedment anchor).

Free-fall: The free-fall technique is adequately described in Figure A-1.

Dragging: Application of the requisite horizontal force at great depths require excessive gear, and it is difficult if not impossible to sense the amount of movement and depth of embedment achieved.

Therefore, the current procedure is to determine the amount of tension desired in the connecting apparatus during installation and then to drag the anchor until this amount of tension is achieved. Next, the tension on the line is kept within established maximum and mimimum limits until installation is complete. Thereafter, it can be deduced by performance of the moor whether or not the anchor is embedded as desired.

Direct embedment anchor (Propellant-actuated anchor): The anchor assembly and the launching system is lowered into the sea. When a probe protruding 26 inches below the fluke tip contacts the seafloor, the firing sequence is initiated. After penetration is complete, a small pull on the main cable causes the fluke to keg (rotate into its resistive position). The launch vehicle is retrievable where the anchor deployments in water depths is less then 500 feet.

SELECTION OF ANCHORING SYSTEM

Anchor designs, special appertenances, connective gear, and special operational techniques are among the more critical areas that need improvement. The single most important consideration is the anchor design as improved anchors can alleviate problems in the other areas. While considerations such as expense, ease of handling and storage will influence the choice of anchor, the single most important characteristic is the anchor's holding power. It is this fact which makes the holding power to weight ratio so important. For any given anchor, this ratio is not fixed, but depends particularly on the nature of ocean bottom. Certain anchors are designed for certain types of bottom considerations and therefore may not work well when they encounter other than the design type surfaces. Formulas to roughly estimate the holding power of various types of anchors for different bottom conditions have been developed, but for any given situation the actual holding power of an anchor should be determined in tests at the site.

Selection of a specific anchoring system for the Sea Solar Power Plant cannot appropriately be made without the inclusion of a number of considerations outside the scope of this paper. Figure A-2 details a schematic for one approach to anchor system selection. While not all background information required for full use of this schemitized method is presented in this paper, the required steps and step flow ordering can still be understood.

At some stage of development of a Sea Solar Power Plant, a similar analysis will be required to determine an actual anchoring system. Beyond the general discussion included in

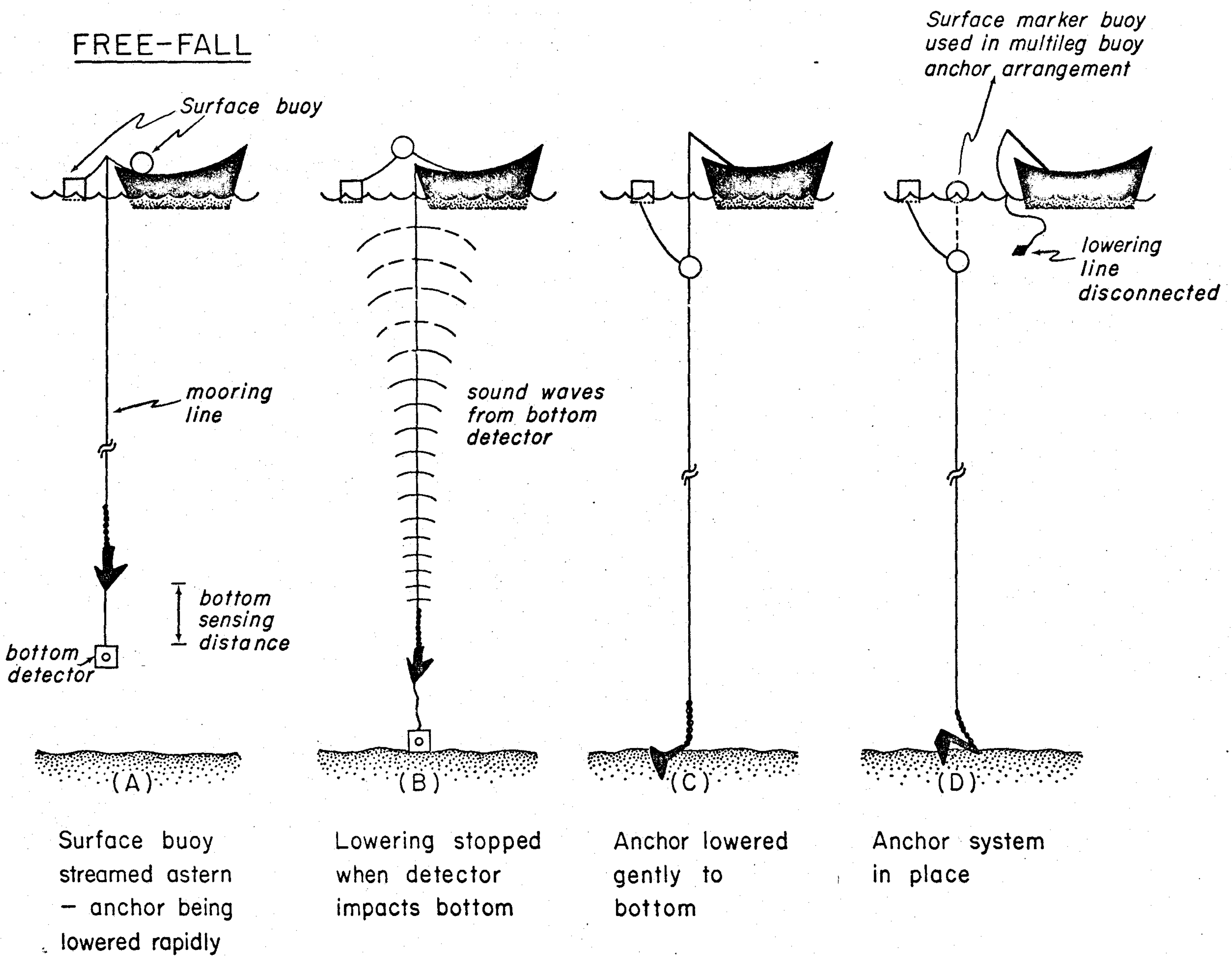


FIGURE A-1. Free-Fall Installation Technique

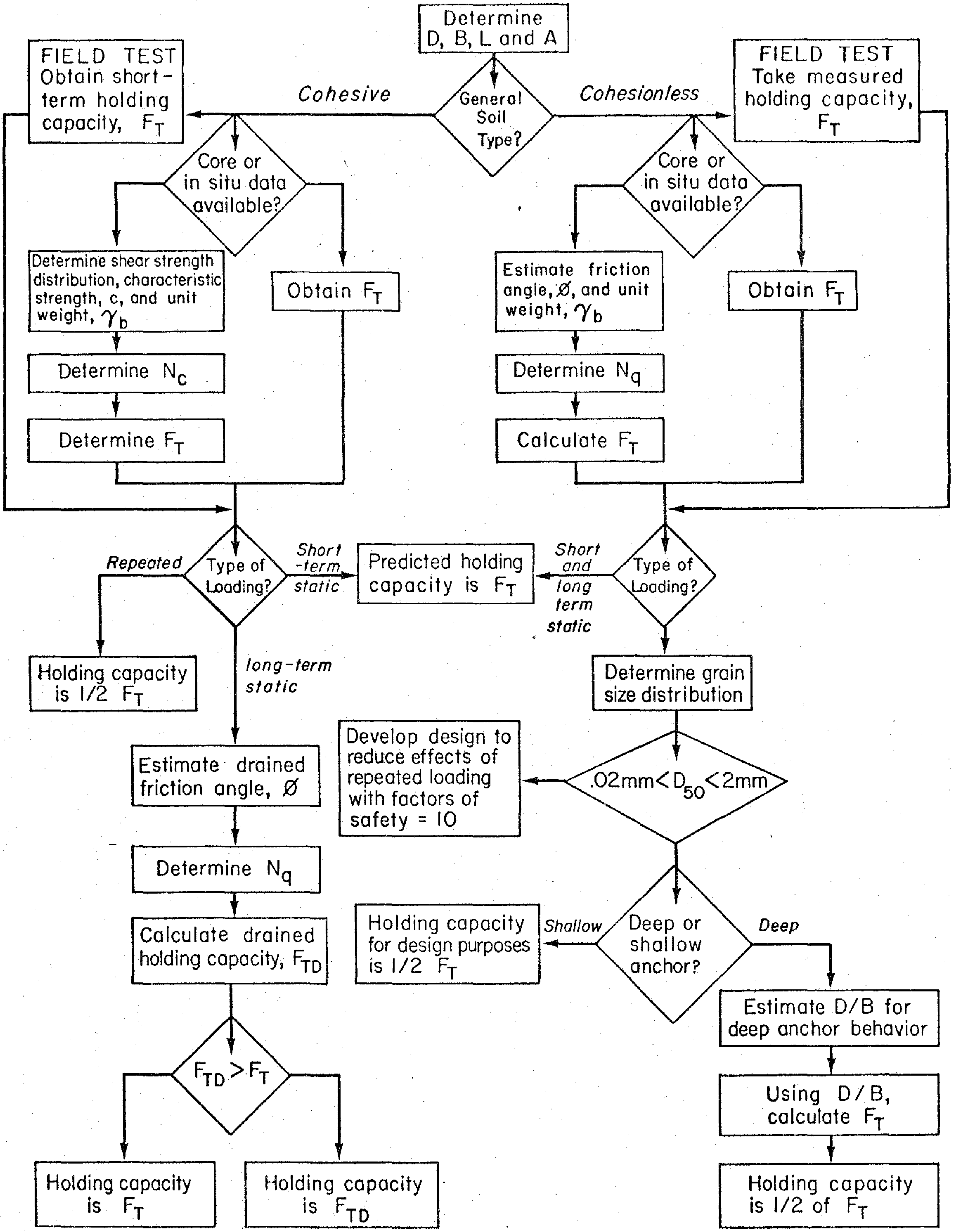


FIGURE A-2. Block Diagram Illustrating Suggested Procedure for Predicting Embedment Anchor Holding Capacity this Annex, Table A-2 lists the characteristics of a number of anchoring systems which could be used to support such an analysis.

Certain general conclusions may be drawn about the deep water anchoring system:

1. Conventional Anchors (see Squaw Anchorage System): To prevent uplift force, deadweights were used. This method can be applied where the bottom is essentially level and currents in the area are of low magnitude.

2. Direct Embedment Anchors: The propellant-actuated type, as compared with other types of embedment anchors (vibratory, hydrostatic, free-fall), offers simplicity, reliability, and economy. Results indicate a 300-lb embedded anchor can develop 20,000 lbs holding power.

TABLE A-2. ANCHOR SYSTEM CHARACTERISTICS

Type	Characteristics
Mass anchor.	Simple. Holding power to weight ratio is low. Handling and transporting is expensive. These anchors are not reliable on sloping hard bottoms and are easily moved laterally when loaded with a vertical and horizontal load.
Mushroom anchor.	Poor penetrating powers. It is considered a good mud anchor usually sinking by its own weight through soft mud to firm strata, and getting good bearing against a large volume of mud ahead of it. However, if the anchor should ever drag under load, it will probably come out.
Stock (Admiralty) anchor . .	Old fashioned or common anchor. The area of the palm is not so large for its weight and, therefore, it depends on digging deep to get good holding power.
Navy stockless anchor. . . .	Approximate holding-power-to-weight ratio of 6 to 1 in a sand bottom and 2 to 1 in a mud bottom. Anchors weighing from 20,000 to 30,000 pounds do not dig into sand until the fluke angle has been restricted to 35 degrees. A stabilizer bar increases the holding resistance by approximately 10 percent.
Lightweight anchor	Fluke angle is 30 degrees. The holding power i in sand does not vary as a constant times the weight of the anchor, but approximately as a constant times the weight raised to the 9/11th power. As the size of an LWT anchor increases, the holding power drops off in relation to its weight.
NAVFAC STATO anchors	Rates of holding power to weight larger than 20 to 1 in sand or 15 to 1 in mud can be obtained if the anchors are preset, or if the mooring site will permit drag distances longer than 50 feet. Anchors are not to be loaded beyond 30 times the anchor weight.
Umbrella pile anchor	High holding capacities and resistance to uplift where conventional anchorage systems are unsuitable. It is limited to uncemented homogeneous soil. Driving anchor in over 200 feet is difficult. A hammer with energies from 15,000 to 20,000 foot-pound is required for emplacement.

TABLE A-2. cont.

Stimson deep-sea anchor . . .	High holding power by its own weight plus the resistance to pull out due to its embedment. It should be lowered with sufficient tension to ensure its landing in an upright position.
Free-fall embedment anchor. .	The holding-power-to-weight ratio is approximately 3 or 4 to 1. The free-fall anchor concept as it applies to the specified objectives and criteria is not feasible. A minimum ratio of 7 to 1 is considered necessary.
Screw type embedment anchor .	Suitable for pipeline anchoring system. Shallow water use. Driving anchor in over 200 feet is difficult.
Explosive (Propellant-actuated) embedment anchor. .	Opening fluke and shieldlike shape type. Three different flukes are needed for sand, clay and rock conditions. The holding power of the 50-kip rated in coastal waters has been recorded at 60,000 pounds in sand and clay and the 300 kip rated anchor has been recorded at 200,000 pounds holding power in sand and clay. Quick and easy handling, assembling.
Padlock anchor.	A tripod configuration with bearing pads located at the extremities of the arms. The reliability of this system depends on the reliability of the embedment anchors. Successful discharge and test loading to the depth of 6,000 feet have demonstrated the potential of this anchor complex for deep ocean application.
Pulse-jet anchor.	Two principal parts called a mass drag reactor and a ballistic embedding anchor. It was unable to achieve an experimental model of the design envisioned.
Vibratory anchor.	Large flukes for low strength soils and small flukes for high strength soil. Concept is feasible, exceptional holding capacities were achieved in several tests. Anchors take vertical pull over 80 times its own weight. Handling is simple. Balanced design that matches the energy required to achieve proper embedment with fluke size to obtain the rated holding capacity in different types of seafloors.

SEA SOLAR POWER PLANT - GEOLOGY AND BIOLOGY OF THE MIAMI PLANT SITE

by

Bruce Higgins

ABSTRACT

The criteria for an idea site are outlined and the pros and cons of this site are discussed in the geological and biologicallaspects. The narrow continental shelf gives good access to cold, deep water as well as the Florida current continuously refresshing the supply of both surface and bottom waters. The offshore sediments, their strength and composition are discussed. Nearby cores are brought to light for the anchoring of the plant and the animals which will be living there. The fishing, both sport and commercial, are discussed. An outline of the specific studies to be conducted prior to construction are included. A brief investigation of the variability of sediments and transits in animals is brought into focus.

SEA SOLAR POWER PLANT - GEOLOGY AND BIOLOGY OF THE MIAMI PLANT SITE

by

Bruce Higgins

INTRODUCTION

The proposed site for an offshore thermal power plant is the Miami, Florida area. We will look here at the basic geological, physical, and biological features of the area. The reasons this site was chosen become apparent when viewed in the light of the general site analysis. The principal elements and critical factors to be investigated include:

1) Bottom Survey
The water should be deep nearshore and have a hard bottom. The area should be geologically stable.

2) Water Temperature
The water should be very stratified, and have a minimum of seasonal variations.

3) Currents
Depending on the design, they can be avoided or greatly depended on. They should be well accounted for and not changing seasonal in site area.

4) Wave, Wind, and Tide
These are important if the plant is to remain long on the surface for repairs and effort should be made to minimize their presence.

5) Geothermal
Other sources of hot water should be looked for other than the ocean and incorporated if feasible.

6) Water Quality
A broad term which includes chemistry and biological activity in the area. Effort should be made not to disturb "Mother Nature" more than absolutely necessary.

PHYSICAL AND CHEMICAL PARAMETERS

Basic physical and chemical background information has been compiled in Table 1. The facts look dry but some very important observations come from this information.

TABLE 1. MIAMI AREA

Physical Parameters	Depth 200 ft	1000 ft
Temperature	21° - 26° C	9° - 10° C
Salinity	36.3°/00	35°/00
Current Velocity	2.0 fps	0.9 fps
Total Transport	26 x 10^6 m^3/sec	
Weather - Hurricanes	1 per 3 years for 70 years	

Chemical Parameters	Depth 200 ft	1000 ft
Dissolved Oxygen	5.5 mℓ/ℓ	2 mℓ/ℓ
Total Nitrogen	1 μg-atom N/ℓ	6 μg-atom N/ℓ
Nitrate	1 μg-atom N/ℓ	50 μg-atom N/ℓ
Inorganic Phosphorus	0.1-1 μg-atom P/ℓ	4 μg-atom P/ℓ
Dissolved Organic Phosphorus	0.01 μg-atom P/ℓ	0.5 μg-atom P/ℓ
Organic Carbon	1 mg C/ℓ	8 mg C/ℓ
Silicate	10 μg-atom Si/ℓ	160 μg-atom Si/ℓ
Carbon Fixation	150 1150 mg/m^2day	--
PH	8.2	8.1
Light	1/30 to 1/50th of surface value	none

The physical parameters show the site to have a large stable temperature gradient which is disturbed primarily by hurricanes which can leave large cold water pockets near the surface for up to eight days. These have been observed in the Gulf of Mexico and there are some theories on their formation. This is an area of research where attention should be paid to future developments. The velocity of the current and the mass transport are not as stable as one might believe. So, plots were obtained for yearly mass transport (Figure 1) and for average surface velocity (Figure 2) which show interesting variations. The designer will have to study the specific site with care, but the peak values would be useful.

The chemical parameters show that there is a wide fluxuation in values with depth. A graph of dissolved oxygen (Figure 3) demonstrates the magnitude of the fluxuations of which the Florida waters are capable. In most cases it appears that there is a larger quantity of desired nutrients at depth. The transport of these to surface layers by the plant is being studied for aquaculture potential. A monitering station and search for more complete data in the area would prove very valuable.

SITE GEOLOGY

The geology of the area provides very deep water very near shore. This region has one of the narrowist continental shelves on the eastern sea board. The sediment has a knoll like feature due to possible solution holes from previous up lifted time. The included chart of the area (Figure 4) does not include all of the details known of the area, but the overview information is sufficient at this time. The near shore and sub bottom geology are included in Figure 5 to present a general picture of the composition of the coast.

Charts in Figures 6 and 7 provide more detail of on site bottom sediment and composition as this information is critical to anchoring system design. The sediment is thick with 300 feet appearing maximum closer to shore and the area composed of calcareous ooze and forminiferal sand. The escarpment though is as free as can be found of sediment with bed rock showing in places. The deep trough at the base of the escarpment is an errosional feature and is maintained by the Florida Current.

A fault zone through the area brings up the question of area stability. In recent years, the number of recorded quakes in this area are few, and of very low magnitude. The detailed record of these could not be found but would be very useful information.

The sub bottom geology is most easily seen through the use of Table 2 which includes information on the penetrometer readings of a nearby site. More detailed information for the specific site is desired but this available data provides insight as to the range of numbers expected at the sea solar power plant site.

SITE BIOLOGY

The biological species of the region are best handled in the form of lists as their behavior in the off shore waters of the plant site are not fully understood. Table 3 gives the major benthic types found in these waters, at the bottom depth of the plant site. Major fish species are summarized on Plates 1 - 7. Details on the characteristics and tolerance limits of each one are out of the scope of this report. The fish are known to school and the catches of economically important fish are listed on Table 4 along with the catch obtained by fishing fleets of the region. The shell fish of the retion are also very important and a list of these is included in Table 5. The size of the numbers reflects the catches near shore as it has been observed that few fish are taken in water deeper than 300 feet depth and that there are no important fishing grounds east of 80°W. The phytoplankton of the region will surely travel through the plant no matter what plant design is used. Species known to exist in the site area are included in Table 6.

Typical diurnal migration of these zooplankton is shown in Figure 8 which gives magnitude of the population and depth variation with time in one day. From this figure we see that the plant will not be able to avoid influencing a portion of the plankton population in the region. More detailed work is needed near the plant site to isolate the dominant fish and plankton species.

CONCLUSIONS

This site is found to be a very favorable location. The questions of the fault and depth of sediment still must be looked into but are not drastically different from most of the east coast. The biology of the site is favorable in that there is diverse, and extremely active populations which is much more adaptable to the changes that might be imposed by the plant. This paper was intended to be an overview of the information on the Miami area. Much more detailed on-site work will be required to define the environment of this area and the impact of a sea solar power plant upon it.

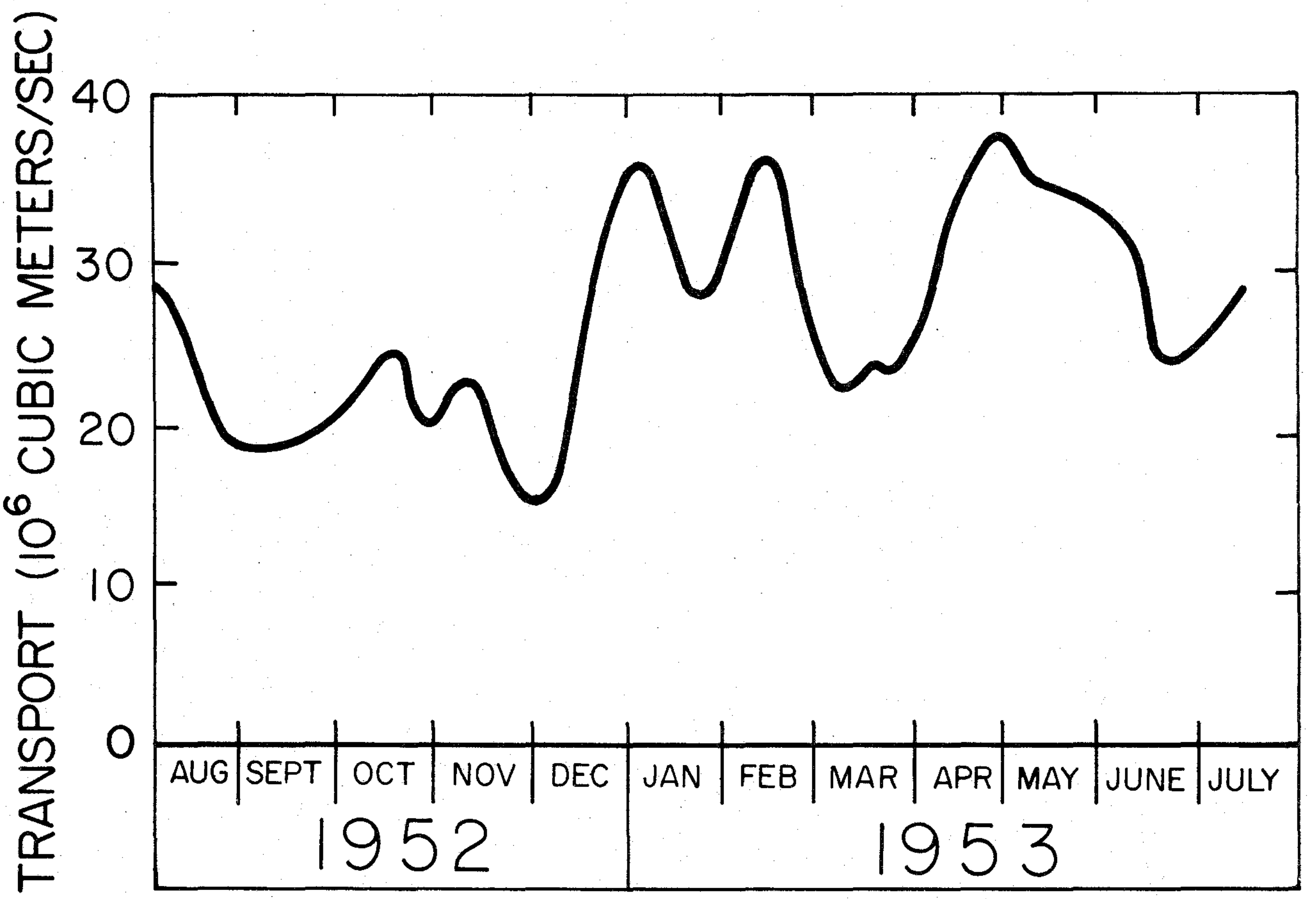


FIGURE 1. Mass Transport of Florida Current - Key West to Havana (Stommel, 1965)

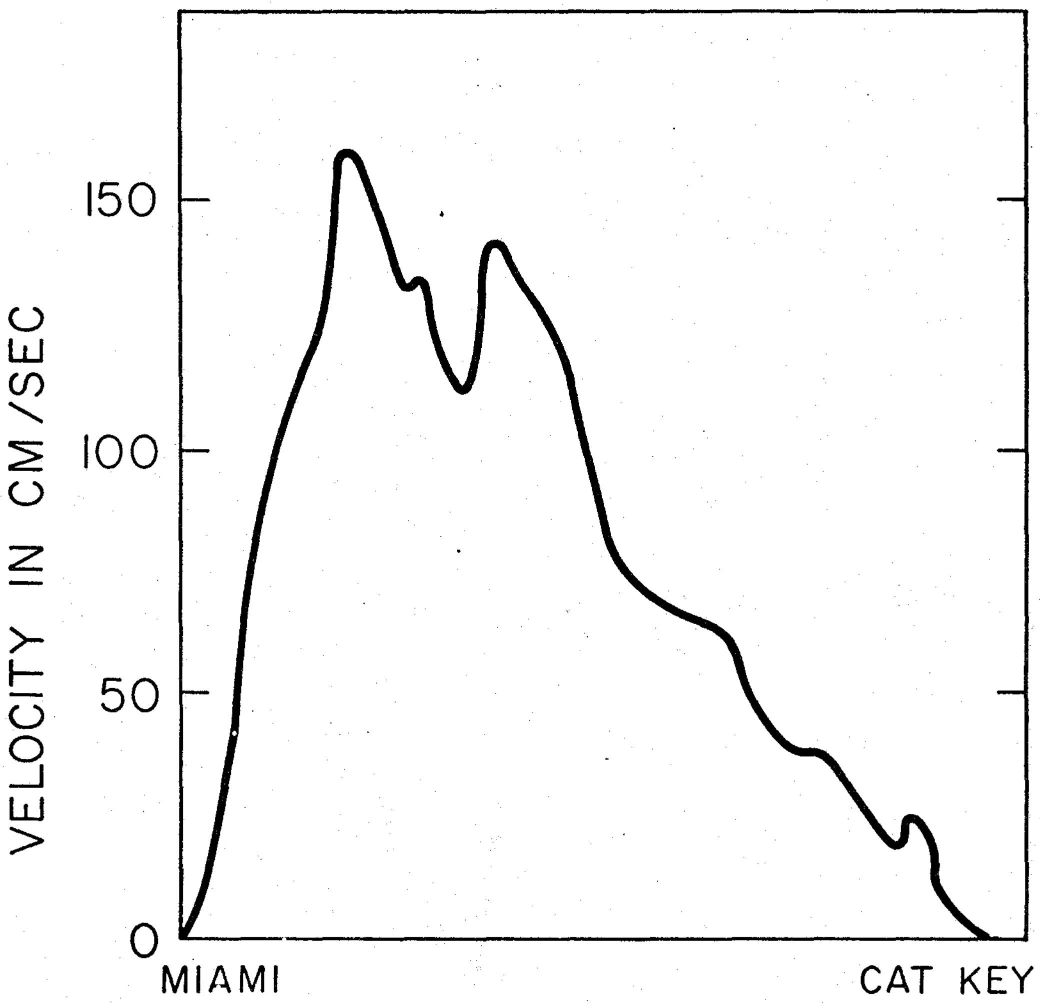


FIGURE 2. Average Surface Velocity of the Florida Current (Murry, 1952)

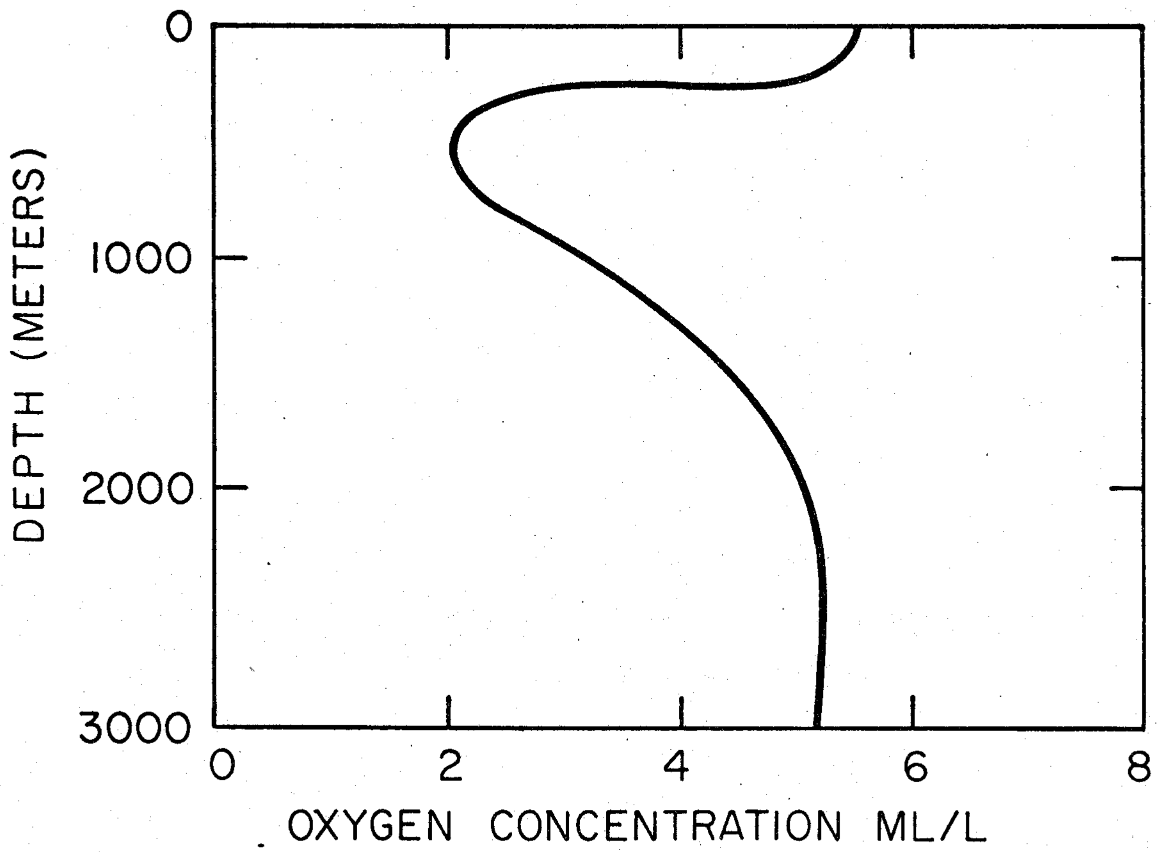


FIGURE 3. Characteristic Oxygen Profile of the Atlantic (Stommel, 1963)

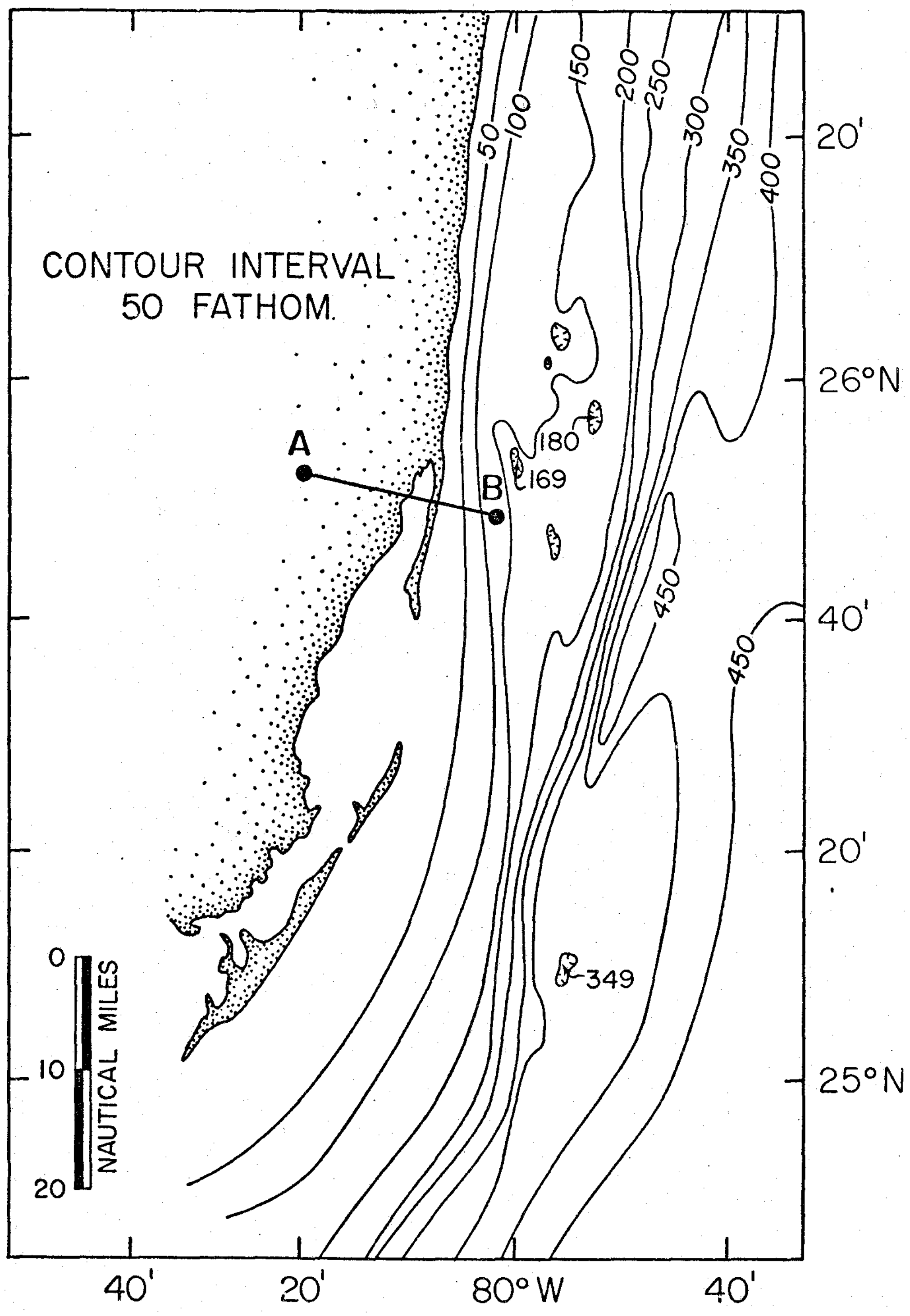


FIGURE 4. Bathymetry of Miami Terrace
(Koford and Malloy, 1965)

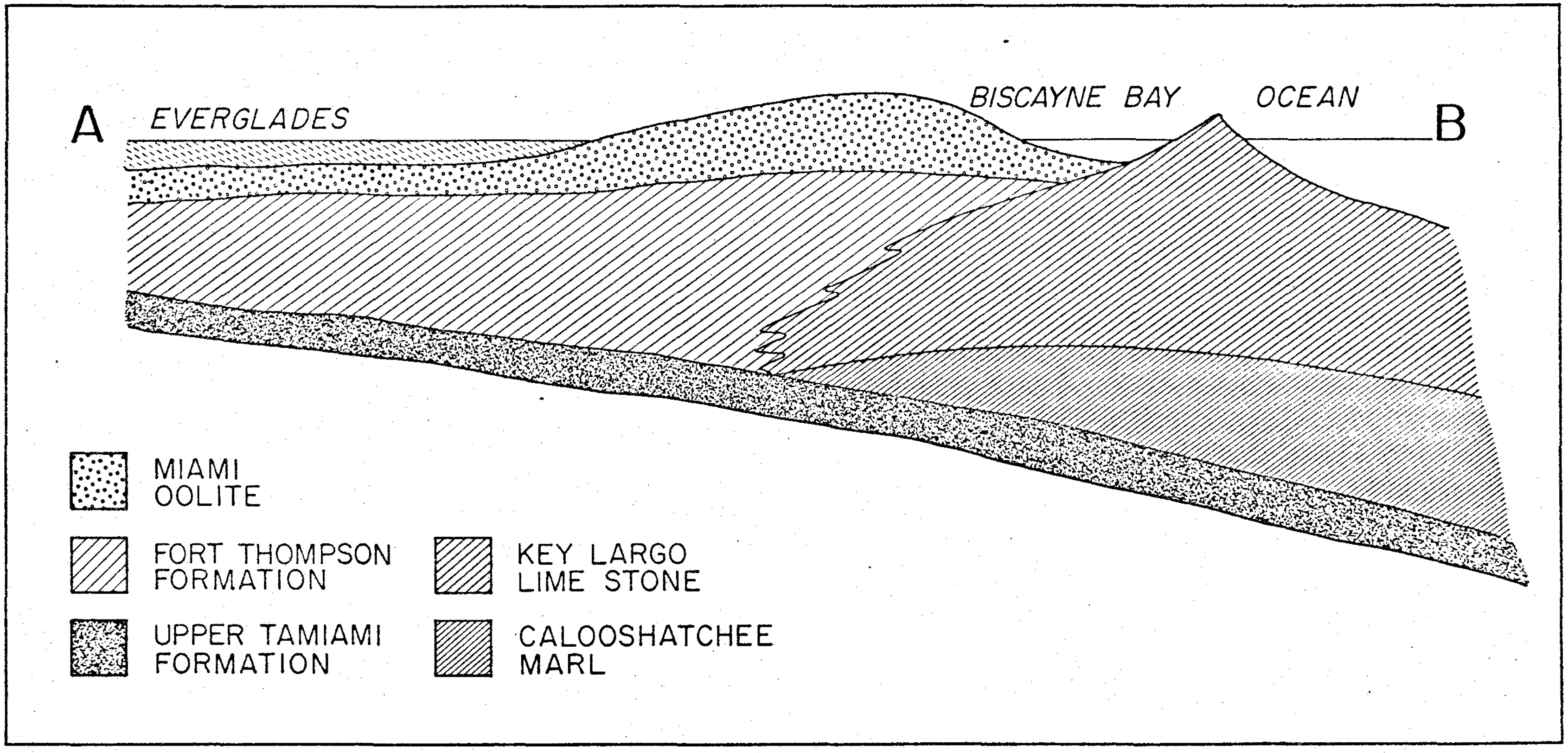


FIGURE 5. Biscayne Aquifer - Dade County Florida
Generalized Geologic Cross-Section (Runnels, 1971)

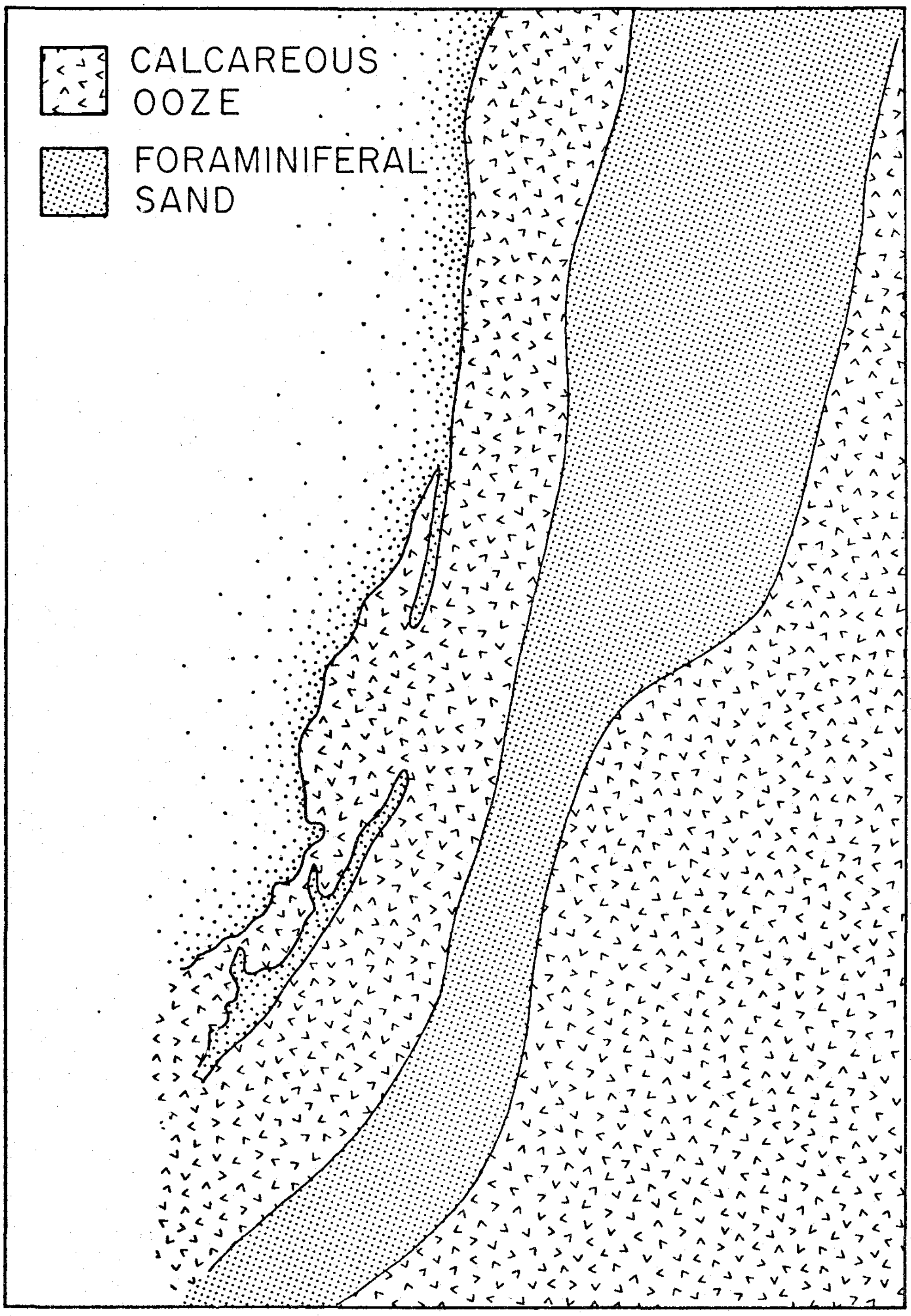


FIGURE 6. Sediment Distribution Near Miami (Pratt, 1971)

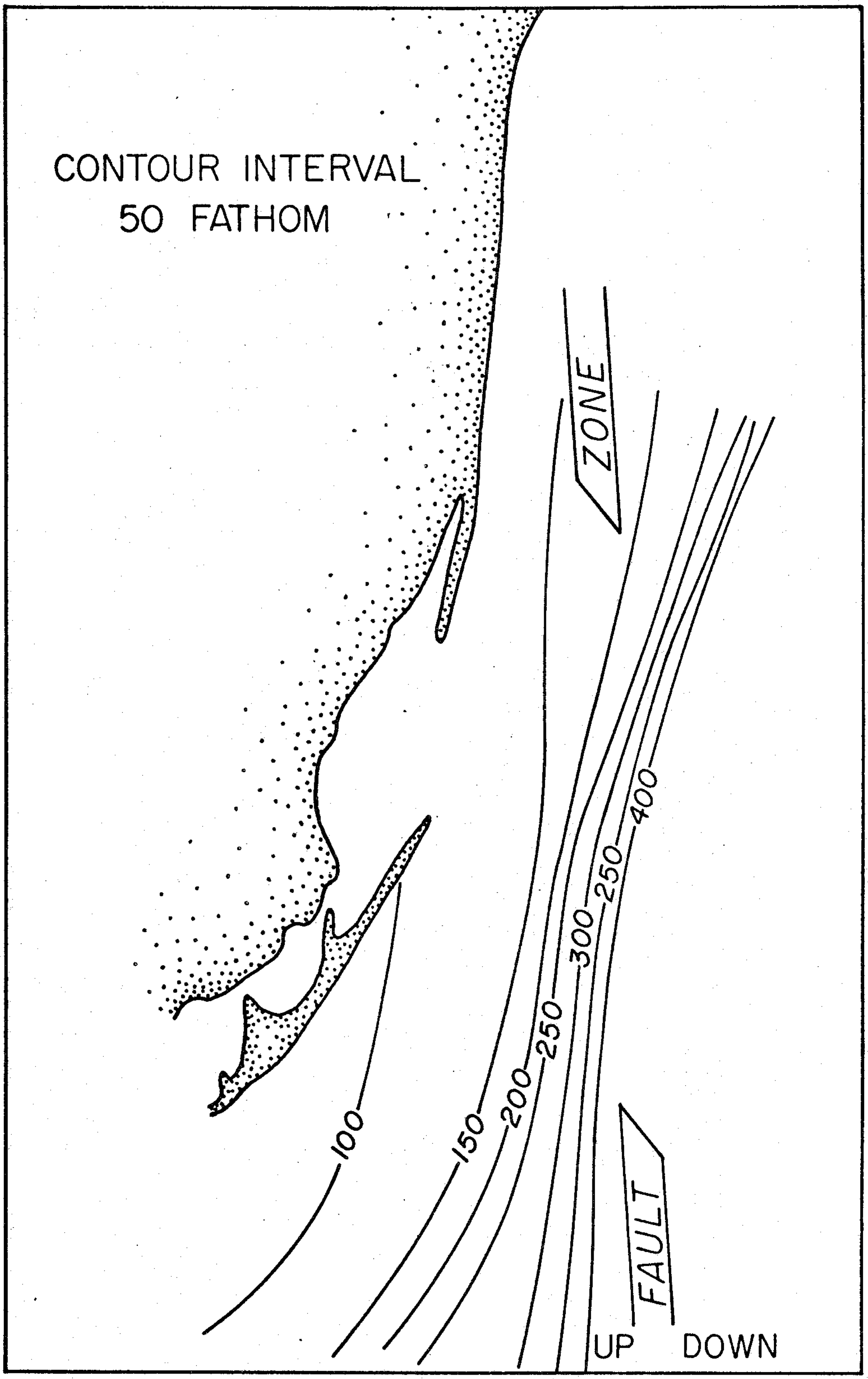


FIGURE 7. Bed Rock Contours and Fault Trace
(Kofoed and Malloy, 1965)

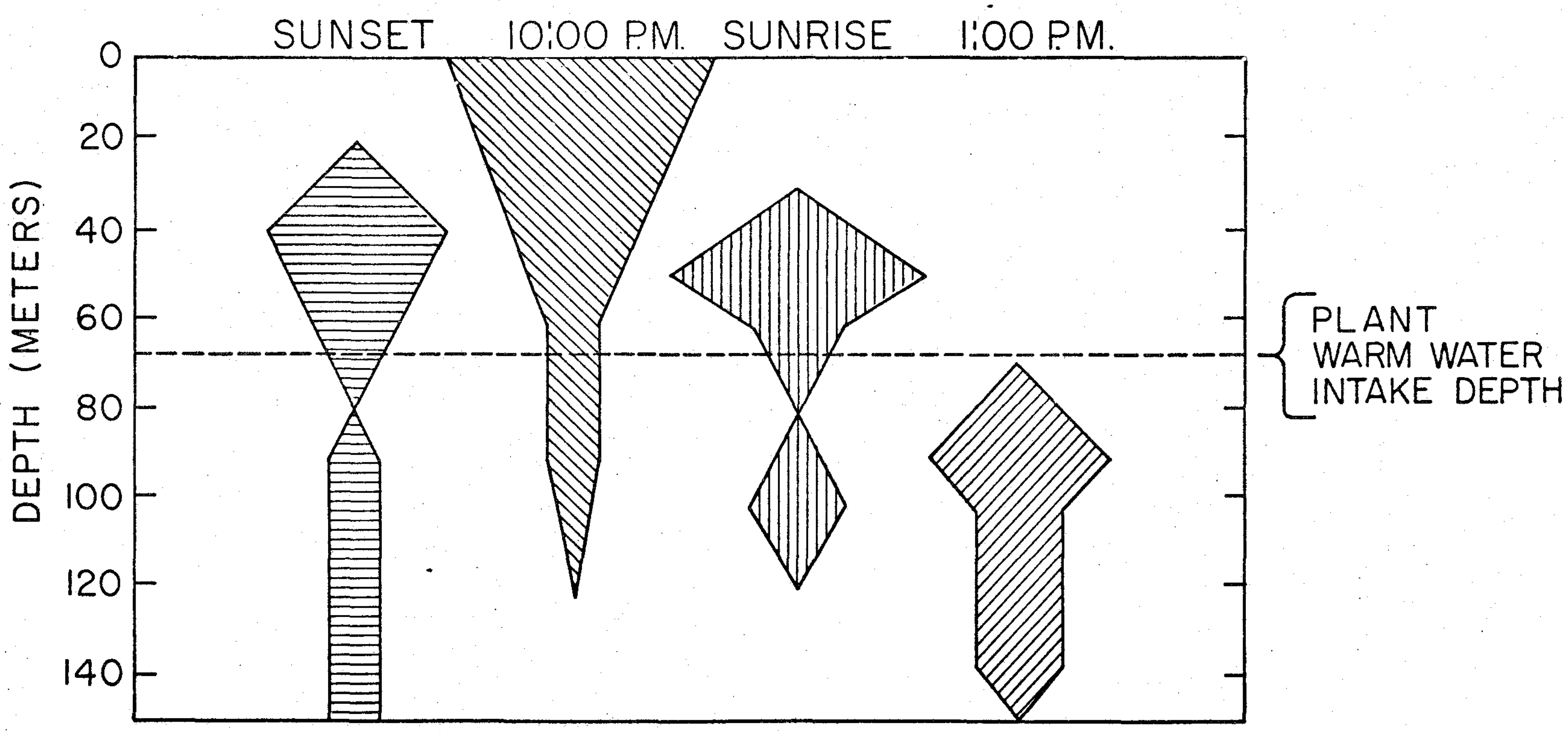


FIGURE 8. Diurnal Migration of Adult Female Calanus Finmachiuus (Sverdrup, 1942)

TABLE 2. GLOMAR CHALLENGER SUB-BOTTOM GEOLOGY DATA

Site: Hole 48

Location: Just south of Miami (25° 22.95' N, 77° 18.68' W)

Sub-bottom Characterization:	0-120 M:	Foram nanno ooze which spans Holocene-Pleistocene to the Oligocene period. Baleocene reflector at 280 M.

Penetrometer Readings:

Depth	Penetrometer Reading
5 M	9 M
15 M	12 M
20 M	6 M
225 M	6-7 M

Site: Hole 4

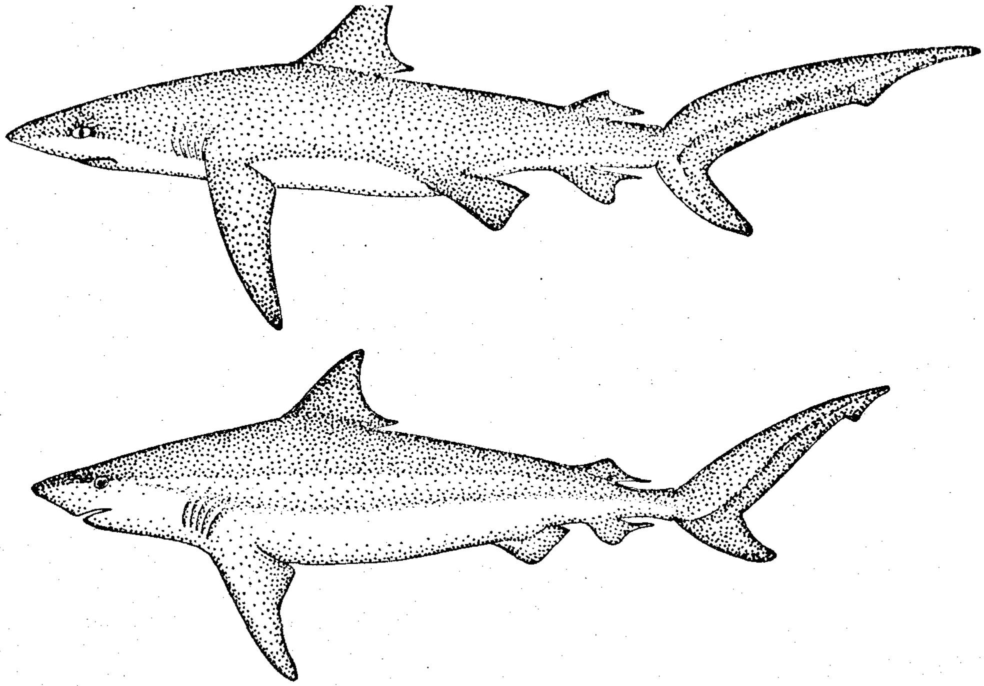
Location: Farther south and east (24.5° N, 73.8° W)

Sub-bottom Characterization:	0-10 M:	Calcarenite foram overlying clay
	10-75 M:	Calcarenite largely foram and nanno planton marl
	75-130 M:	Pebbly mud stone with limestone clasts Matrix of clay and nanno marl Some interbedded with nanno chalk
	130-190 M:	Nanno marl and chalk with chert
	190-250 M:	Nanno chalk

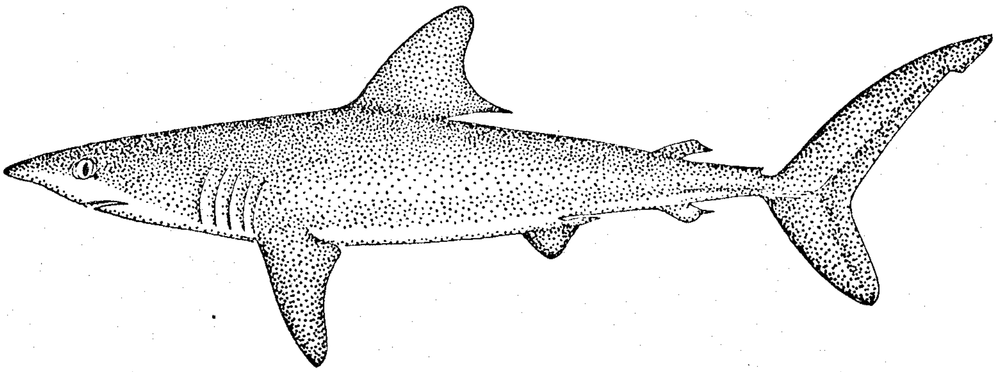
TABLE 3. LARGE DEEP-SEA ANIMALS BELOW 200 M (From: Heezen and Hollister, 1971)

Sessile
- Coelenterates
  - Sponges
  - Gorgonians
  - Pennatulids
  - Actinarians
  - Antipatharians

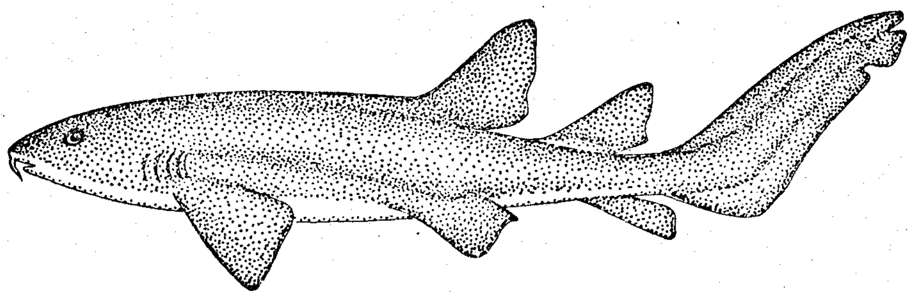
Mobile
- Echinoderms
  - Crinoids
  - Astezoids
  - Ophiuroids
  - Echinoids
  - Holothurians
- Molluscs
  - Cephopods
- Annelids
  - Polychaets
- Arthropods
  - Isopods
  - Decapods
  - Arachnoids
- Chordates
  - Hemichorates
  - Tunicates
  - Fish



REQUIEM SHARKS

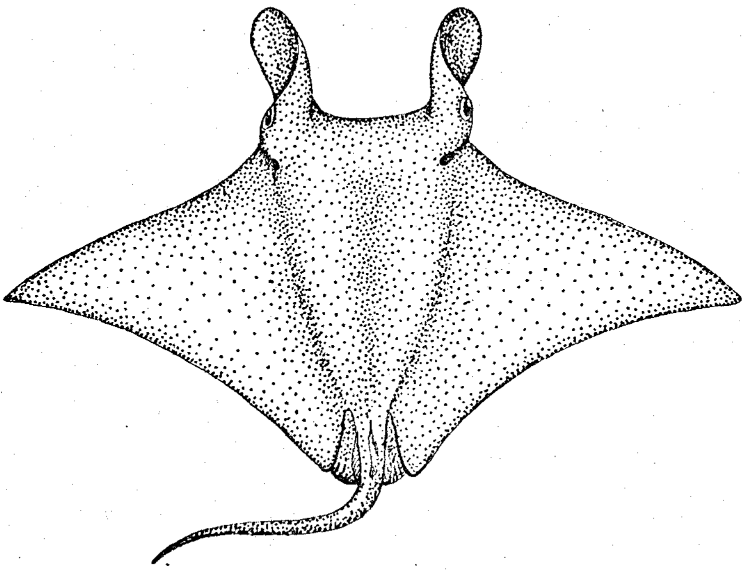


MACKEREL SHARKS

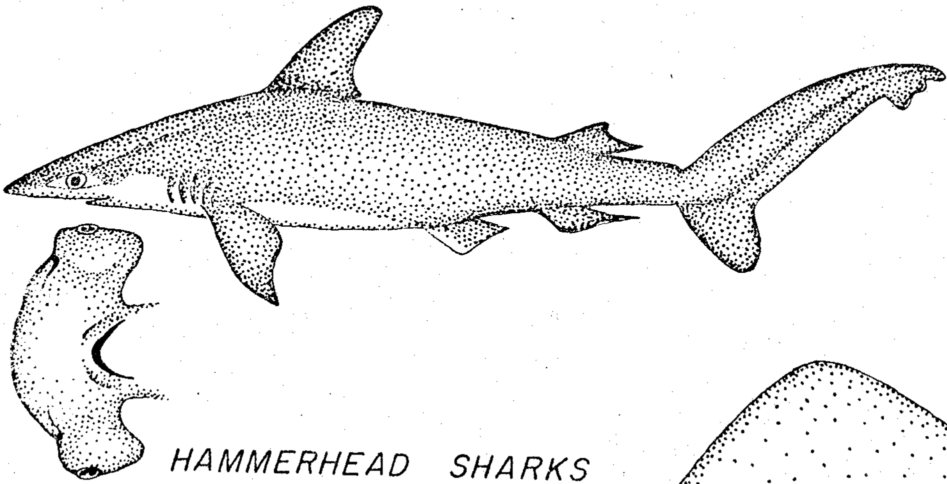


NURSE SHARKS

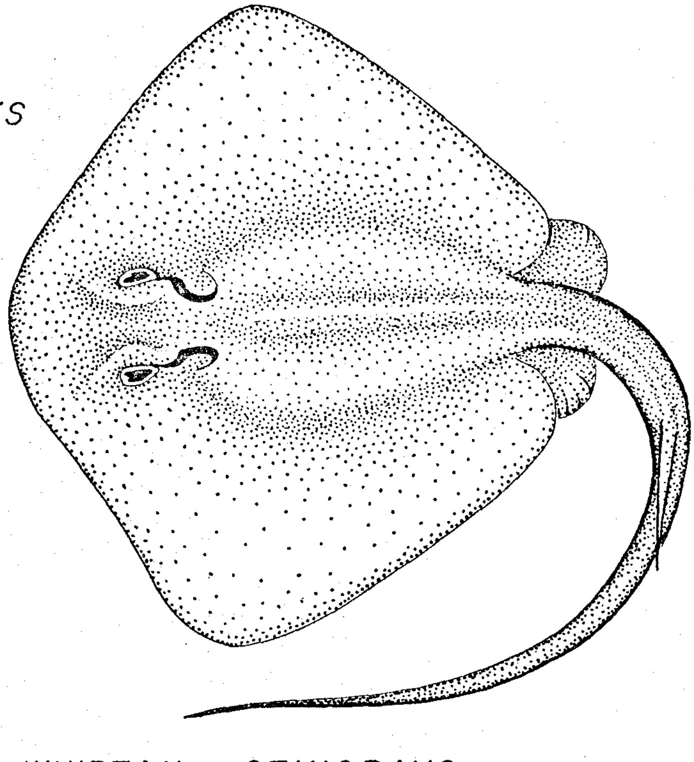
PLATE 1. Primary Area Sharks



MANTAS

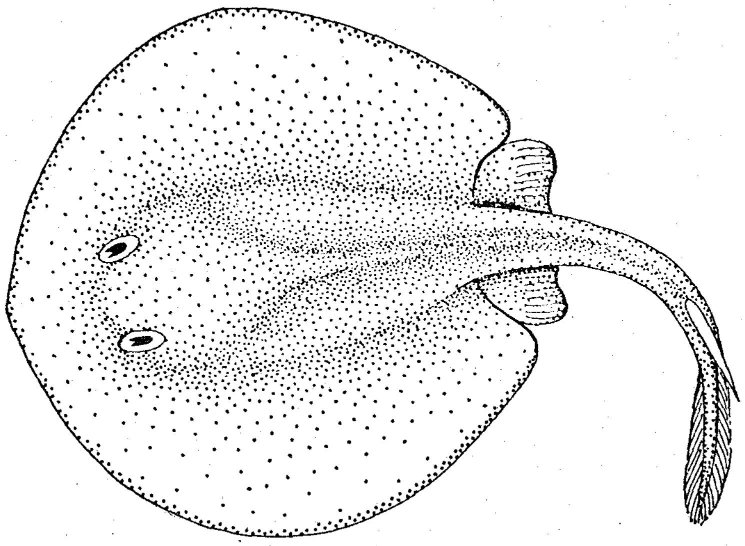


HAMMERHEAD SHARKS

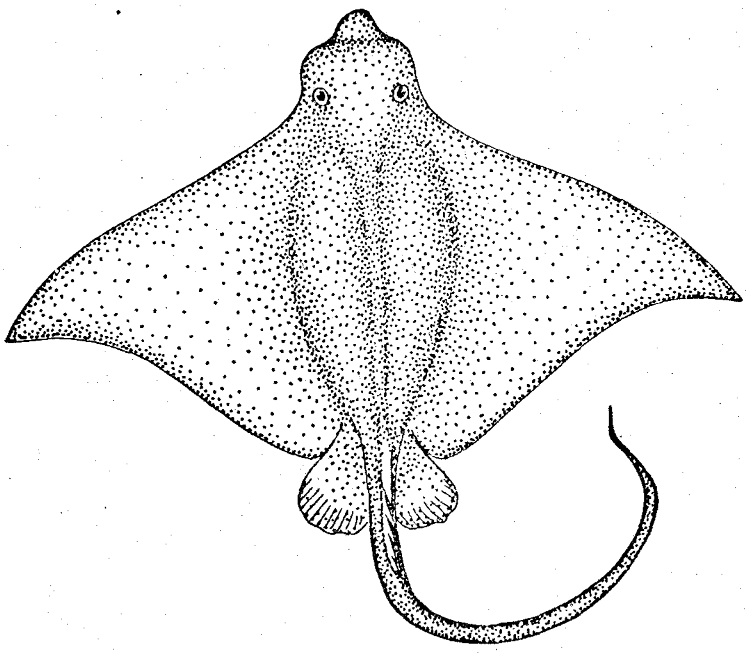


WHIPTAIL STINGRAYS

PLATE 1. Continued

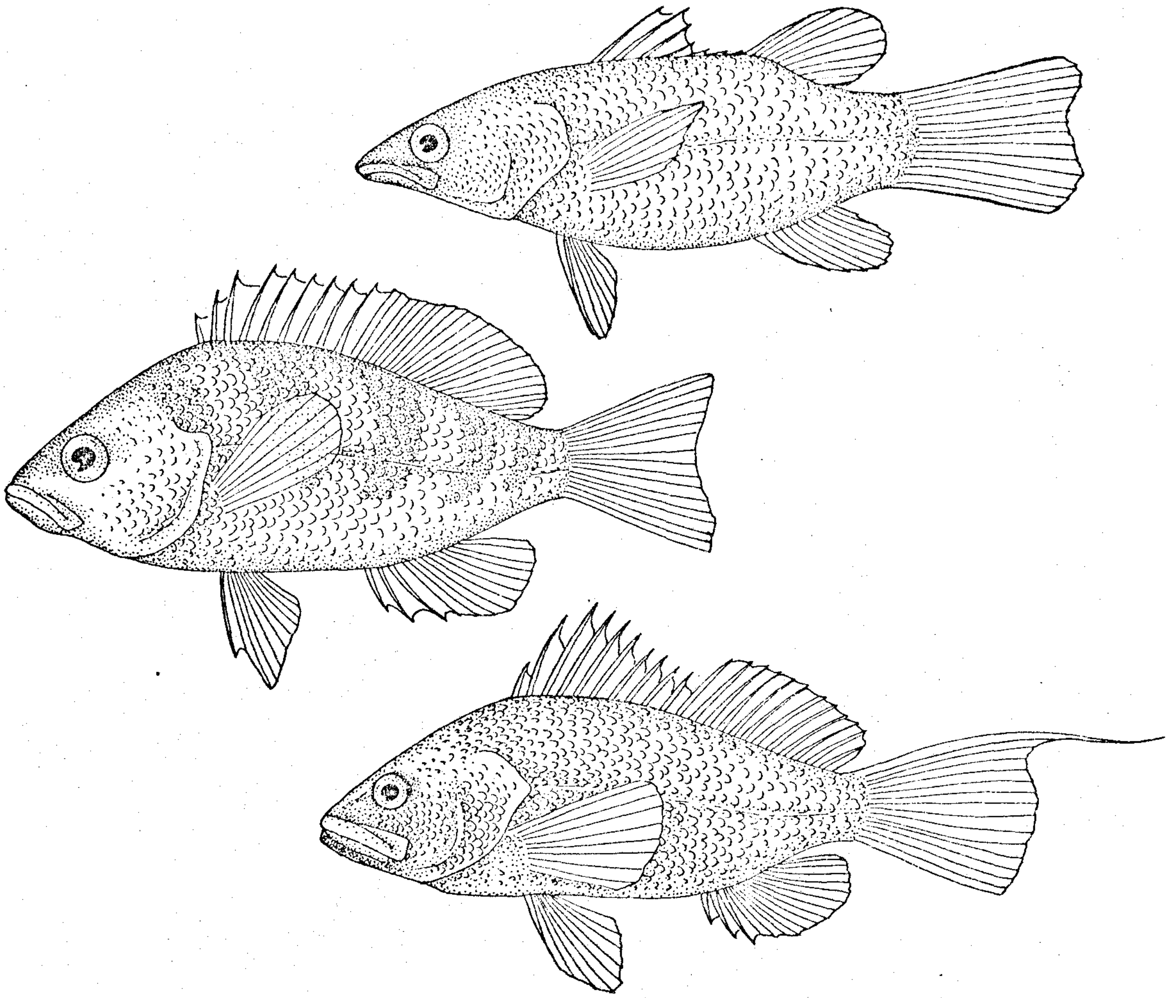


FINTAIL STINGRAYS

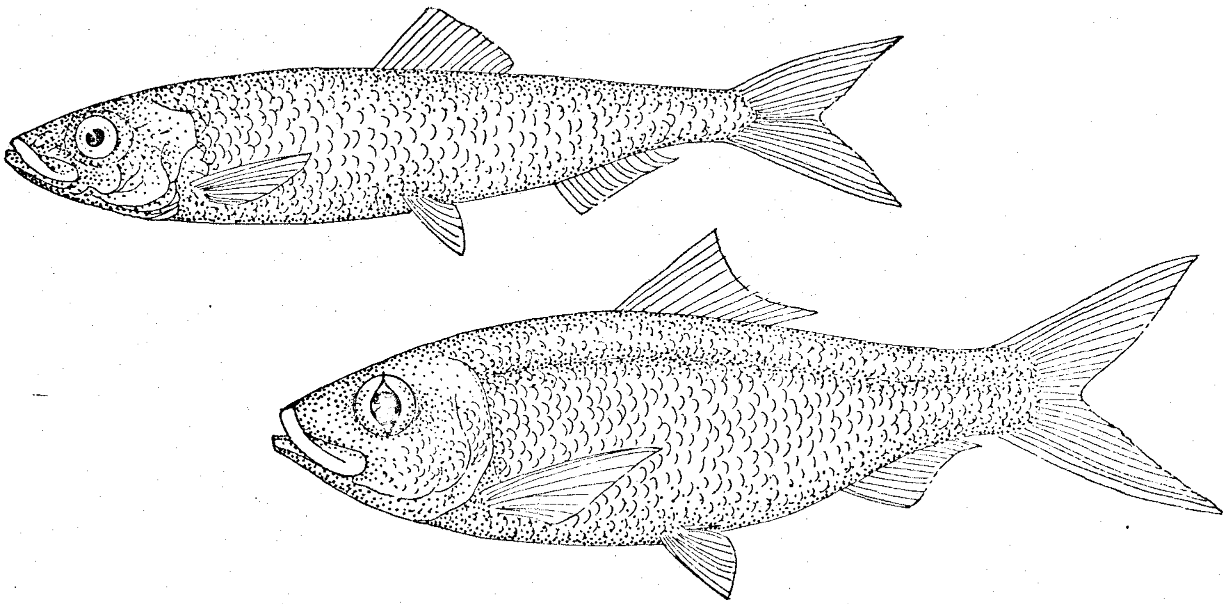


EAGLE RAYS

PLATE 1. Continued

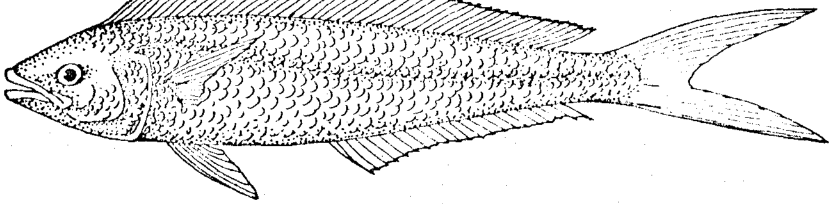


SEA BASSES

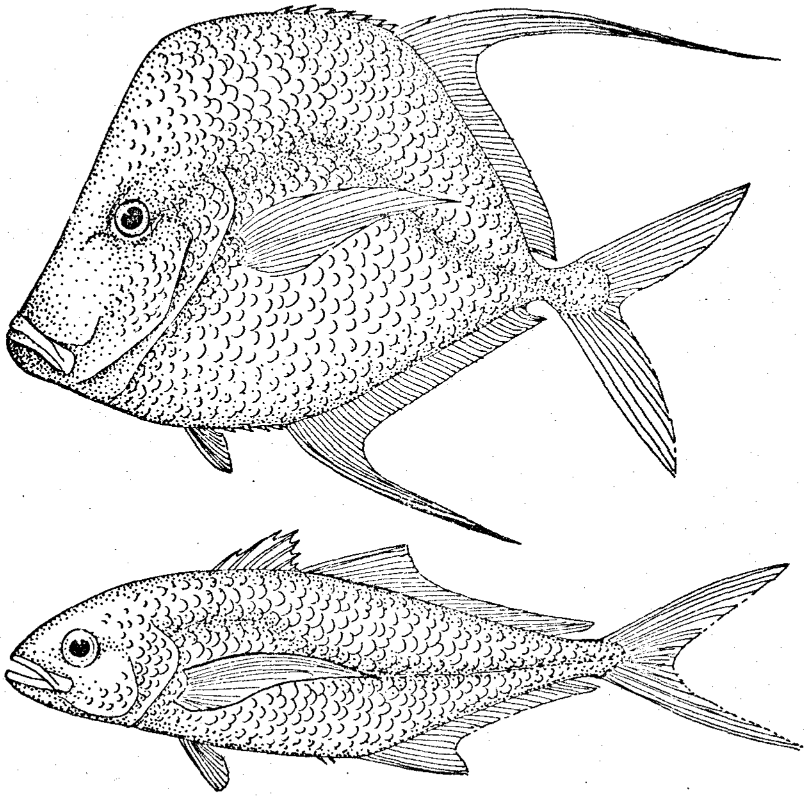


HERRINGS

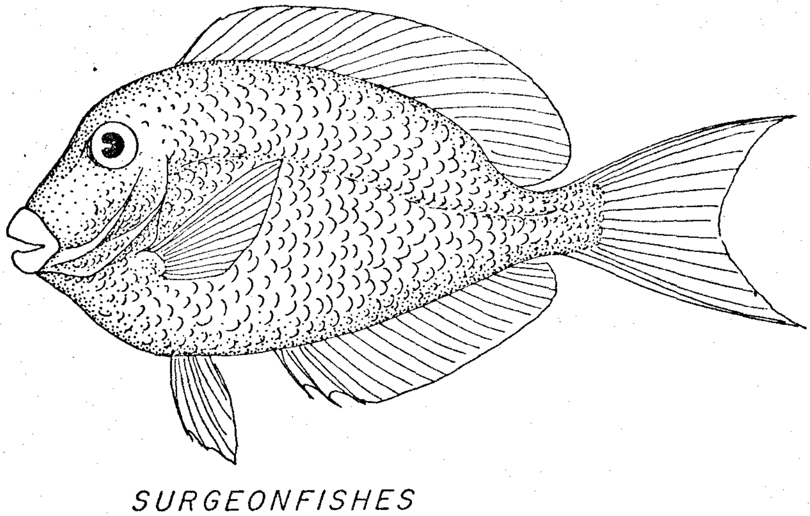
PLATE 2. Primary Game and Commercial Species



DOLPHINS

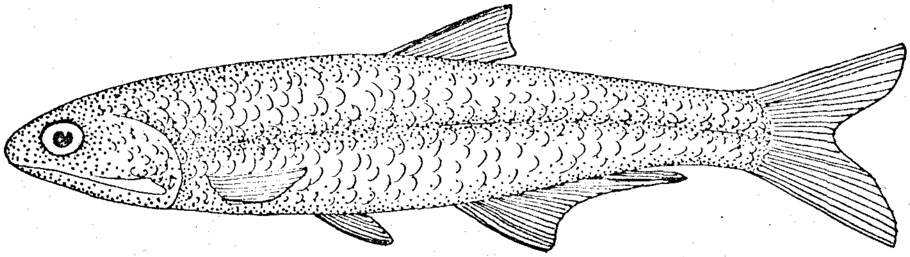


JACKS, SCADS & POMPANOS

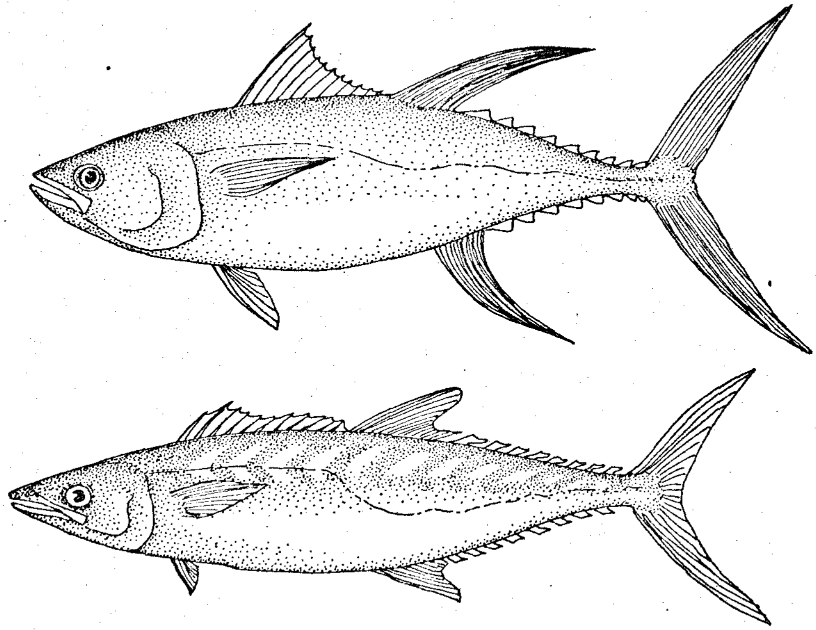


SURGEONFISHES

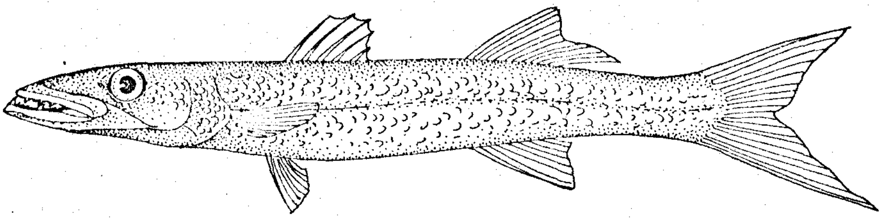
PLATE 2. Continued



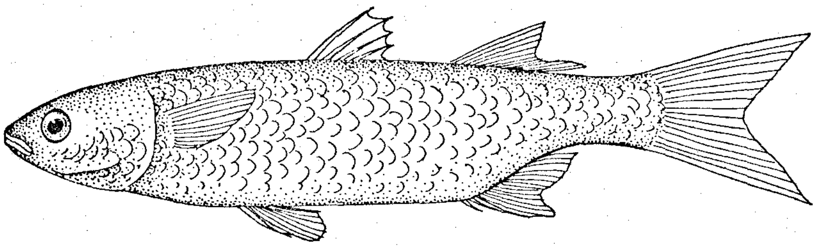
ANCHOVIES



TUNAS AND MACKERELS

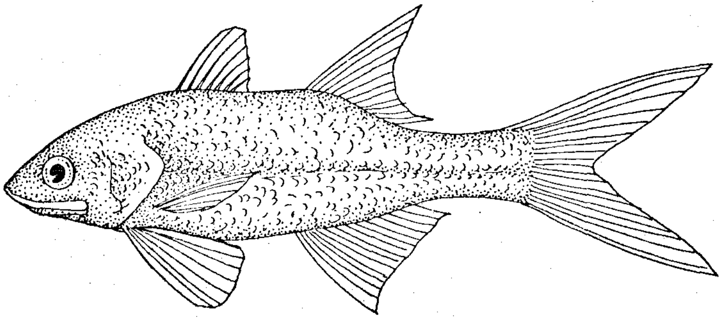


BARRACUDAS

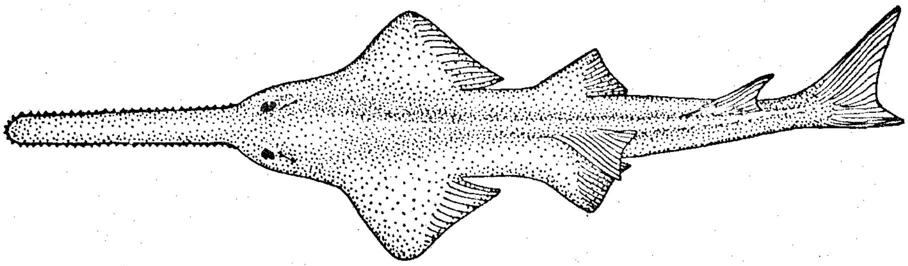


MULLETS

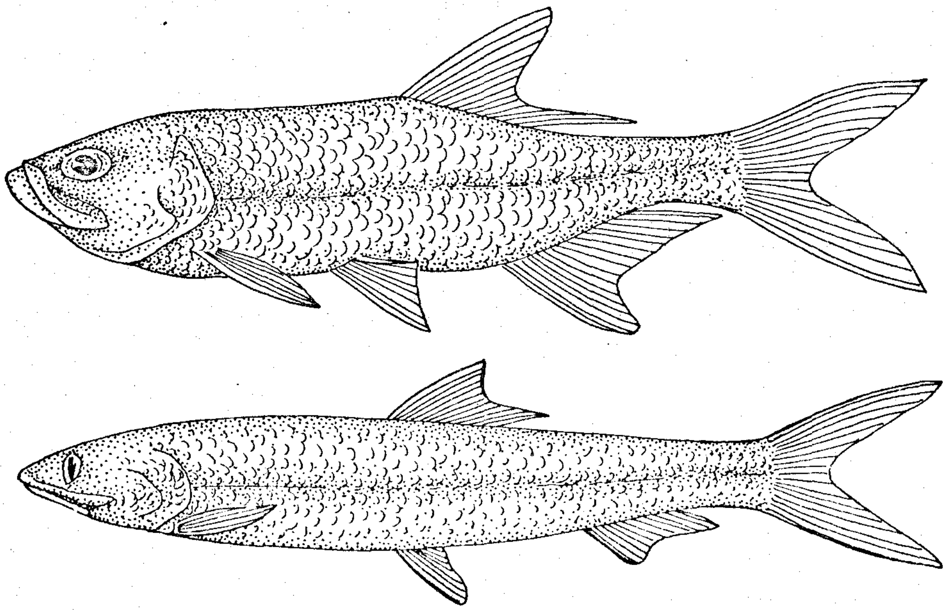
PLATE 2. Continued



THREADFINS

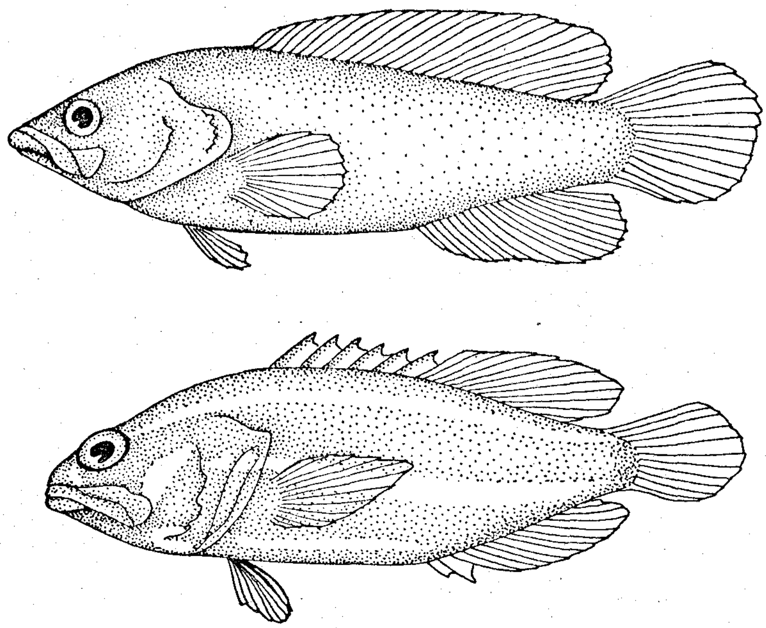


SAWFISHES

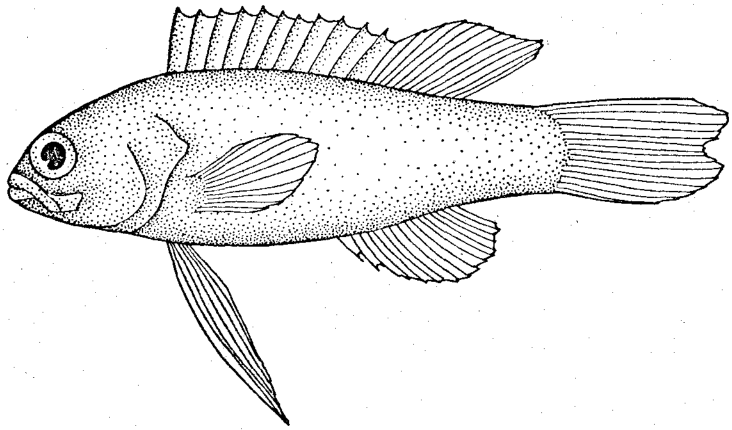


TARPONS & LADYFISHES

PLATE 3. Primary Game and Commercial Species

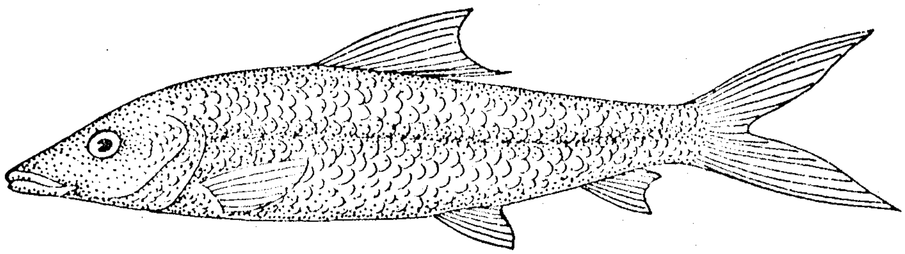


SOAPFISHES AND ALLIES

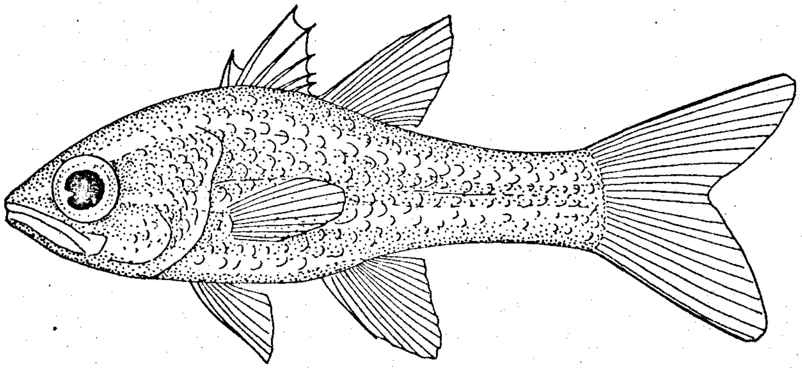


BASSLETS

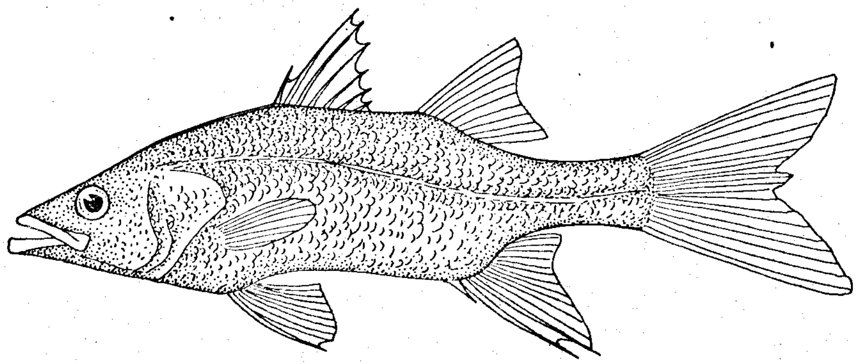
PLATE 3. Continued



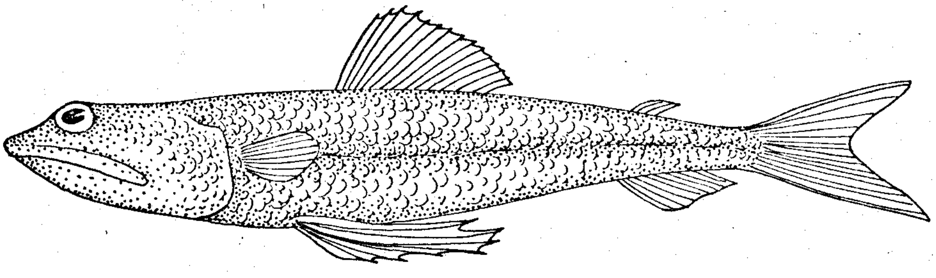
BONEFISHES



CARDINALFISHES

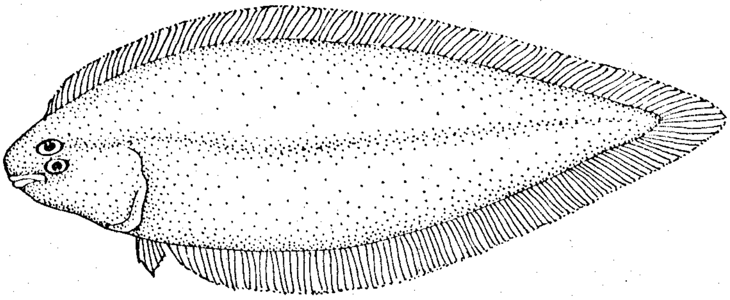


SNOOKS OR ROBALOS

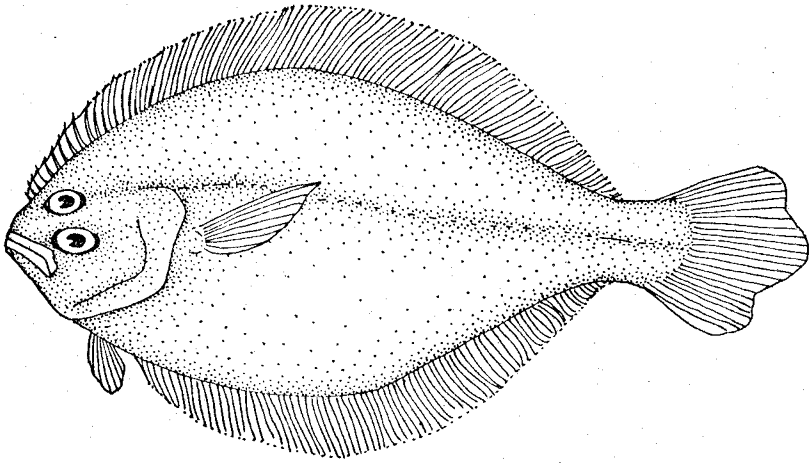


LIZARDFISHES

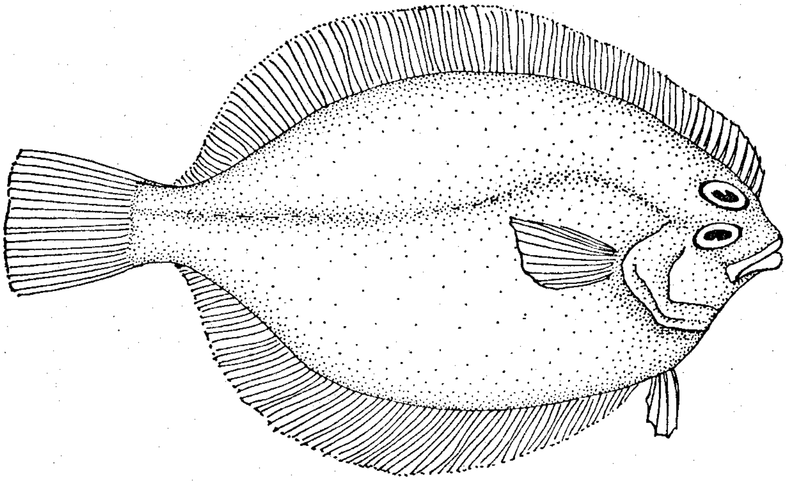
PLATE 3. Continued



TONGUEFISHES

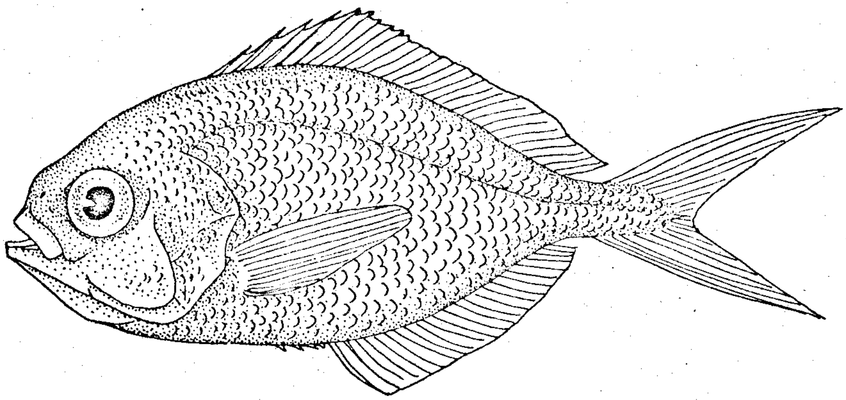


LEFTEYE FLOUNDERS

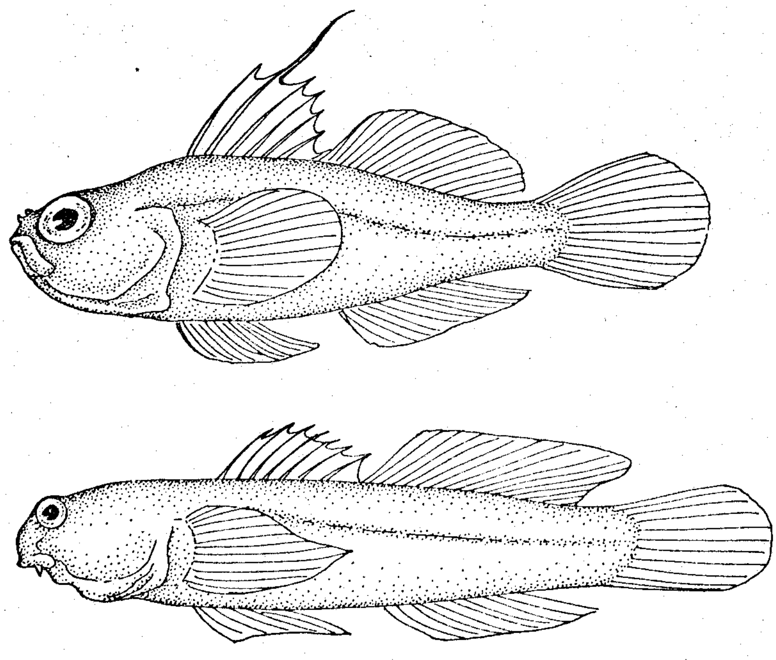


SOLES

PLATE 4. Primary Bottom Fish

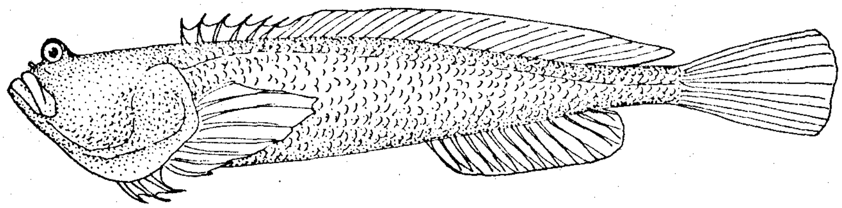


BUTTERFISHES

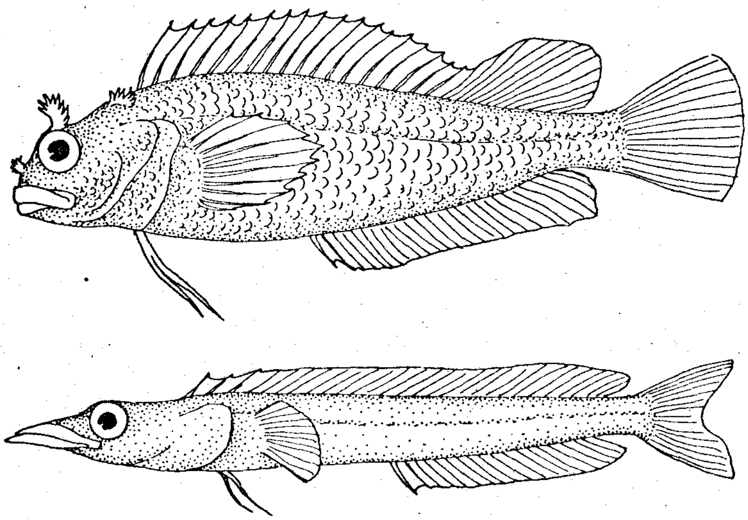


GOBIES AND SLEEPERS

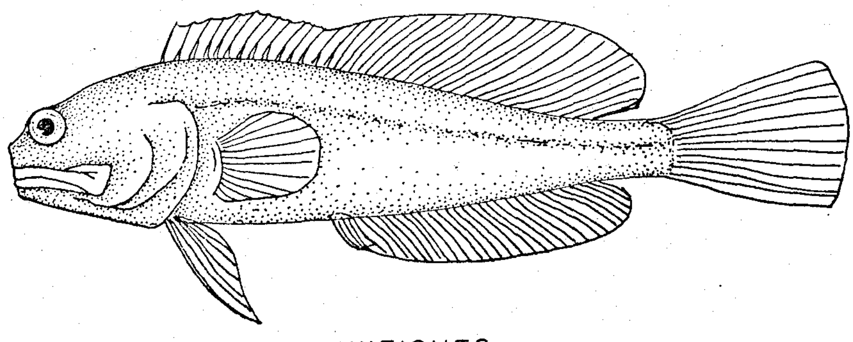
PLATE 5. Minor Gamefish



SAND STARGAZERS

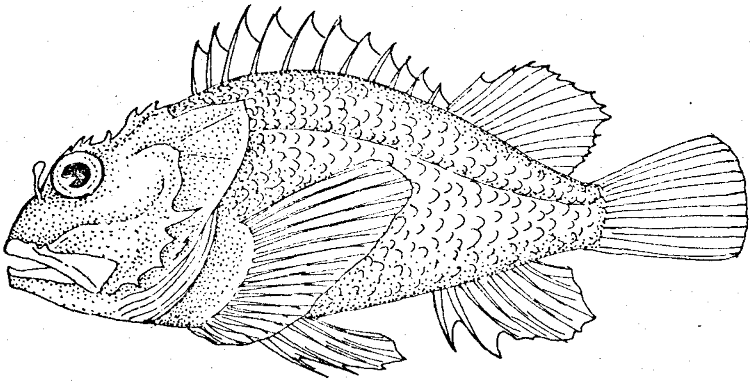


CLINIDS

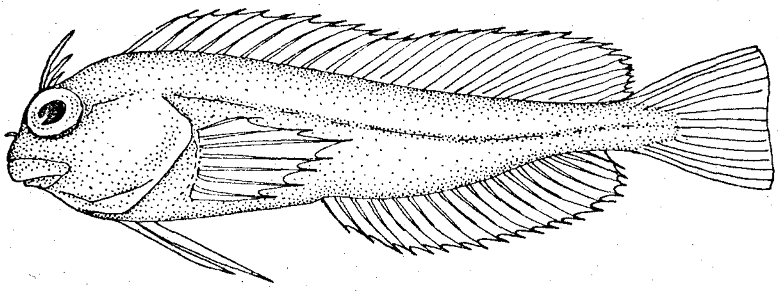


JAWFISHES

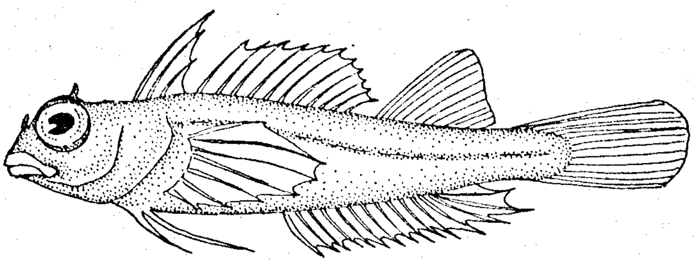
PLATE 5. Continued



SCORPIONFISHES

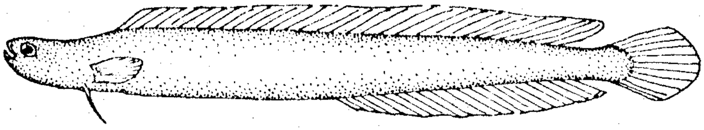


COMBTOOTH BLENNIES

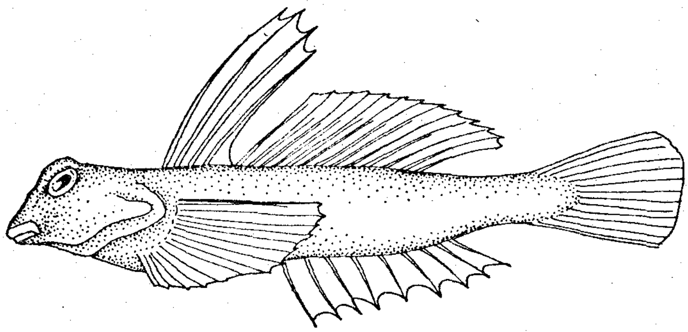


THREEFIN BLENNIES

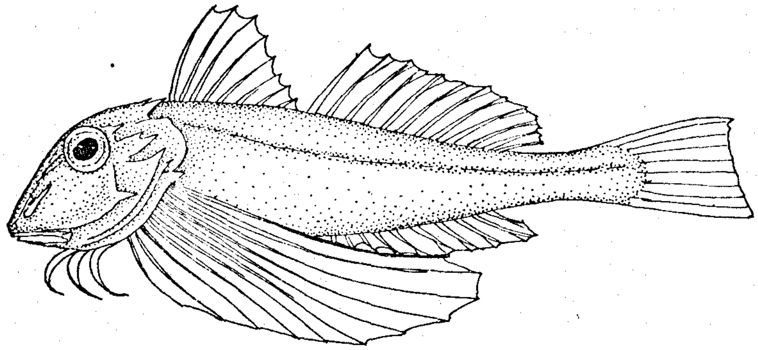
PLATE 5. Continued



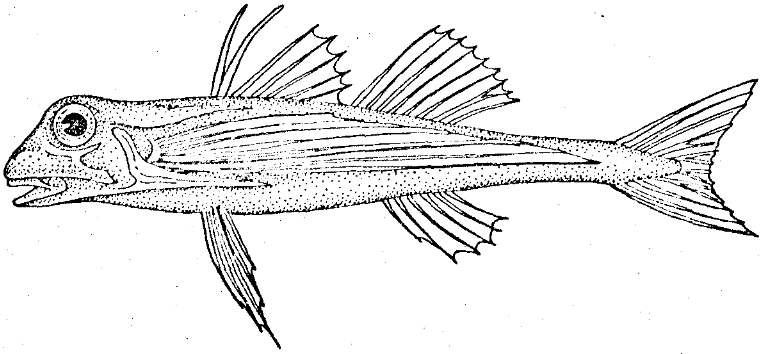
WORMFISHES



DRAGONETS

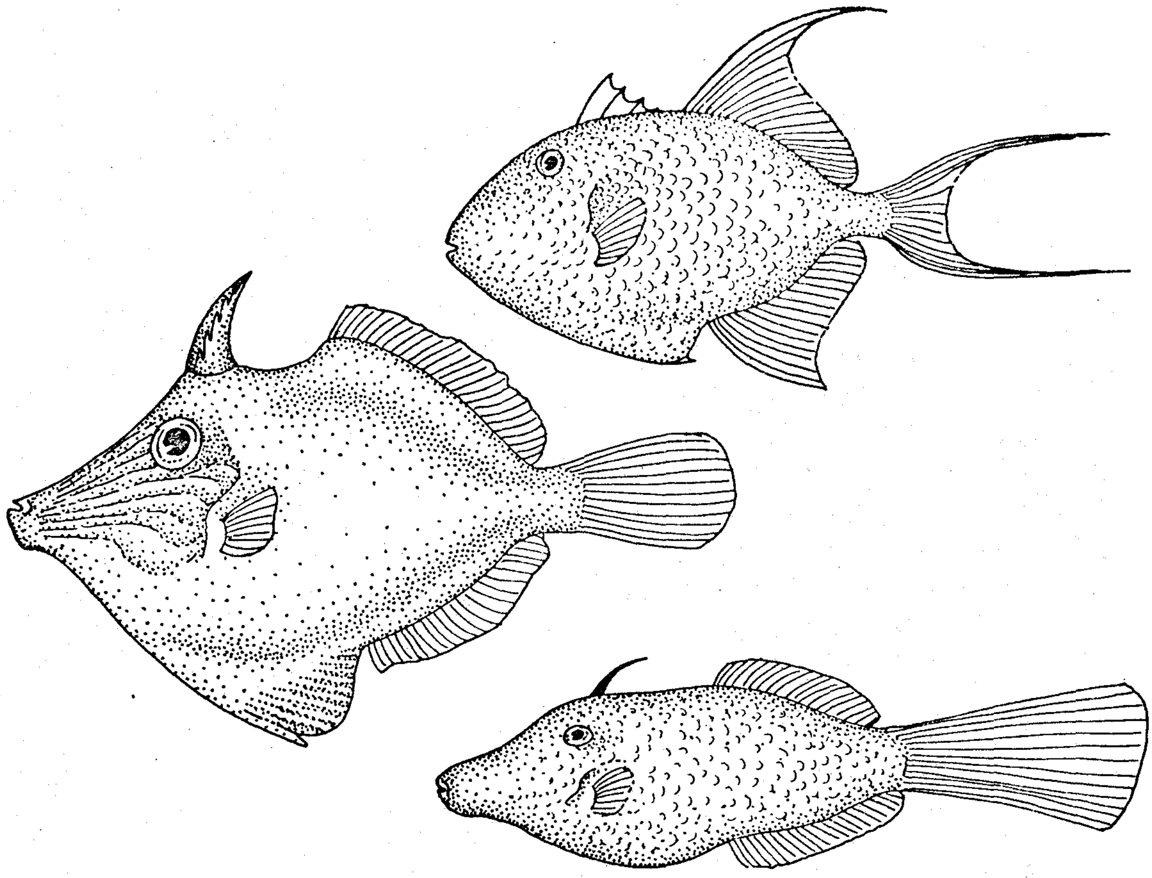


SEAROBINS

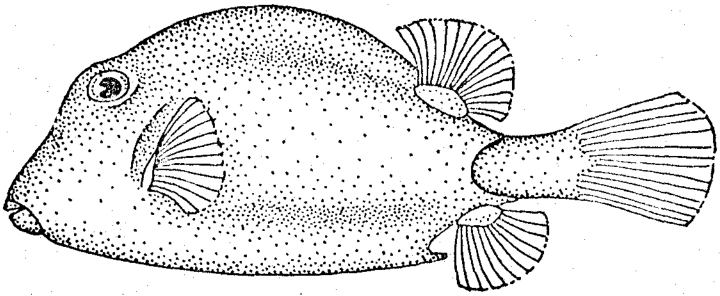


FLYING GURNARDS

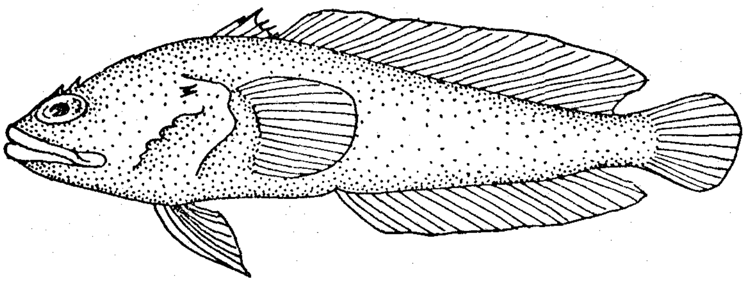
PLATE 5. Continued



TRIGGERFISHES & FILEFISHES

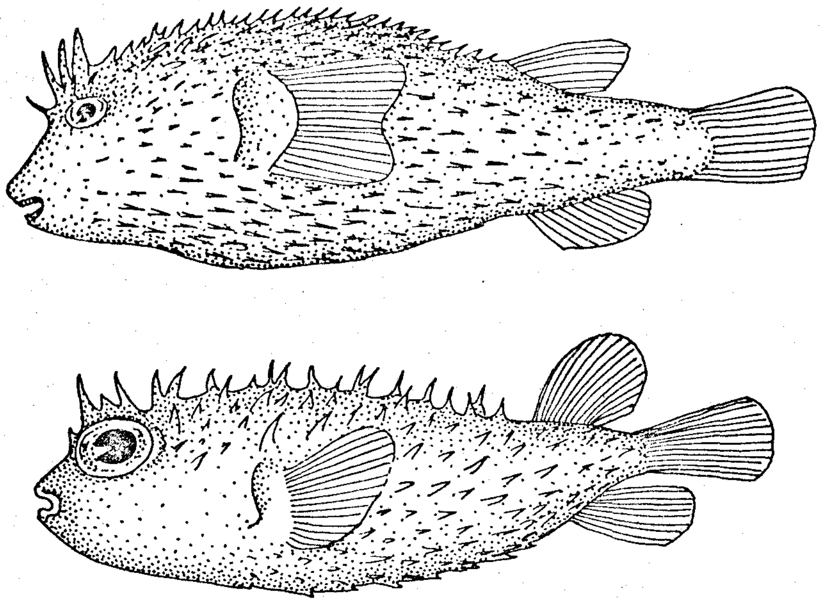


TRUNKFISHES

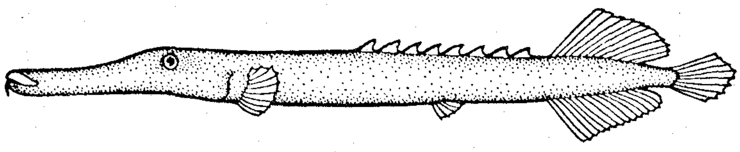


TOADFISHES

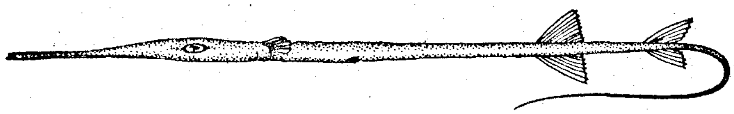
PLATE 6. Minor Area Species



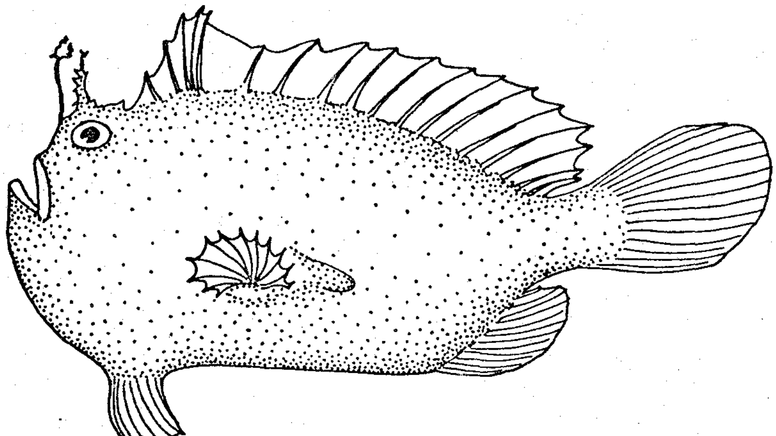
PORCUPINEFISHES



TRUMPET FISHES



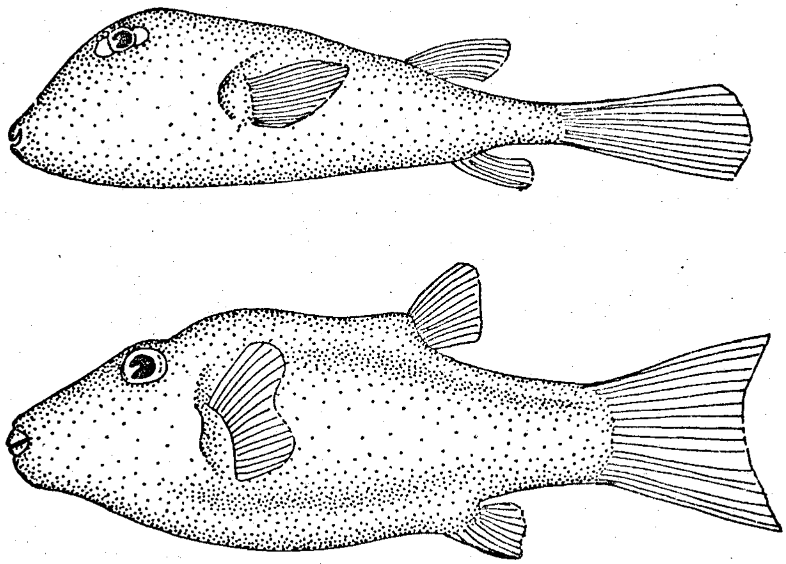
CORNET FISHES



FROGFISHES

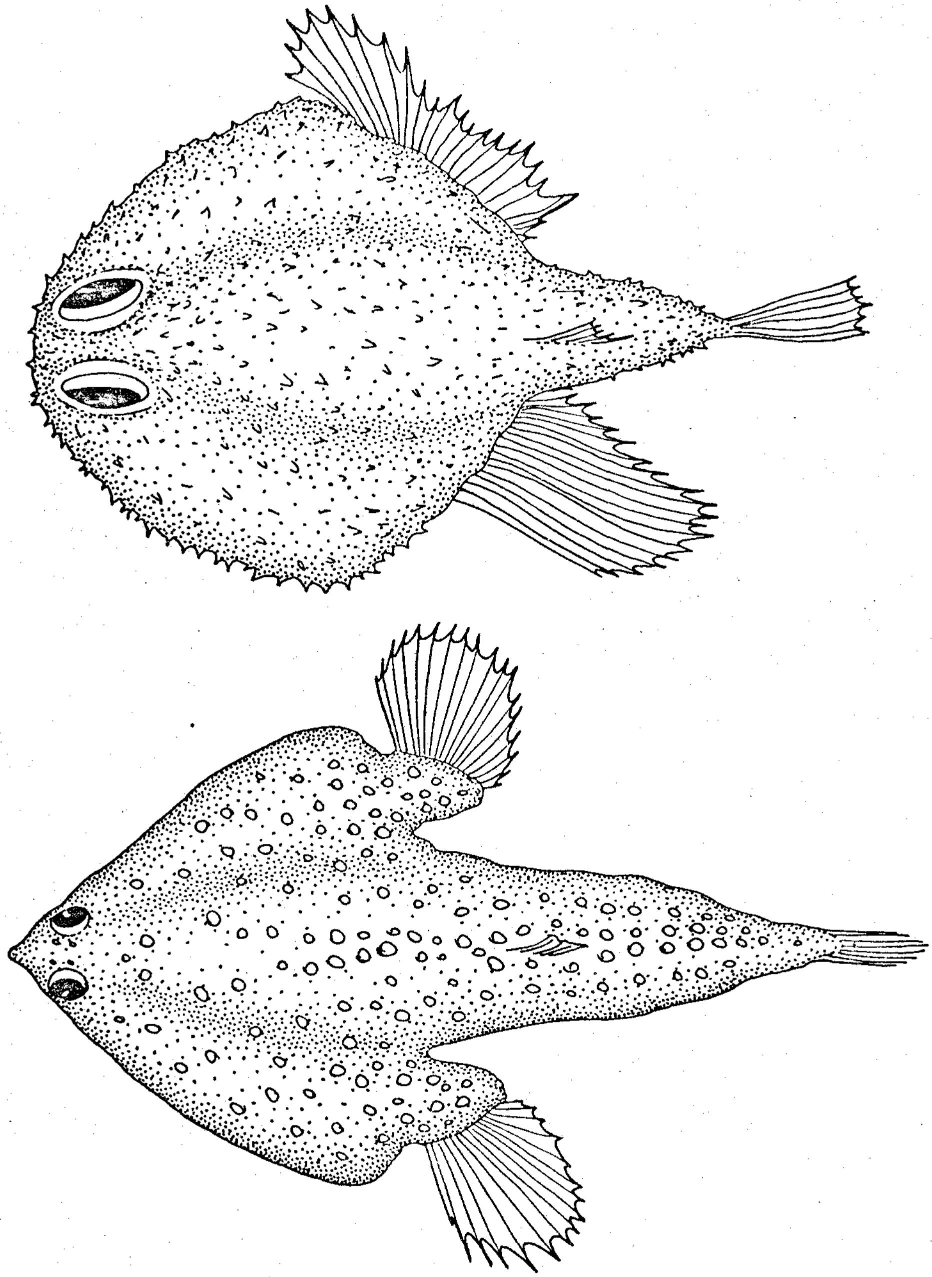
PLATE 6. Continued

SEAHORSES AND PIPEFISHES

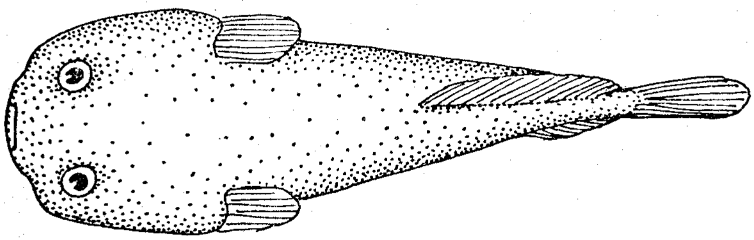


PUFFERS

PLATE 6. Continued



BATFISHES



CLINGFISHES

PLATE 6. Continued

TABLE 4. FLORIDA LANDINGS IN DADE COUNTY 1970 (Florida Dept. of Natural Resources, 1970)

Species: Fish	Pounds
Ballyhoo (Halfbeak)	142,856
Blue fish	5,988
Blue Runner	11,259
Cabio	225
Crevalle (Jacks)	6,095
Croaker	5,236
Dolphin	4,205
Drum (Black)	587
(Red)	1,113
Groupers	70,745
Grunts	39,314
Hog fish	2,393
Jew fish	696
King Mackeral	51,237
Mullet (Black)	50
(Silver)	211,846
Permit	18
Pompano	12,585
Scup	2,408
Sea Trout (Spotted)	8,282
Sharks	3,250
Sheepshead	1,480
Snapper (Lane)	977
(Mangrove)	26,107
(Mutton)	42,410
(Red)	108,169
(Vermilion)	111
(Yellow tail)	153,902
Spanish Mackeral	327,164
Unclassified	21,882
	Total 1,262,590

TABLE 5. FLORIDA LANDINGS IN DADE COUNTY 1970 (Florida Dept. of Natural Resources, 1970)

Species: Shellfish, etc.		Pounds
Conchs		30
Crabs (Blue, hard)		12,570
(Stone)		109,751
Lobster (Spiny)		2,766,858
Turtles (Green)		744
(Loggerhead)		469
Sponges (Grass)		4,533
(Sheepswool)		4,505
(Yellow)		2,549
	Total	2,902,009

TABLE 6. PHYTOPLANKTON SPECIES COMMON TO THE GULF OF GUINEA, THE WESTERN TROPICAL ATLANTIC, CARIBBEAN SEA AND STRAITS OF FLORIDA (Wood, 1968)

DIATOMS

Actinocyclus ovatus
Asterolampra grevillei
A. marylandica
Asteromphalus brookei
A. flabellatus
A. heptactis
A. hookeri
A. roperianus
Bacteriastrum delicatulum
B. varians
Biddulphia chinensis
Cerataulina pelagica
Chaetoceros affine
Ch. coarctatum
Ch. concavicorne
Ch. decipiens
Ch. denticulatum
Ch. laeve
Ch. lorenzianum
Ch. messanense
Ch. neapolitanum
Ch. pendulum
Ch. similis
Ch. socialis
Ch. vanheurckii
Climacodium frauenfeldianum
Corethron criophilum
Coscinodiscus africanus
C. concinnus
C. excentricus
C. lineatus
C. marginatus
C. radiatus
Dactyliosolen mediterraneus
Detonula confervacea
Eucampia cornuta
Fragilaria linearis
F. striatula
Gossleriella tropica
Guinardia flaccida
Hemiaulus hauckii
H. membranaceus
H. sinensis
Hemidiscus cunieformis
Leptocylindrus danicus
Mastogloia rostrata
Navicula acus
Nitzschia closterium
N. gracilis
N. longissima
N. seriata
Planktoniella formosa
P. sol
Pleurosigma capense
Pl. directum
Pl. hippocampus
Pl. naviculaceum
Pseudoeunotia doliolus
Rhizosolenia alata
R. bergonii
R. calcar avis
R. castracanei
R. cylindrus
R. delicatula
R. imbricata
R. setigera
R. stolterforthii
R. styliformis
Schroederella delicatula
Striatella interrupta
Synedra superba
S. ulna
Thalassiosira rotula
Thalassiothrix frauenfeldii
T. longissima
T. mediterranea
Tropidoneis affirmata

CYANOPHYCEAE

Richelia intracellularis
Trichodesmium thiebautii

DINOFLAGELLATES

Amphidinium klebsi
A. turbo
Amphisolenia bidentata
A. globifera
Blepharocysta splendormaris
Ceratium arietinum
C. azoricum
C. belone
C. buceros
C. candelabrum
C. carriense
C. extensum
C. falcatiforme
C. furca
C. fusus
C. gibberum
C. gravidum
C. kofoidi
C. massiliense
C. minutum
C. pentagonum
C. praelongum
C. pulchellum
C. ranipes
C. setaceum
C. symmetricum
C. teres
C. trichoceros
C. tripos
Ceratocorys armata
C. horrida
Cochlodinium faurei
Dinophysis exigua
D. ovum
Dipeopsalis lenticula
Erythropsis sp.
Exuviaella baltica
Goniaulax birostris
G. diegensis
G. kofoidi
G. monacantha
Goniodoma polyedricum
Gymnodinium flavum
G. gelbum
G. marinum
G. mirabile
G. simplex
Murrayella biconica
Ornithocercus magnificus
O. quadratus
O. steini
O. thurni
Oxytoxum belgicae
O. constrictum
O. milneri
O. scolopax
O. tesselatum
O. turbo
Parahistioneis para
Peridinium brochi
P. cerasus
P. depressum
P. divergens
P. globulus
P. grande
P. grani
P. pendunculatum
P. steini
P. subinerme
Phalacroma cuneus
Ph. dolichopterygium
Ph. doryphorum

TABLE 6. cont.

Ph. hindmarchi
Ph. mitra
Ph. ovum
Ph. parvulum
Ph. rotundatum
Podolampas palmipes
Pod. elegans
Pod. bipes
Porella perforata
Prorocentrum dentatum
Pr. micans
Pr. scutellum
Protoceratium reticulatum
Pryocystis fusiformis
Py. lunula
Py. pseudonoctiluca
Pyrophacus horologicum
Warnowia atra

OTHER FLAGELLATES

Coccolithus huxleyi
Michaelsarsia elegans
Scyphosphaera apsteini
Dictyocha fibula
Eutreptia viridis
Halosphaera viridis
Isochrysis sp.
Platymonas sp.
Pyramimonas sp.
µ-flagellates

REFERENCES

Bohlke, and Chaplin. Fish of the Bahamas and Adjacent Tropical Waters. Miami: Livington Publishers. 1971.

Florida Department of Natural Resources. Summary of Florida Commercial and Marine Landings. 1971.

Gentry, R.C. "Hurricanes - One of the Major Features of Air-Sea Interaction in the Caribbean Sea." Symposium of Investigations and Resources of the Caribbean Sea and Adjacent Regions. 1968.

Heezen and Hollister. The Face of the Deep. New York: Oxford University Press. 1971.

Hill, ed. The Sea, Volume 3. New York: Inter Science Publishers. 1965.

Kofoed, J., and Malloy, R. "Bathymetry of the Miami Terrace." Southeastern Geology. 6(3): 117-175, July, 1965.

Pratt, R.M. "Lithogy of the Rocks Dredged from the Blake Plateau." Southeastern Geology. 13(1): 1-89, May, 1971.

Shepard, F.D. Submarine Geology. Scripps Institution of Oceanography, 1973.

Stommel, A. The Gulf Stream. University of California Press. 1965.

Sverdrup, H.V. The Oceans, Their Physical Chemical and General Biology. New Jersey: Prentice Hall, Inc. 1942.

Wood, E.J. Ferguson. "Phytoplankton Distribution in the Caribbean Region." Symposium of the Investigation and Resources of the Caribbean Sea and Adjacent Regions. 1968.

SEA SOLAR POWER PLANT - EFFECTS OF ENTRAINMENT
and
INTAKE SCREENING CONSIDERATIONS

by

Ken Haven

ABSTRACT

Effects of entrainment into the plant on biota through both warm and cold water intakes and screening methods to prevent such entrainment are discussed. Changes in physical and chemical properties and their affect on biota as the environment is entrained through the various unit operations in the plant are analyzed. These unit operations include: intake screens, intake pipes, pumping systems, internal piping, condensor/evaporator systems, and discharge. In general no part of the environment emerges from the discharge flow unchanged. As none of these effects are beneficial to the entrained environment, discussion of potential screening systems for both the warm and cold water intakes is included. The objectives of such screening are to: 1) prevent entrainment of any object or organism large enough to be potentially harmful to the plant, 2) prevent entrainment of as many living organisms as possible so as to avoid the ill effects of entrainment on them, and 3) to prevent harm to biota during the exclusion process. In general, optimum screening systems will be a function of the local environment at the intake, critical pipe dimension of the plant, and value of local marine populations. Some minimum level of screening will be required to protect plant operation. Beyond that, additional screening protection capabilities may be purchased for improved environmental protection depending on the results of a trade-off between construction and operation cost effectiveness and environmental protection.

SEA SOLAR POWER PLANT - EFFECTS OF ENTRAINMENT

and

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INTRODUCTION

A sea solar power plant may be viewed as an externality thrust upon the natural aquatic environmental ecosystem. The presence of the plant will cause some changes, or rebalancing, of that ecosystem. In evaluating the total costs and benefits of the power plant then, some attempt should be made to internalize into the calculations costs and benefits which occur to the environment as a result of the plant. Such an accounting would have to include costs and benefits in a number of general areas including: effects on the marine biotic environment and behavior from the physical presence of the plant; effects of excretions from the plant (leaks, sewage, biofouling retardation agents, etc.); effects of entrainment of a portion of the environment into and through the plant; effects on local circulation and current patterns as a result of the plant. This paper will evaluate only the effects of entrainment and will introduce some basic considerations for plant circulation and recirculation in Annex 1.

Once the effects on local organisms of entrainment have been evaluated, they can be internalized into plant cost considerations in the form of improved screening systems required to prevent entrainment.

EFFECTS OF ENTRAINMENT

Entrained seawater is introduced into the plant through both the warm water (200 ft depth) and cold water (1000 ft depth) intakes and passes through the following plant elements before discharge at a 270 ft depth: intake screens, intake piping, pumping systems, internal piping, condenser/boiler tubing, and discharge. This report will consider the effects of each element seperately before summarizing the total effects of entrainment on the environment. Hydraulic and temperature gradelines throughout the plant are shown in Figure 1.

Intake Screens:

Water velocities across these intake screens are 2 fps and 4 fps for the warm and cold water intakes respectively. Actual screening systems proposed for use

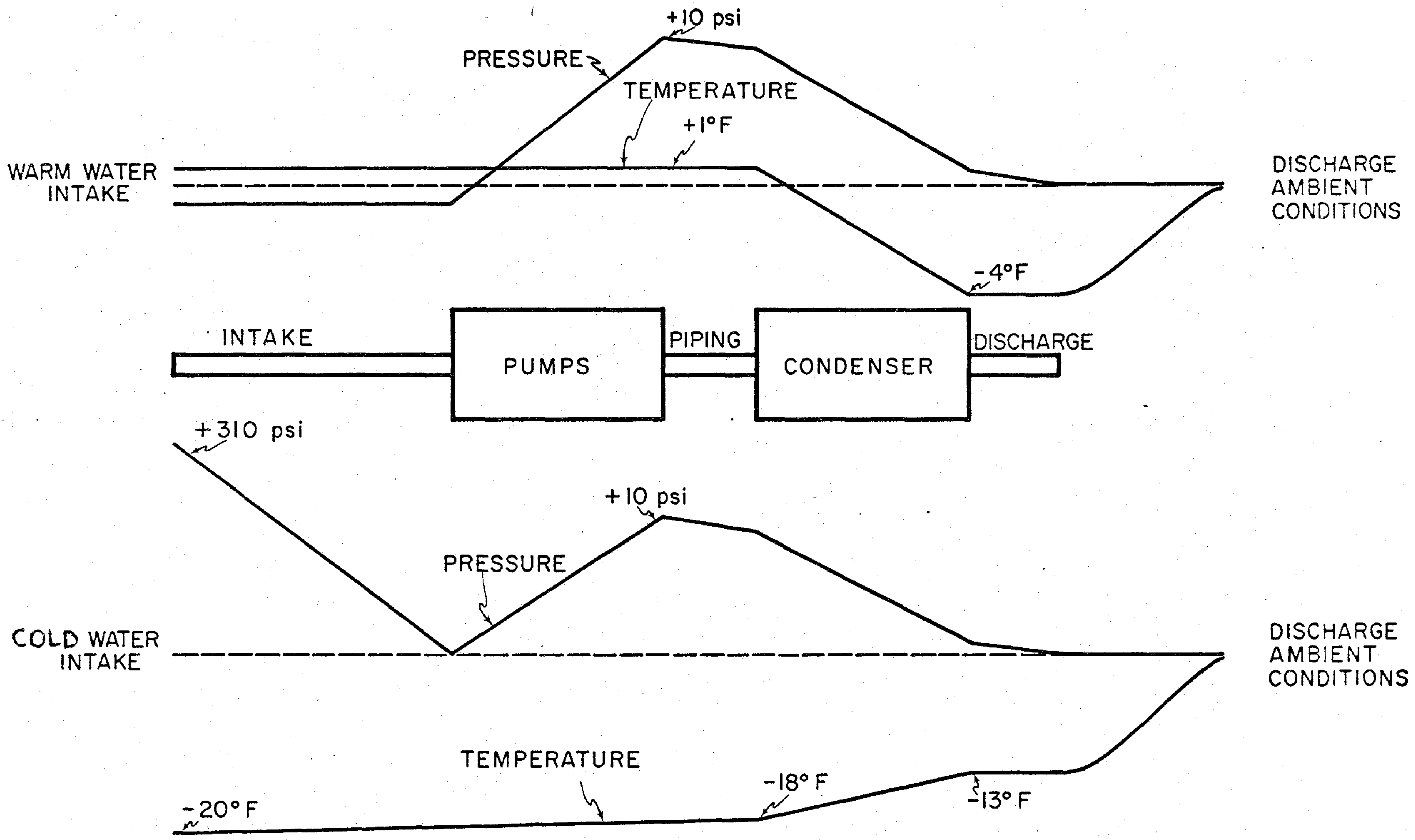


FIGURE 1. Plant Hydraulic and Temperature Gradelines

at each intake are discussed later in this report. It is sufficient at this point to say that the main screens at each intake will be 1/8 inch mesh screens, either traveling or serviced by a rotating scraper. The effect then of forcing the environment across these screens is as follows:

Chemical: No chemical changes or effects will result from transition across the screens.

Physical: No major physical changes will accompany traverse of the screens. There will be a minor head loss experienced. However, this is not significant either in itself or to the life forms associated with it.

Phytoplankton: Phytoplankton are microscopic plant forms, including such species as algae, which form the basis of the marine food chain. They lack independent motion, and are found exclusively in the euphotic zone of the ocean (that top layer of the ocean which receives direct solar light). This zone in the Florida current normally extends downward to the 250 ft layer (Murry and Hjort, 1965). Thus phytoplankton are not found in the environment of the cold water intake and are a concern only of the warm water flows.

Intake screens, in general, have no effect on phytoplankton. The extremely small size of this group means they are too small to be trapped by, or impinged upon, the intake screens, and will pass through into the plant itself. There will be some mechanical damage from collision of individual phytoplankton with mesh wires. This damage however will be minimal as the wire surfaces of the mesh represent only 21% of the gross mesh area and are "no-flow" areas. Thus, the natural flow of water around the mesh will tend to direct phytoplankton away from the wires.

There are several species of phytoplankton sufficiently large to be impinged (Sverdrup, 1942). Their total percent of this open ocean population cannot be known without an on site population survey. However, this percent, both in species numbers and in population number, is generally small (Hill, 1965). Those few who are impinged are, with rare exception, killed (Clark and Brownell, 1973). Intake screens do not pose a major threat to phytoplankton, and the vast majority of this population will pass through the screens undamaged. A much lesser number will suffer some mechanical damage, and a very few will be impinged and killed.

Zooplankton and small invertebrates: This class includes members of several invertebrate phylums which generally constitute zooplankton (Phylum Protozoa and Arthropoda which contain among others, dinoflagellates and crustaceans respectively) (Sverdrup, 1942) and the larval and juvenile stages of many other species. This group is found predominately in the euphotic zone, but is represented by a number of species at depth. For example, over 30 species of crustaceans are found at 1000 ft and several are centered at that depth range (Sverdrup, 1942). Certainly the total zooplankton population at the 1000 ft depth is a small percentage of that in the upper layers.

Many zooplankton, like phytoplankton, are sufficiently small to avoid impingement and pass through intake screens. However, a substantial number of species are large enough to be impingable (Hill, 1965). As zooplankton also lack any real swimming capability, these biota will be impinged and killed. Thus zooplankton suffer somewhat more damage at the screens than do phytoplankton. Mechanical damage is increased slightly and kill rates are increased substantially.

Large invertebrates and vertebrates: This group includes all of the larger invertebrates, as jelly fish, starfish, etc., and all true vertibrates in the ocean. The population is spread throughout the nektonic regions with essentially all classes, except for several benthic phylum and classes, represented at all depths. Population densities are generally much higher in the upper layer, due to increased food supply there, but substantial numbers still inhabit the depths (Murry and Hjort, 1965).

Intake screans are designed primarily to exclude this class of nekton from entry into the plant. With the exception of a very few of the smaller species, this is precisely what happens. However, this does not say they are not damaged and/or killed, only that they are excluded from the plant itself. Clark and Brownell (1973) report on a number of large screen kills which have occurred at existing large power plants and conclude that, "Screen kills are a major and growing problem with large power plants located on estuaries, using once through cooling...". If intake screens are not designed specifically to avoid such fish kills, these screens can have a greater impact on the vertebrates than would the rest of the plant were the screens not present.

The potential cause for these vertebrate and large invertebrate kill rates at this plant are two-fold. First, fish are naturally attracted to the type of velocity gradients associated with this plant's intake flows (Sonnichsen, 1973). Once there they tend to allign with those gradients. In a relatively short time, depending on the fish size, type and current velocity, they lose the ability to resist the intake current, and through exhaustion, are pulled into the intake screens and impinged. Once impinged, they will either remain impinged until death occurs, or will be capable of freeing themselves for escape. Those that do escape normally have suffered some degree of mechanical damage, and many die some time later as a result (Sonnichsen, 1973). The second potential for impingement of this plant is in the fact that the intakes are vertically oriented. Fish are much better able to resist a horizontal current than a vertical one (Sonnichsen, 1973). Thus any fish brought under the influence of these intakes have a higher probability of being pulled into the screens than they would if the intakes were horizontal.

Screens can pose a very large danger to large invertebrates and vertebrates. Screen, options listed later can minimize this impact. The potential impact of screens, however, is large kill rates and extensive mechanical damage.

Intake Pipes:

As the warm water intake is directly adjacent to its associated pumping system, there is no intake pipe to consider. The cold water intake pipe is a major structure and has a large impact on the entrained environment. These effects include:

Physical: Travel up the cold water intake pipe entails a vertical assent of 700 ft. Associated with this assent will be a pressure change of over 310 psi. In addition the water temperature will rise slightly - something less than 1°F - due to warming from the ambient water outside the tube. At 4 fps this trip will take 175 seconds, or just under 3 minutes. Thus, the time rate of change of pressure, or pressure gradient will be just over 1.75 psi per second.

Chemical: No significant chemical changes should be experienced.

Phytoplankton: As phytoplankton will not be present in the cold water flow, there will be no effects on phytoplankton from intake piping.

Zooplankton and small invertebrates: It has been reported by Sverdrup (1942) that "(because of the seawater stability) ...the organisms (primarily zooplankton) in general, have not developed highly specialized integuments and regulatory systems to protect themselves against sudden and intense environmental changes ...It follows that even slight changes in the aquatic medium are promptly brought to play upon the population."

The intense, rapid pressure change associated with this assent will severly affect most zooplankton species. Some of the more sensitive species will show significant kill rates, and the vast majority will undergo some form and degree of shock (Sonnichsen, 1973).

Large invertebrates and vertebrates: No general effects can be identified for this group. Effects will vary drastically from species to species. Those with swim bladders will generally be unable to survive this pressure change. The gradient is more severe than that which most species are able to compensate. They will literally explode. Those without such bladders should be basically unaffected by this pressure differential. However, at a depth of only 1000 ft, there are relatively few species without swim bladders (Thorson, 1971). The temperature change is sufficiently small that few species will be affected by it; however, many species will suffer some form of mild shock from general environmental changes (temperature, velocity, etc.) (Murry and Hjort, 1965).

Pumping Systems:

Pumping systems for this plant are currently designated as variable pitch, variable speed, propeller type pumps. Effects related here are geared as specifically as possible to this type of pump without considering differing effects from alternative pump types.

Physical: The physical effects of pumping will generally be confined to the pressure increase designed for the pumps. In this case, that pressure increase is slightly under 10 psi. Associated with this there will be a slight temperature increase, but again this should be less than 1°F in either the warm or cold water flow.

Chemical: Again, no significant chemical effects are foreseen.

Phytoplankton: Effects to phytoplankton will include: 1) some mechanical damage from the pump blades and pump housing, and 2) effects of the sudden pressure change. The effects of sudden changes in pressure are different for different species of phytoplankton, but in general these effects are a function of the absolute magnitude of the change, the duration of the change before return to normal conditions, the rate of change, or gradient, and the species involved (Sonnichsen, 1973). Insufficient data exists at this time to specifically evaluate the effect of any or all of these variables on a given species. In general, as any of these factors increases, the detrimental effects of that change will also increase.

In the case of this plant, the magnitude of that pressure change will be approximately 10 psi, the increasing pressure gradient across the pumps will be extremely steep, and a 20 second trip through the evaporator will be consumed returning that pressure to ambient conditions.

One case is related by Clark and Brownell (1973) where in a 15 psi pressure change held for 10 seconds had minor to moderate affect on the sample. It is felt that for this plant, the effect of pumping and associated pressure change will be relatively light on phytoplankton species, with most of these detrimental effects concentrated in the more sensitive species.

Zooplankton and small invertebrates: Greater mechanical damage will be experienced by this group than was felt by the phytoplankton due to increased average organism size. This mechanical damage should be light to moderate, with most of the damage concentrated in the larger species. Zooplankton reaction to pressure changes appear to be similar in nature, and in dependence on the magnitude, duration, and gradient of the change, to that of phytoplankton (Sonnichsen, 1973). However, the tolerance levels of most species of zooplankton are considerably higher than for phytoplankton species. Thus it is felt, though neither substantiated nor refuted by a literature search, that damage to zooplankton from pumping will not be severe. An emperical validation of this assertion is that planktonic samples are often run through pumps during on board ship processing with little if any damage (Curl, 1974). Certainly the pressure change from such a pumping operation is much less than that from the pumping system at this power plant; however, the analogy does add weight to this viewpoint.

Large invertebrates and vertebrates: Pumping effects on this group must be termed severe. Extensive mechanical damage will result from the size of this group of organisms and the speed of the pumps. Very few vertebrates will pass through such a pumping system without undergoing extensive damage. Clark and Brownell (1973) sight several examples of this type of inner plant kill in which many millions of fish were killed by mangling within a relatively short period of time.

Internal Piping:
There is a minimum of internal piping in this plant due to the large flow rates involved. It is only a very few feet from the pumps to the condenser/boiler assemblies. Over this short a distance there will be very few changes brought to bear on the environment. There will be a minor pressure drop due to piping head loss, and there will be additional minor mechanical damage on larger individual species. Beyond this, internal piping will have no significant effect on the environment.

Condenser/Evaporator System:
The physical composition of these systems is fully described in other reports. The critical dimension of these assemblies is the 1/4 inch diameter of the actual tubes. Anything larger than 1/4 inch will be impinged on the tube entrance plates. Velocities through the tubes will remain 4 fps. The environmental effects of this system include:

Physical: Pressure drops of under 10 psi will be experienced uniformly over the 18 second trip through these tubes. Simultaneously there will be a uniform temperature decrease of 4°F for the warm water flow and a temperature increase of 4°F for the cold water flow.

Chemical: Minor effects only will be experienced by the chemical environment. It is possible that the relatively high dissolved carbonate levels in the cold water flow, and especially the calcium carbonates will begin to precipitate out of solution under the influence of reduced pressure and warmer temperatures (Sverdrup, 1942). Should this occur, efficiency of the condenser would be significantly impaired.

Phytoplankton: Most of the effects of the changes associated with the evaporator system are residual effects tied to the initial pressure change at the pumps. That is, the reduction back to normal levels here will merely define a limit on the effects of the initial 10 psi pressure change rather than define a new effect on the phytoplankton. This total pressure effect has already been discussed. The temperature change of only 4°F would be insignificant by itself (Clark and Brownell, 1973, and Sonnichsen, 1973).

However, it may have some additive effect when combined with the plant pressure changes. At present this is not known; however, it appears that any effect on phytoplankton from this slight temperature change would be minimal and would be concentrated in the more sensitive species.

Zooplankton and small invertebrates: Simularly to phytoplankton, the effects of this pressure drop will be tied to the effects associated with the initial pressure increase at the pumps, and will not, therefore, be an independent effect of the condenser/boiler system. Turbulence at the entrance to the individual tube will increase mechanical damage somewhat. Finally, the temperature change of 4°F will have minor effect on all but the most sensitive zooplankton species. One test indicated that 80°F temperatures would result in, "death or damage to an appreciable portion of the more sensitive species" (Clark and Brownell, 1973). They further showed that sustained 95°F temperatures were required to kill essentially all zooplankton species tested. Temperatures in the warm water flow will never exceed the mid 70's (°F) and the evaporator condenser drop will leave the flow temperature prior to discharge in the mid to high 60's (°F). Similarly, the cold water flow temperature will never have a gradient or magnitude of change large enough to significantly affect zooplankton populations prior to discharge.

Large invertebrates and vertebrates: In reality very few live functioning members of this group would ever penetrate to the condenser/evaporator systems. Those that do penetrate to this point will encounter severe mechanical damage from turbulence at the tube entrance and from the small size of the tubes themselves. Few species of this class are small enough to negotiate the ¼ inch tubes. Therefore the majority of those delivered to this point will become impinged at the tube entrances and will die.

Clorine added: One of the probable biofouling reagents added to the flow at some point, especially to the warm water flow, will be clorine gas in very small smounts. This action would affect phytoplankton in that flow since as little as 0.5 ppm of clorine gas can be lethal to phytoplankton (Hirayamo and Hiramo, 1970). This effect, of course, is a function of the individual species, and not all species are affected at that concentration level. However, phytoplankton kill rates from such added clorine could be significant.

Discharge:

The discharge of warm water flow is at a 270 ft depth and is directed vertically downward. That for the cold water flow is at a 270 ft depth and is also directed vertically downward. Effects of this action include:

Physical: The warm water flow, at discharge, is at ambient pressure and is approximately 4°F below ambient temperature. As the discharge is directed downward, most of this water would penetrate to lower depths before its kinetic energy was disipated. Thus the temperature difference between this flow and the mean final ambient conditions is close to 3°F. The cold water flow is discharged at 11°F below the ambient conditions. Whether this discharge will form an integral plume and retain certain of its own characteristics or will disipate into local current flows and achieve ambient standards is not know at this time.

Chemical: Chemical changes for the warm water flow at this point will be negligible as ambient chemical environment is essentially identical with that at the warm water intake. The cold water discharge, however, will meet significantly different chemical conditions. Dissolved oxygen level is very low at 1000 ft but is much higher at the lower edge of the euphotic zone. pH is slightly higher at 300 ft than at 1000 ft. More important, nitrogen and phosphorous are some 8 and 5 times more concentrated at depth than at 300 ft, respectively. As these are critical to marine life, their release at this level could be significant (Svedrup, 1942), even though it is being released near the bottom of the euphotic zone.

Phytoplankton: As mentioned above the phytoplankton are being released with a downward velocity which should carry them out of the euphotic zone. As phytoplankton require the light present in the euphotic zone for growth and reproduction, this choice of release point and direction will result in a very high percent of population effective kill.

Zooplankton and Small Invertebrates: Discharge will have little effect on this group, especially for those in the warm water discharge. A significant number of the more sensitive species in the cold water flow will undergo some degree of shock to the 11°F temperature rise if that outflow mixes fully with ambient waters. (Clark and Brownell, 1973).

Large Invertebrates and Vertebrates: There will be little if any reaction to discharge by species in the warm water flow. Some, more sensitive species in the cold water flow will undergo a numbing type of shock, or disability, at the suden temperature change associated with discharge. Compared to other effects of the plant on species of this group, however, this final effect will be relatively minor.

A summary of these effects of entrainment on each segment of the marine biota is contained in Table 1.

INTAKE SCREENING CONSIDERATIONS

The previous section established a very low probability of survival for marine organisms entrained into either plant intake flow. As examples of this sort of damage the following inner power plant kill data is presented (Clark and Brownell, 1973):

1. Brayton Point Power Plant, Mass.: 50 million fish killed in 11 days by "mangling" in August 1971.

2. Millstone Power Plant, Conn.: 36 million fish killed in 16 days, November 1971.

Certainly these cases are somewhat extreme and not representative of normal operating conditions. However, they do dramatically demonstrate the potentail danger of entrainment. The question of whether this offshore, submerged solar power plant could duplicate such kill rates is yet to be definitely answered. With considerably less dense populations out of the shore/estuarine areas and especially with the low density populations at the 1000 ft depth (Sverdrup, 1942), the same entrainment force would entrain fewer organisms. However, while many shorebased plants use intake velocities in the .8 to 1.2 fps range, this plant will use a 2+ fps intake for the warm water flow and a 4 fps velocity for the cold water intake flow. This will significantly increase the attractive power of this plant. Available data indicates the start of a sharp upward trend in the percent of available fish entrained at intake velocities of around 0.9 fps (Clark and Brownell, 1973). Further, most shore based intakes are horizontally oriented, while the intakes for this plant are vertical. It has been demonstrated (Sonnichsen, 1973) that fish have a natural ability to oppose horizontal

TABLE 1. ENTRAINMENT ENVIRONMENTAL IMPACT SUMMARY

Environmental Element	Cold Water Intake	Warm Water Intake
Physical	Net ΔP = 310 psi, net ΔT = 20°F Net depth change = 700 ft. Net Δ time = 230 sec.	Net ΔP = 42 psi, net ΔT = 2°F Net depth change = 100 ft. Net Δ time = 42 sec.
Chemical	Substantial salinity, pH, N, P, and O level changes after discharge and mixing with ambient water	Negligible
Phytoplankton	No phytoplankton present	Light mechanical damage Light kill from ΔT and ΔP Severe kill from discharge below euphotic zone
Zooplankton and small invertebrates	Moderate mechanical damage Moderate-extreme kill from ΔP and ΔT	Moderate mechanical damage Minor kill from ΔT and ΔP Moderate impairment and shock
Large invertebrates and vertebrates	Massive kills from mechanical damage Light kills from ΔT and ΔP Severe shock, exhaustion, and impairment to living individuals	Massive kills from mechanical damage Severe shock, exhaustion and impairment to living individuals

currents, but not vertical. Lastly, the quantity of water passing through this plant, $7.5x10^3$ ft^3/sec per intake, is much higher than for conventional plants.

Whether or not these factors balance, or overbalance, the population reduction at the plant site is not known, and is an appropriate subject for further on site study. For the present it seems fully appropriate to assume that the potential exists for large marine organism entrainment damage, and that this damage represents a viable problem for plant design.

To avoid the ill effects of this entrainment, screening devices are used to exclude as much of the marine population as is physically or economically practical. This report will present considerations for evaluating the limits and desirability of these practicalities, and screening design options to meet them.

DESIGN PURPOSE AND PROBLEMS

The purpose of intake screening is in fact twofold. To this point, only environmental protection has been mentioned, and this certainly is one of the purposes of intake screening. The other purpose is plant protection. This second purpose is the original purpose for which screens were introduced, and still the prime motivation for screening. Screening for plant protection consists of the exclusion of any organisms or matter capable of damaging or clogging any part of the power plant. The most vulnerable element of the plant is normally the evaporation and condenser tubes and plant protection screening is geared to their protection requirements. Thus screen design is often taken as a function of condenser tube material type and diameter. The obvious importance of this type of screening is that clogging of, or damage to, these tubes reduces heat transfer on which the operation of the plant is based. The benefits from screening to protect the plant, then, are internal to the economic operation of the plant, and represent a positive return on the screening investment.

Conversely, screening designs to protect local marine organisms are, in general, screening over and above that required for plant protection. Such screening may consist of more elaborate or more extensive mesh networks or some provision for directional stimulation and intake escapes. These devices, however, represent diseconomies, or externalities, to economic power production at the plant. Additional dollars spent for this type of screening will not produce a return of either greater power output, or higher plant reliability. Thus additional biota protection screening represents an internal cost with an external benefit which cannot accrue to the plant. There is no economic motivation for this type of screening. Large screen fish kill figures attest to this lack of motivation. Nonetheless, society as a whole does accrue at least a portion of the fish protection benefits. It is to society's benefit to have the plant provide protective screening, and political and legislative pressures are being applied in that direction for all coastal zone industries. This tradeoff between cost reduction and fish protection is manifested over a range of options in screen design. This report includes discussion of the basic design type in this range.

The screen designs presented in this range are not identical with comparable shore based facilities designs because several unique problems are present at this plant which are not present at conventional shorebased, or even conventional offshore intakes. These include:

1. No free surface: Most screening systems take advantage of a free water surface for fish direction, exclusion, or for fish escape routes (Fair and Geyer, 1971). No such feature is present at either intake to this plant.

2. Temperature isolation: The sea solar power plant receives both heating and cooling from ocean waters. The temperature difference between the heat source and its sink is thus only 20°F or less. Once the cold water is entrained

at the 1000 ft depth, it must remain thermally isolated from the ambient water outside the intake pipe. Thus any fish escape mechanism for the cold water intake must maintain this isolation while still allowing fish to pass from the cold water tube out.

3. High intake velocity and volume: High intake velocities increase the head loss across the intake screen, especially for such fine mesh screens as will be used at this plant. Further these velocities increase impingement on the screens. The high volume flows require larger screen systems which increase power consumption of the screen system and reduce plant power output.

4. Differing local environments at the 200 ft and 1000 ft levels will result in differing screening criterion for the warm and cold water intake screens.

SCREENING REQUIREMENTS

Screening requirements for plant protection will be considered as mandatory requirements for any screening system. The impact of additional measures for marine biota protection may then be evaluated by the added complexity and cost of the screening system.

Requirements for plant protection screening must first be viewed by answering, "Protection from what?" Certainly, in general it is protection from the marine environment, but the specific elements within that environment from which the plant must be protected include:

1. Debris: While debris screening (trash sacks) are an integral part of shorebased, surface intakes, debris cannot be considered as a threat to this plant. The two intakes for the plant are each several hundred feet away from an ocean boundary (bottom or surface). As debris will either float on the surface or tend to sink to the bottom, the probability of debris being present in the vicinity of either of these intakes is sufficiently low so that the low risk does not seem to justify protection specifically designed against it.

2. Nekton: Nekton pose a threat to the power plant in two ways: first they may damage or clog some elements within the plant, as pump blades, piping, or condenser/evaporation tubes. This would reduce the efficiency of that element, and thus reduce that of the entire plant. Second, Nekton can be impinged upon the screening device. This reduces the net area of the screen and increases the head loss across it. Certainly no one fish could create a significant increase in head loss; but impingement of even small schools of fish could easily create such a loss. Moreover, it has been demonstrated that, when confronted with sudden velocity and current direction changes, fish tend to align with and remain aligned with that gradient (Sonnichsen, 1973) Thus the intake area of a plant will tend to be a congregational point for local populations. This tendency, combined with the previously mentioned inability of fish to resist vertical currents, could tend to increase the percent of locally available fish pulled into the intake apparatus, and thus increase impingement and associated losses.

3. Plankton: In general, plankton (both phytoplankton and zooplankton) are of such small size as to represent no threat to the plant (Hill, 1965 and Sverdrup, 1942). There are, however, a substantial number of zooplankton species sufficiently large to pose a threat of clogging to condenser/boiler tubes at this plant. Screens designed to exclude nekton from the plant should exclude these species of zooplankton equally well. However, it must be remembered that, lacking the ability to resist any current, these zooplankton will definitely be impinged.

4. Biofouling agents: The biofouling problem is a very important one for this plant and is dealt with in a separate report by T. Heinecke.

5. Plants: Large sea plants, such as kelp should not pose a threat to this plant as these forms tend to be coastal zone and/or surface layer inhabitants and are found in large quantities only in these areas (Raymont, 1967). Stray plants approaching the warm water intake would be impinged upon the intake screen.

Now knowing what is to be screened, the next question is how to screen it. A current standard criterion for plant protective screening is that the screen must not allow penetration of any matter which is greater than one half of the condenser tube diameter (Sonnichsen, 1973). As the diameter for the sea solar power plant is 1/4 inch, the screens must exclude everything larger than 1/8 inch in diameter. This requirements is much more stringent than that for conventional plants which often use 3/8 inch mesh screens (Clark and Brownell, 1973). Further, this requirement excludes bar screens as a potential main screening device. In order to exclude everything with a dimension greater than 1/8 inch these bars would have to have gaps around one third that width or 1/24 inch, as a common height to breadth ratio for many juvenile species is 3 to 1 (Sverdrup, 1942). Thus the prime screening device will be 1/8 inch mesh screen. Monel wire is recommended for its corrosion resistant properties in this environment, and is commonly used for shore based intake screens (Sonnichsen 1973). In addition to this screen, larger bar screens may be used to exclude anything sufficiently large to damage the screen. Lastly, this sort of screen is often either equipped with a rotating scraper or is a traveling screen with stationary scrapers so as to remove accumulated impinged material.

In order to now provide some additional measure of protection for those marine organisms pulled into the intake, we must first define the consequence of the type of screening system just defined on the various organisms in the marine environment. While these screens will exclude fish and larger invertebrates from the plant and thus prevent inner plant kills, they do not automatically reduce fish kills. In fact, fish kills by impingement on, or damage from intake screens have been as devastating as the inner plant kills these screens were to prevent. Examples of such massive fish kills include:

1. Millstone Power Plant, Conn: kill of over 2 million menhaden in 1971 (Clark and Brownell, 1973).

2. P.H. Robinson Power Plant, Texas: 7.2 million fish impinged in 12 months (69-70).

3. Suny Power Station, Va: 6 million herring killed in 2-3 months (1972).

The conclusion by Clark and Brownell was that, "Screen kills are a major and growing problem with large power plants located in estuaries using once through cooling..." Thus inadequate screening can easily do as much harm as good from an environmental viewpoint. Design criterion to improve the environmental protection capability of these screens by marine biota groups include:

Plankton: Screening, in general, cannot be designed to offer any protection to plankton. Lacking a true swimming capability plankton will be pulled into the plant if the mass of water they occupy is pulled in. To exclude even the larger species of zooplankton merely results in their impingement and near certain death on the screen. (Clark and Brownell, 1973). As it is, this large class of zooplankton fare best during entrainment. Screens represent a net "bad" for the plankton class regardless of the type of screen. They would be better off with no screen at all.

Nketon: It is possible through directional motivation measures to appreciably affect the probability of kill for members of the class. Such control measures include: velocity, temperature, physical barriers, light, electric shock, and others. These may be combined to form a wide range of directional stimuli for fish. The Bonneville Power Authority has conducted extensive testing in this area with salmon. Other similar research is summarized by Sonnichsen (1973). While this research shows some of these stimuli measures produce either inconsistent or non-uniform motivational results, the effects of partial physical barriers, such as screens, combined with sudden velocity gradients produce unifrom predictable results. Further, these phenomena are readily available at the sea solar power plant. Sonnichsen also details the application of this result into the angled barrier concept, shown in Figure 2. The basis of this concept is a fish's natural tendency to align perpendicular to and pointing away from the barrier.

Another concept potentially applicable to this plant is the establishment of a screening line outside of the influence are of the intake. Any organism at such a screen would not feel any influencing force from the intake current. This concept would result in an extremely larger spherical screen around the intake, and presupposes that most, if not all, fish under the influence of only the natural currents, will, when presented with a spherical screen mesh in that current, be capable and willing to circumvent it in their natural meanderings and migrations. Our literature search disclosed nothing to either substantiate or refute this assertion.

As was true with plant protection requirements, mesh size is critical to proper fish exclusion. Normal mesh size criterion to exclude fish in power plant design is given by:

$$M = .04\ (L-1.35)F \quad 5 \leq F \leq 6.5 \ \ldots\ldots. (1)$$

and

$$M = .03\ (L-0.85)F \quad 6.5 \leq F \leq 8 \ \ldots\ldots.. (2)$$

where M = mesh size
L = fish length in inches
D = fish body dpeth in inches
F = L/D = fineness ratio

For example, for a fish 15 inches long with 3 inches of body depth, F = L/D = 15/3 = 5. Thus equation (1) applies and,

$$M = .04\ (15-1.35)5$$

$$M = 2.7 \text{ inches}$$

Initial site population study indicates that the mesh size required to exclude fish, using the above equation and based on predicted vertebrate size distribution, will be larger than the mesh size required to protect the plant. To match that 1/8 inch mesh size, the controlling fish size would be less than 2 inches in length and 0.4 inches in depth. Certainly there is a full spectrum of biota sizes in any sector of the ocean. However very few organisms of this size range will be found at a depth of 1000 ft, and a very small percentage of the vertebrates in open waters. Thus if the mesh is maintained at 1/8 inch for plant protection, we are virtually guaranteed of near total vertebrate exclusion from the plant, and the entrainment damages to vertebrates mentioned earlier may be avoided.

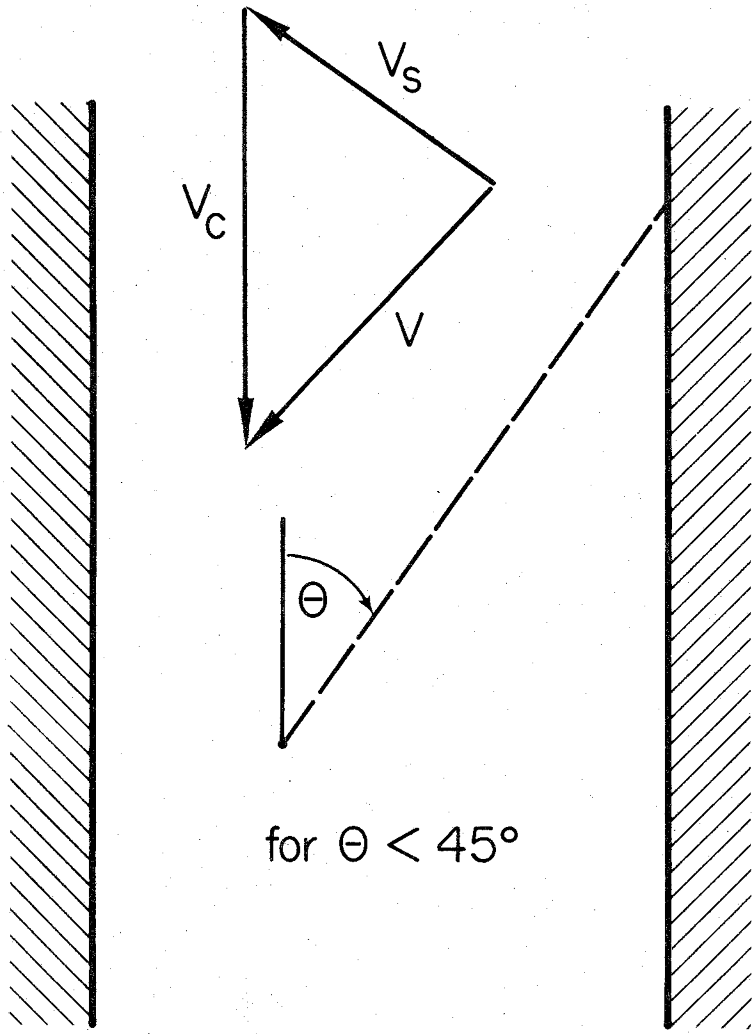
SCREEN OPTIONS

There are substantial differences in the environments and in the marine populations at 200 ft and at 1000 ft depths in the Florida Current. Due to these differences it will be easier to consider screen options for each intake separately. This is not intended to be an exhaustive list but a list representative of the various major types of screening options and trade offs.

Warm Water Intake Options:

1. No screen: This represents a "least initial cost" solution. As stated this plan would also benefit plankton classes. However, increased real cost due to increased risk to evaporator/condenser tubes is far greater than the dollar savings in initial costs. This does not appear to be a viable solution unless some new pumping design can be found which will guarantee break-up of any matter entering the plant greater than 1/8 inch. Even with such a new pump, the no screen solution will not reduce fish kills, and so is rejected as a potential screening plan.

2. Bars only: A bar screen consists of parallel bars placed over the intake entrance. Normal dimensions for these bar screens on shore based intakes are 1/2 to 5/8 inch bars at 4 inch to 4 1/2



V_C = current velocity

V_S = fish swimming velocity

V = resultant actual velocity

FIGURE 2. Physical Barrier Direction of Fish

inch bars at 4 inch to 4 1/2 inch centers. As the sea solar power plant does not have to plan for large debris and surf zone forces, these dimensions could be reduced to 3/8 inch bars at 3 inch centers. Head loss across the bar screen may be calculated through a number of methods.

Applicable formula include:

$$h = \beta (w/b)^{4/3} h_v \sin\theta \quad \text{(Fair \& Geyer, 1971)} \ldots\ldots (3)$$

where h = head loss
w = max. width of bars = 3/8 in.
b = minimum spacing between bars = 2 5/8 inch
θ = the angle between the screen and the flow
and β = a coefficient dependent on the shape of the bars.

β values vary from a high of 2.42 for rectangular bars, to 1.79 for circulare bars, and to a low value of 0.76 for "tear drop" shaped bars. In order to minimize head loss, tear drop shaped bars (semicircular facing the current tapered to small semicircular end facing downstream) will be used and β = 0.76. As indicated in Figure 3, however, the orientation of this bar screen to the actual flow direction, when that flow arrives at the screen, is not a constant and in fact varies from near 20° up to 90°. For approximate calculation purposes, a mean value of 45° may be used. Then with θ = 45° from equation (3),

$$h = 0.76 \; [(3/8)\cdot(21/8)^{-1}]^{4/3} \cdot 4 \sin(45°)$$

$$h_L = .149 \text{ ft}$$

A second method, and more consistent with other available data, is one proposed by the U.S. Department of Interior. Their criterion is:

$$h_L = K_t \frac{V_n^2}{2 g} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

where V_n = water velocity across the screen
and, $K_t = 1.45 - 0.45(a_n/a_g) - (a_n/a_g)^2 \ldots (5)$

where a_g = gross area across the screen
and, a_n = net area across the screen.
Thus, a_g = (75')(75') for a 75 ft square screen
or $a_g = 5625 \text{ ft}^2$
and $a_n = a_g$ -(area of bars)

If the bars are on 3 inch centers then there are

$$\frac{(75 \text{ ft})(12 \text{ ft/in})}{(3 \text{ in/bar})} = 300 \text{ bars}$$

thus,

$$a_n = 5625 - 300\,(3/8)(1/12)(75) = 5625 - 352$$

$$a_n = 5273 \text{ ft}^2$$

then,

$$a_n/a_g = 5273/5625 = .937$$

from equation (5),

$$K_t = 1.45 - (.45)(.937) - (.937)^2 = 1.45 - .421 - .878$$

$$K_t = .15$$

and from equation (4) using a 4 fps stream velocity,

$$h_L = K_t V_n^2/2g = .15(4^2/64.4)$$

$$h_L = .0384 \text{ ft}$$

for an intake velicty V = 2 fps,

$$h_L = .0096 \text{ ft.}$$

The head loss figures by this second method will be used from this point on.

An additional head loss is associated with this and all subsequent designs. The basic intake design for this plant approximates an inward projecting pipe into a reservoir, the ocean. Associated with this device is a K value of 1.0 (Fairbanks, Morse, 1959). From equation (4) this yields an h_L = 0.062 ft. It

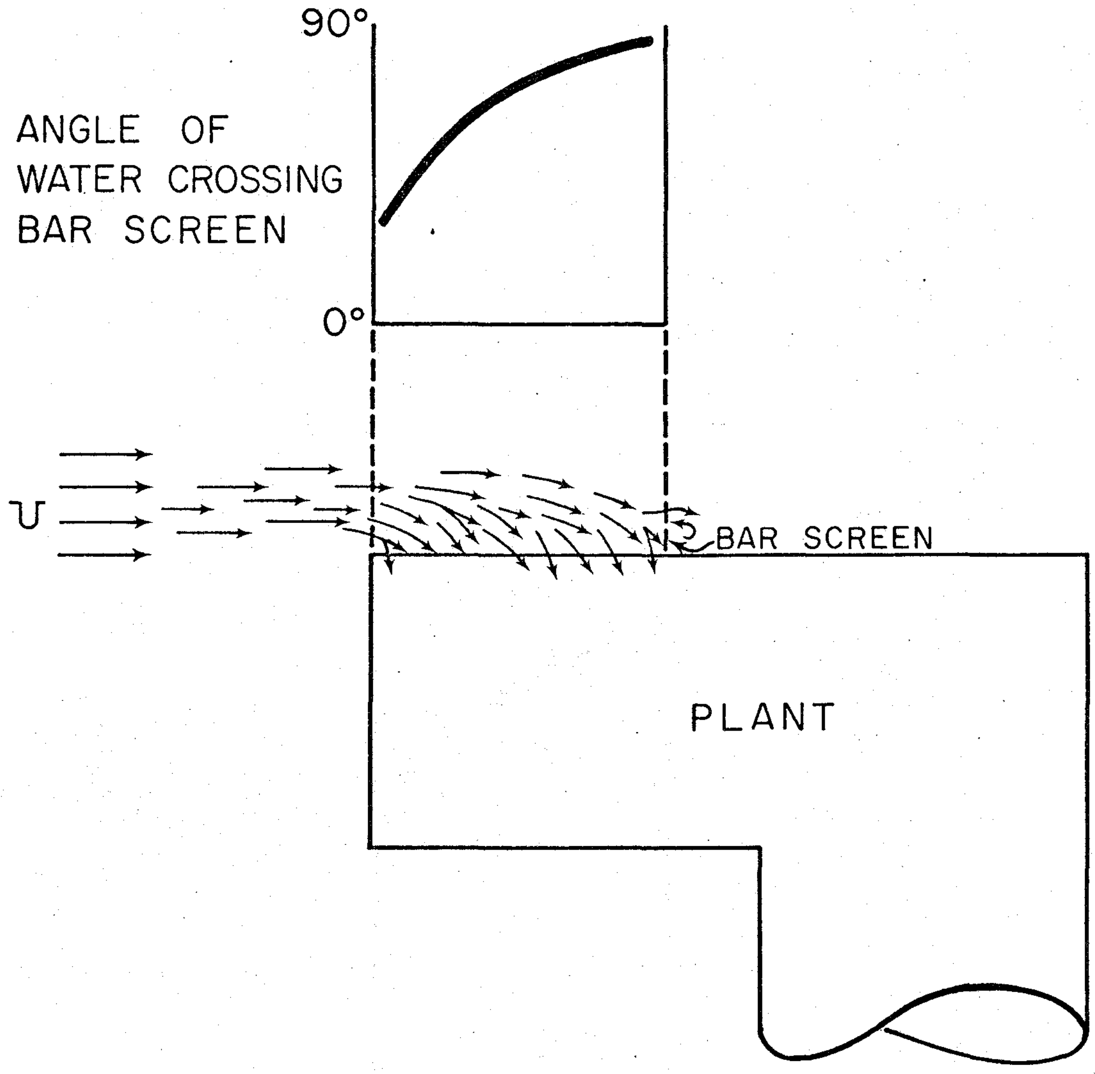


FIGURE 3. Current Patterns Across Horizontal Bar Screen

is not known how the addition of a 2 fps current perpendicular to the intake will affect the distribution of this loss. It is very probable that it will absorb some of the loss, and possible that it will absorb essentially all of it. In either case the resultant h_L associated with the internal pumping system is small and is identical for all intake screen systems. Further research is required to evaluate the actual value of this loss and it will not be included in the calculations for any of the following screen systems. As previously mentioned, a bar screen system alone will not provide the exclusion required for plant protection. This screening technique therefore cannot be considered as a principal screening device.

3. Stationary screen with rotating scraper: This type of plant intake screen is a fairly popular one with conventional power plant intake designs and should provide adequate plant protection for the sea solar power plant. This system consists of circular monel 1/8 inch screen mesh over the intake with a multiple blade rotating scraper operating over it. For this system 4000 ft^2 of the mesh would be required for the 75' circular warm water intake. Operating costs in the form of power requirements for the scraper motor are also introduced with this system. However this power requirement should remain less than 1 KW which represents 0.025% of the output power of the plant. Considerably more power will be used for lighting the crew quarters. The cost, then, of this system remains extremely low compared to the benefits it provides in terms of plant protection. This screen, however, does nothing to prevent fish kills. It has been stated that fish will tend to congregate near these intakes, and that kill rates from this type of screen can be high. This is the borderline then between mere plant protection, and plant and environmental protection.

Normally a bar screen would be employed to protect the mesh from damage by large fish or objects. Head loss for the total system may now be calculated.

We have already established a head loss for the bar screen.

$$K_{\text{bar screen}} = 0.15$$

To find a_g and a_n for use in equation (5) for the screen the mesh must first be defined. The 1/8 inch mesh has 1/64 inch thick monel wires surrounding 1/8 inch square gaps. This wire thickness is sometimes used in larger meshes (Clark and Brownell, 1973) and is used here to provide greater strength and durability. Taking a 9 inch by 9 inch unit area and letting N = the number of these 81 sq. in. areas in the mesh we have:

$$a_g = (9)(9)(N) = 81\ N\ in^2$$

a_n = N(81) - [area of wires in one direction + (area of wires in second direction - area of junction)]

or,

a_n = N(81) - [(64 w ins) (9 in) (1/64 inches/wire) + 64 [(9)(1/64)-64 (1/64)(1/64)]]

$$a_n = 64N\ in^2$$

then $a_n/a_g = 64N/81N = .79$

and from equation (5),

$$K_t = 1.45 - 0.45(.79)-(.79)^2$$
$$= 1.45 - .355 - .624$$

$$K_{mesh} = .47$$

$$K_{total} = K_{bar} + K_{mesh} = .15 + .47$$

$$K_t = .62$$

then from equation (4),

$$h_L = .62\ V^2/2\ g$$

for V_n = 2 fps at warm water intake,

$$h_L = .0386\ ft$$

4. Angled traveling screen: The placement and general operation of this screen is shown in Figure 4. The diagonally traveling 1/8 inch screen mesh provides adequate directional stimulus to direct

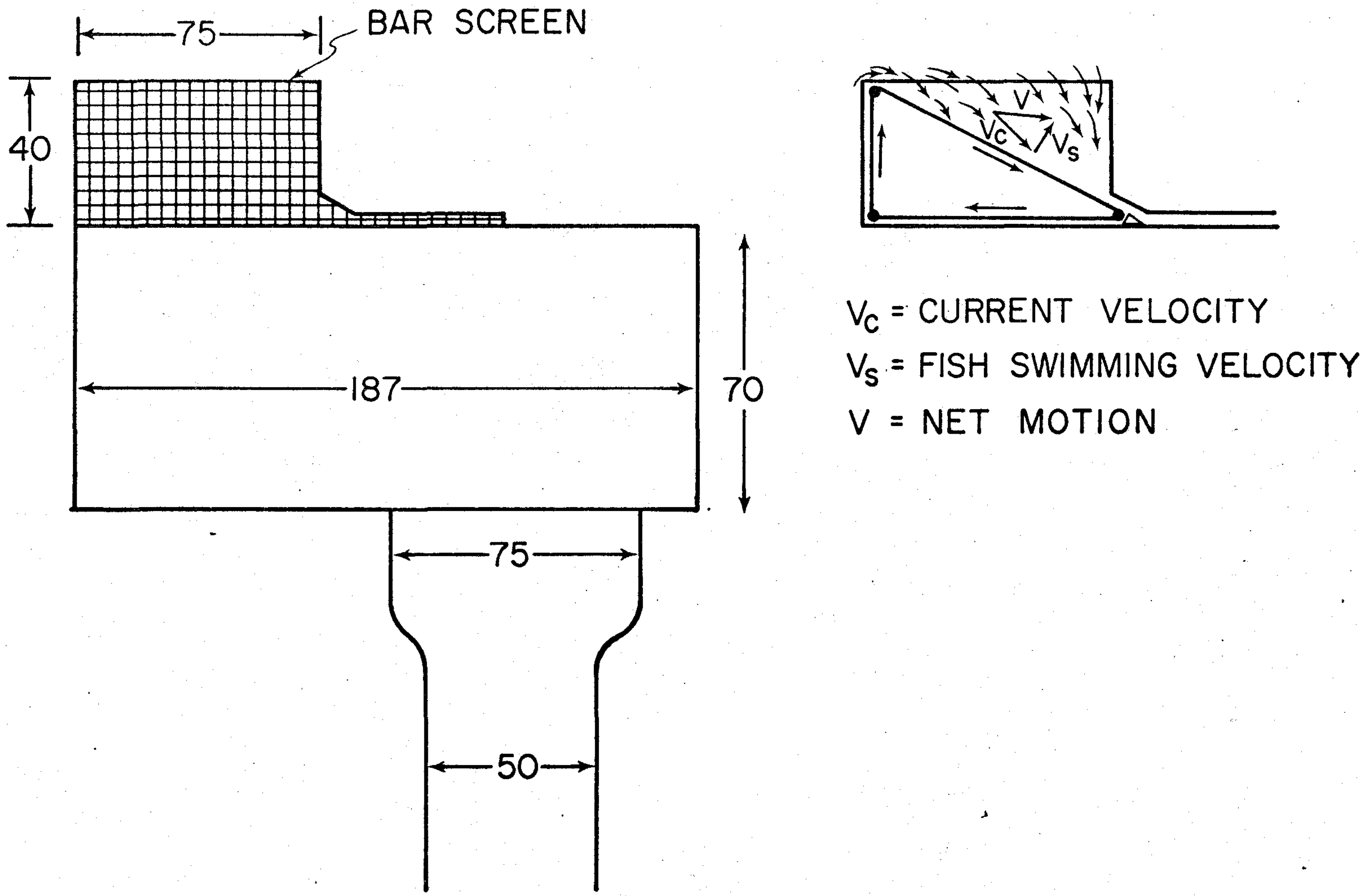


FIGURE 4. Traveling Warm Water Intake Screen System

most fish to the downstream (back) end of the screen well where a horizontal fish escape is located. It will be remembered from Figure 3 that the current across the bar screen at the top of the screen well is not a vertical current except at the back edge of the well. Thus, when a small fish first penetrates the bar screen and begins to align with the mesh screen and velocity gradients in the well, the natural currents will force him toward the back of the screen well. The farther back in the well a fish goes, the more vertical is the current attacking him. The fish's inability to withstand this current forces him deeper into the well. Eventually he will be forced toward the lower rear wall where he will pass beyond the back wall of the screen well and out of the vertical current to the escape tube where he is once again in a horizontal current forcing him out of the intake.

It is assumed that horizontal velocity components of the flow as it enters the screen well from the upstream side will create a positive relative pressure at the head of the escape tube, which will result in a net downstream, or escape, flow. Actual flow modeling would be required to substantiate this assumption. Should such modeling reveal a net upstream current through this tube, either modified bucket scraper systems could be adapted to use as scraping and pumping devices to force a downstream flow (Sonnichsen, 1973), or water jets could be added for the same purpose.

Certainly this method of fish direction will not work in every case. Some vertebrates, invertebrates, and all planktonic forms will still be impinged on the screen. As the screen moves toward the lower back edge of the well, it passes under the back wall of the well before reaching the turning sprocket and stationary scraper blades. At this point, without the direct vertical current influence some vertebrates will escape the screen themselves and enter the escape tube. The remainder will be scraped off the screen and carried away from the plant by the downstream current. Small debris remaining on the screen will be removed by the intake current.

This system should furnish adequate marine organism protection. However this protection is purchased at prices which includes:

Construction cost: This screen system uses 15,300 ft^2 of monel 1/8 inch mesh instead of 4000 ft^2 used by the stationary screen. Further this screen system requires roughly 13,000 ft^2 of additional wall construction for the screen well and escape tube and will require construction of a screen support and screen drive system. This screen well will also increase current forces on the structure and thus result in increases in the mooring and anchoring system. It must, however, be realized that these costs represent a small part of the total plant costs.

Operational costs: Operational costs may be divided into two categories. The first of these is screen drive power consumption. No exact figures are available for this power requirement, however 5KW is surely a liberal estimate. Again, this figure represents 0.125% of the total output, and still less than crew quarters lighting requirements. The second of these categories is added head loss. This added head loss is derived from the fact that the intake flow must cross the screen twice (see Figure 3) before it enters the plant.

Thus $K_{mesh\ total} = 2 \times K_{mesh} = 2 \times .47$

or $K_{mesh(t)} = .94$

Total h_L then is derived from $K_t = K_{bar} + K_{mesh(t)} = .15 + .94$

or $K_t = 1.09$

From equation (4),

$$h_L = K_t\, V_n^{\,2}/2g = (1.09)\ V^2/64.4$$

using $V_n = 2$ fps,

$$h_L = 4.36/64.4$$

$$h_L = .0678 \text{ ft}$$

While this is an increase of .029 ft in h_L, it is still within the limits of the plant pumping potential.

One additional benefit of this screening system, covered in more detail in Annex 1, is that it provides an additional 40 feet of vertical separation between the warm water intake and discharge. This added separation could be beneficial in reducing the probability of recirculation at the plant. Complete plant flow modeling would be required to determine the extent to which these 40 feet were needed.

This traveling, angled screen system still meets all requirements for plant protection; and, for the added costs listed above, will provide greatly superior environmental protection. It would appear that this trade off is a good one as the total net benefits are high and the costs moderately low, even though these benefits do not accrue to the plant itself.

5. Horizontal intake: The use of a horizontal intake would eliminate vertical intake currents associated with other systems and thus remove a major cause of fish entrainment. Two possible designs for such an intake are shown in Figs. 5 & 6. Figure 5 shows a single rectangular horizontal opening facing the local current with a vertically traveling screen assembly. Two drawbacks to this design are: 1) A 90° elbow is added to the internal piping system with an associated K value of 0.8; and 2) as demonstrated earlier with the Millstone, P.H. Robinson, and Surry power station screen kill figures, all of which were based on horizontal intake flows, the elimination of a vertical intake current alone does not necessarily reduce fish kill rates. Through the use of a downward traveling screen (see Figure 5) and an adequate fish escape system at the bottom of this screen, the second drawback can be substantially reduced. However the first cannot be reduced and results in a total K value for the screening system of 1.89*

*K = 1.89 derived from summation of K values for bar screen (0.15), twice that for the 1/8" mesh screen (2x 0.47) (the flow must cross the screen twice), and a 90° elbow (0.8).

This relatively high K value would seem to render this system less desirable than that shown in Figure 6. This system uses a cylindrical, horizontal intake under a solid intake well cap. By allowing intake flow from all sides the internal 90° elbow is eliminated. By use of a horizontally rotating screen and either scraper of water jet system with a protected fish escape on the downstream side of the intake adequate environmental protection can be provided.

Costs associated with angular and cylindrical system construction and operation are shown in Table 2.

Finally, the rectangular system will provide a greater increase in total plant drag force and thus result in a larger increase in mooring and anchoring requirements.

Either of these screen systems will provide both adequate plant protection and adequate environmental protection. Their total costs are similar in magnitude to the angled traveling screen system, as are derived benefits.

6. Ball screen: as previously mentioned, and as shown in Figure 7, this screen is based on the principle that if marine biota are prevented from entering the influence area of the intake current they will neither be attracted to it nor be impinged upon the ball screen. The ball screen would be centered over a circular intake opening between 1/2 and 1/3 radius above the intake. The ball radius is estimated to be approximately equal to the intake diameter, or 75 ft. With an average natural current of approximately 2 fps at this depth (Sverdrup, 1942) and with an intake velocity also of 2 fps this size seems more than adequate. Only plant flow modeling can accurately define the limits of intake influences under various actual current conditions. Quite probably the final screen shape would deviate from spherical due to local current influences. Mesh size of this screen would be 1/8 inch.

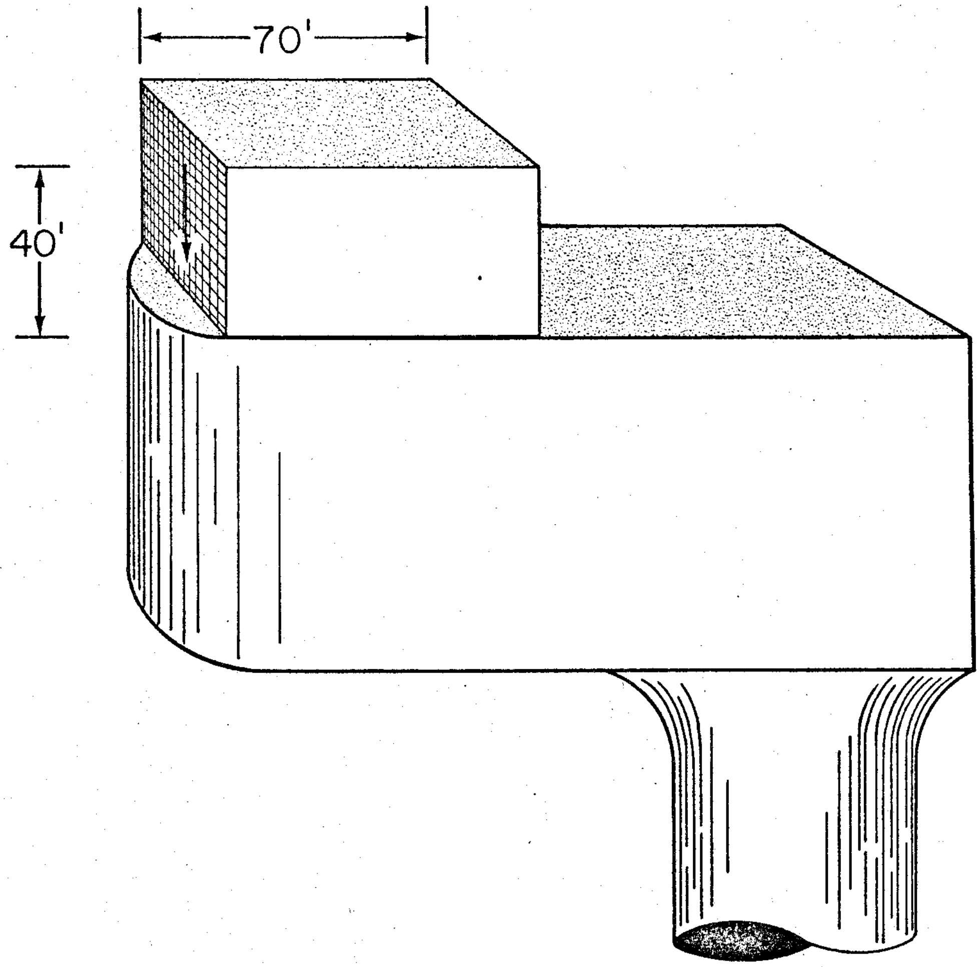


FIGURE 5. Vertically Traveling Screen with Horizontal Intake.

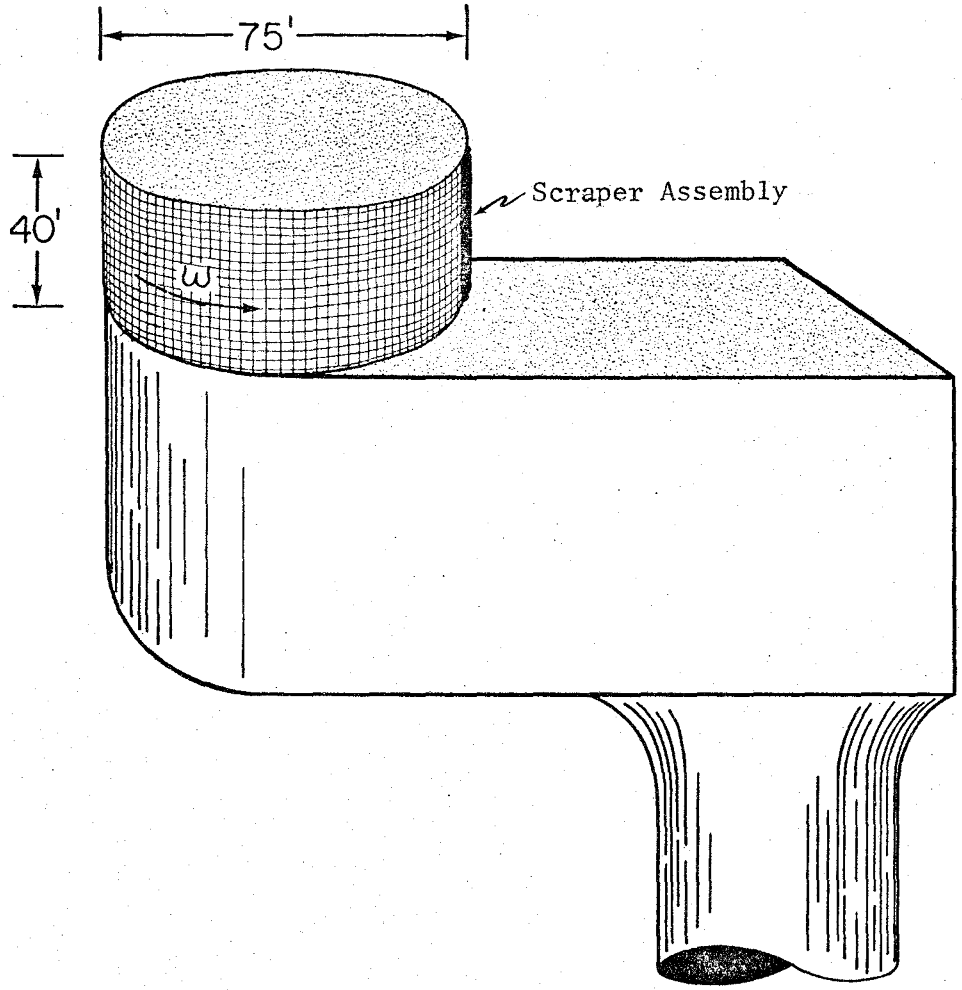


FIGURE 6. Horizontal Traveling Screen with Horizontal Intake.

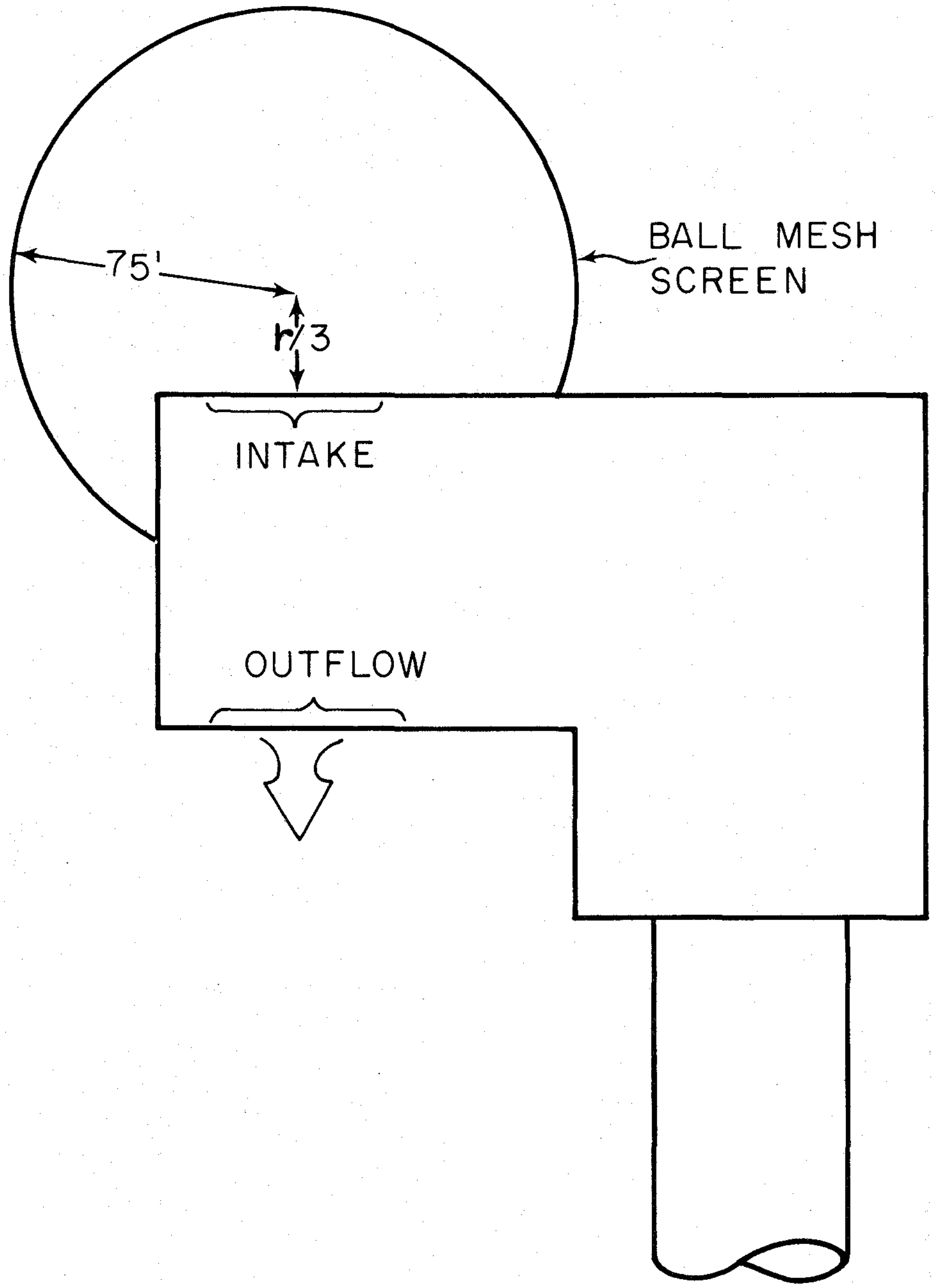


FIGURE 7. Ball Screen Design

TABLE 2. HORIZONTAL INTAKE CONSTRUCTION COST COMPARISON

Construction Parameter	Rectangular Intake	Cylindrical Intake
1/8" monel mesh	11,000 ft^2	9,400 ft^2
wall construction	22,500 ft^2	4,500 ft^2
screen support	minor cost	major cost item
screen drive system	relatively simple	more extensive & complex
power requirements	approx. same as option #4	approx. same as option #4

The costs for this form of environmental protection include:

Construction cost: If a spherical shape with 75 ft radius is assumed, this screen requires 69,400 ft^2 of mesh as compared to 4,000 ft^2 for the stationary screen, and 15,300 ft^2 for the traveling screen. In addition, some screen support system would have to be constructed and this increased plant weight would have to be accounted for and countered.

Operating costs: A plus for this screen system is that there are no associated operating costs. As the screen is outside of the influence of the intake, the loss across the screen will not be absorbed by the intake system. The natural currents will absorb this loss. There are, however, two operational problems associated with the ball concept. First is biofouling. As this screen would be well within the euphotic zone biofouling agents would be free to attach and grow on it. In the long run this reduced screen net area would force the intake system to pull water across the screen. The second problem is, in essence, a solution to the first. This is cleaning. Periodic cleaning will control biofouling. However, cleaning of this screen presents several problems of its own, including size and location of the screen. It is assumed that some economical cleaning system would have to be developed to make this screen effective for the warm water intake.

Cold Water Intake Options:

Before discussing actual options for the cold water intake screen a brief discussion of the 1000 ft depth environment and its population is appropriate. The population at the 1000 ft level is void of phytoplankton (Sverdrup, 1942; Hill, 1965) It also has a small zooplankton population. Even though a large number of species, including, as an example, over 30 species of crustaceans, the total number of biota present is fairly small. Nektonic populations are also somewhat reduced in total numbers of representatives at this level, even though almost every species is capable of existence there. Moreover, nekton at this depth do not tend to "school" and 1000 ft depth populations are made up of individuals or small groups of individuals; thus the threat of sudden mass impingement is greatly reduced.

The natural current at this level is much lower (0.9 fps) than at 200 ft and the design intake velocity is higher (4 fps) due to the smaller intake pipe. Thus the area of influence of this intake will be larger than that of the warm water intake. Finally, game and commercial fish do not tend to populate such depths. It appears then that the net change from 200 ft to 1000 ft is to reduce the benefits of environmental protective screening by reducing both the population to be protected and its direct value to man.

From this basis, cold water intake screen options are:

1. No screen: While this option is somewhat more attractive here than at the warm water intake, its minimal cost savings are still vastly outweighed by the economic rsik of damage or clogging to condenser tubes. In general, even large fish lack the ability once trapped in the intake pipe to resist the vertical current and escape out the bottom (Sonnichsen, 1973). Thus no screening provides few benefits, high risks and is still an unsatisfactory option.

2. Bar screen: A bar screen across the cold water intake similar to that described for the warm water intake would certainly exclude most organisms at this depth. However, they would allow a steady, low volume, flow of fish larger than plant protection requirements will allow. If the decreased cost, and ease of operation of the screen can justify the potential requirements for significantly more frequent plant shut downs for condenser cleaning, this may be a viable alternative. It does not appear at this time that it is. Further, use of a bar screen only allows no provision for reduction of impingement or for screen cleaning. Periodic cleanings are extremely impractical at the 1000 ft level, and the expense to install an on-site cleaning system negates the one real advantage of a lone bar screen. In either case, a lone bar screen does not appear adequate.

3. Ball screen: A modified ball screen based on the criterion that intake influence at the screen will be "minimized" instead of non-existent, could be an appropriate solution to the cold water screening requirements. This change in size criterion would counter the tendency of an increased influence area of this intake to expand the ball screen still larger. The ball screen would offer the same advantages of both plant and expanded environmental protection it did for the warm water intake. Biofouling is not a problem at this depth, and impingement would be minimized by the lower population level especially by the minimal sessile population. Thus cleaning requirements would be minimal and cleaning could be scheduled during major plant shutdown and repair periods.

This screen system entails no operational costs and no operating parts with associated risk of failure at a 1000 ft depth and difficulty of maintenance or repair at that level.

Thus the ball screen is more attractive for the cold water intake than at the warm water intake and appears to be a plausible system for the lower screen.

4. Mesh screen with rotating scraper: This system would place a bar screen at the intake entrance and a horizontal mesh screen with rotating scraper in the cold water intake pipe. It would be benifical for the mesh screen to be as near to the plant as possible to facilitate maintenance on the scraper motor and screen assembly. However, this endangers the required thermal isolation of the cold water flow as an exit port is required for scrapings and for a fish escape. One solution is to equip the escape port with a pump or with a water jet system to prevent warm ambient waters from entering the cold water flow. Either of these devices would also assist in jettison of mesh scrapings and in fish escape. The several disadvantages of the system include:

a. Locating the mesh screen assembly at plant level means entrained biota will rise 700 ft before screening. The full effects of such a rise have been described earlier. In summary, they are all detrimental to entrained organisms. The result is that mechanical damage at the screen will be increased as will scraper damage and damage from the jettison system. This problem may be overcome by locating the screen assembly at the 1000 ft level. However, this additional environmental protection brings with it a requirement to provide increased reliability for the scraper drive and jettison assemblies.

b. Screen head loss: As a result of the increased velocities in the cold water intake pipe and of the stringent plant protection requirements for mesh size, head loss across the screen is significantly increased over that of the warm water intake. If either the mesh requirement (reducing screen K values) or intake velocity could be reduced, screen head loss would be significantly lowered. This relationship is shown in Figure 8.

c. Power consumption would also be increased with this system with both scraper drive and jettison systems in operation. However, construction costs would be reduced from those of the ball screen as only 1,950 ft^2 of monel mesh will be required.

This system provides adequate plant protection. Environmental protection is somewhat reduced, but environmental damage should not significantly increase as a result because of reduced local population levels.

5. Horizontal intake: Addition of a horizontal intake similar to those described for the warm water intake would improve environmental protection at the cold water intake over the previous plan. Such a horizontal intake screen would have to be located at the intake entrance (1000 ft) to avoid bringing fish under the influence of the vertical intake flow. This intake screen, as described for the warm water intake, will provide adequate protection for the cold water intake local environment. However, the cost of either type of horizontal intake screen is significantly higher than that of the stationary screen system. Complexity of the screen drive mechanism is also significantly higher, especially for the cylindrical design. Thus reliability goes down and risk of failure rises. Further, environmental protection at this level isn't worth as much as it is at the warm water intake. There are many fewer total organisms and few, if any, game or commercial catch species present. Thus the total benefit of environmental protection beyond plant protection requirements is considerably lower than at the warm water intake. While this is a perfectly good screening system, the high construction cost and high cost of insuring long term system reliability may be higher than the total returning benefits of environmental protection. Further on-site population study is required to better evaluate this trade off.

6. Angled traveling mesh screen: This screen would be structured similarly to the one for the warm water intake, and would either be located just before the entrance to the plant or at the pipe entrance. If it were located adjacent to the plant, the added expense for directional stimuli would essentially be wasted, since, as mentioned above, fish arriving at that site are severely impaired, and, in general, would not respond properly to the stimuli (Sonnichsen, 1973). The additional costs associated with locating this screen at the cold water intake entrance are prohibitively high for the increased benefits associated with the reduced local population levels. This screening method then seems to represent environmental "overkill" for the cold water intake and is not recommended.

A summary of potential screen systems for each intake is shown in Table 3.

CONCLUSIONS:

1. Initial entrainment is a function of intake velocity and orientation. The vertical intakes, and high intake velocities associated with the plant will significantly increase the number of organisms entrained at both intakes.

2. Cumulative affects of entrainment on the living biota of the natural environment include:

Phytoplankton: Light to moderate mechanical damage from screens, pumps and boilers; light kill rates from pressure changes; moderate impairment from pressure and temperature changes; complete cessation of growth and reproduction, and large kill rates from location and direction of discharge.

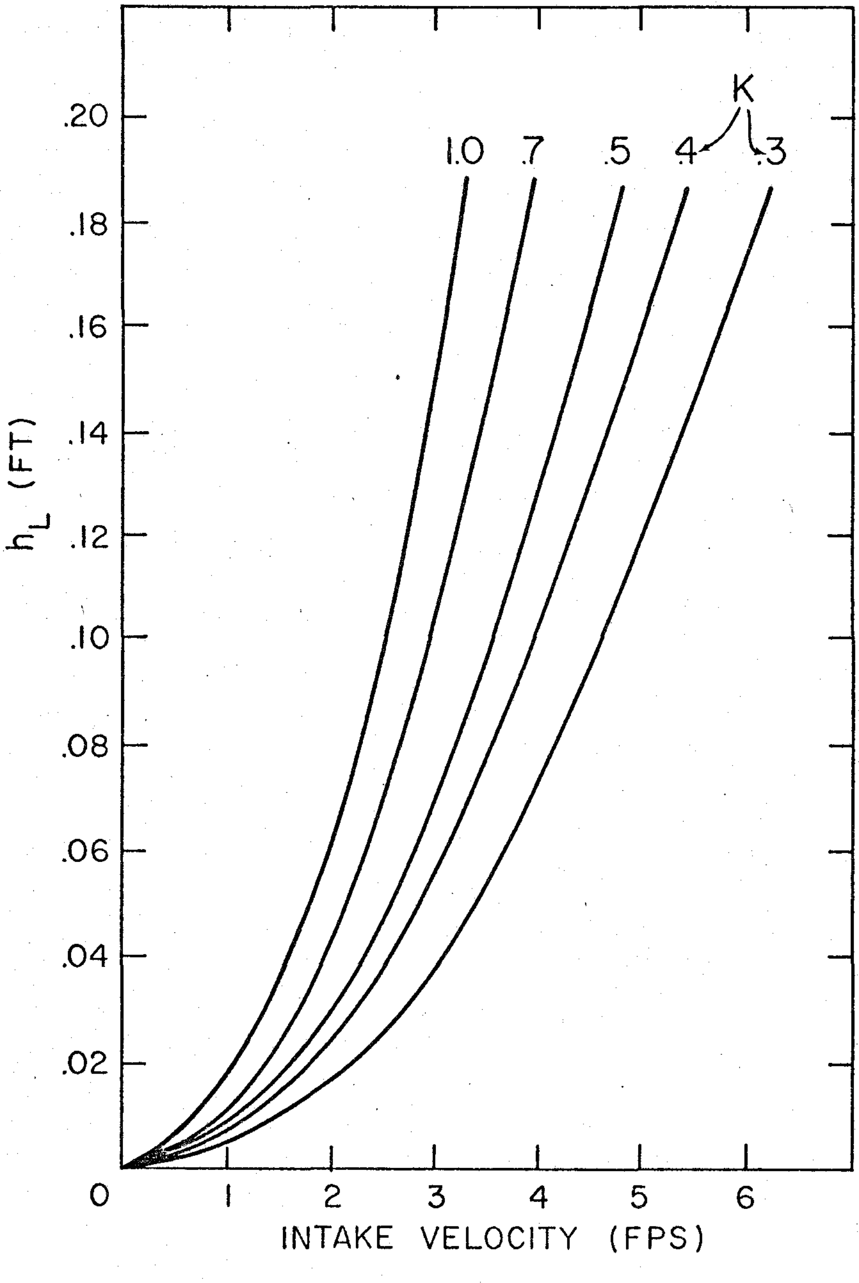


FIGURE 8. Screen h_L as a Function of K Values.

TABLE 3. PLANT INTAKE SCREENING OPTION SUMMARY

Screen System	Cold Water Intake	Warm Water Intake
Bar screen	Inadequate	Inadequate
Stationary screen with scraper	Probable optimum	Adequate plant protection Inadequate biota protection
Ball screen	Acceptable	Impractical; excessive biofouling
Horizontal intake screen	Acceptable; may have excessive cost	Acceptable
Traveling screen system	"Overkill;" excessive cost	Acceptable

Zooplankton and small invertebrates: Moderate to extensive mechanical damage from screens, pumps and boiler/condensers. Minor to moderate kill rates and extensive shock and impairment from presure and temperature changes. Net effect: this class should fare better than anyother, however, a majority of entrained organisms will still undergo death or severe impairment before discharge.

Large invertebrates and vertebrates: Under normal operation screens will exclude essentially all of this group. Kill rates and damage at the screen are a function of screen design. If this group is allowed to be entrained severe mechanical damage would result to most organisms. In the vast majority of cases this damage will lead to death. For the remainder, light to moderate shock should be experienced from pressure and temperature changes, and most vertebrates will suffer from exhaustion from attempts to oppose the intake currents. The net effect will be that essentially all organisms in this group will be killed if entrained. Those who are not killed will be moderately to severely impaired.

3. It seems economically essential to have some form of screening at both the warm and cold water intakes. After this minimum essential level of screening has been achieved, additional dollar expenditures, as both initial investment and as operational power consumption, may be invested to provide increased marine biota protection. For this plant the required increased investment is relatively small for a major increase in biota protection.

4. Differing environments at the two intakes create different screening design criterion for the two intakes. As a result different screen systems appear as acceptable for the two intakes. Acceptable systems for the warm water intake include the angled traveling screen system and horizontal intake system, while the cold water optimum appears to be either a modified ball screen, or a stationary mesh screen with scraper and associated bar screen.

5. Additional research is required to fill several important gaps in the available information. Areas indicated for special emphasis include:

a. Long term on-site population study to determine population densities, monitoring values and size and age characteristics for local environments at both intakes. This study should also include data on seasonal variations.

b. A current flow model study for the plant. This study would include effects of warm and cold water intakes and outflows on local circulation patterns, potential recirculation currents and their prevention, and a definition of the area of influence for each intake. Data should be collected for natural high, mean, and low current velocity levels.

c. Additional study on motivational stimuli response emphasizing species prevalent at this site.

d. Expanded investigation in the effects of entrainment of various species through the various components included in a sea solar power plant.

REFERENCES

Clark, J. and W. Brownell. Electric Power Plants in the Coastal Zone: Environmental Issues. American Littoral Society Special Publication #7, October 1973.

Curl, Herbert. Discussions. School of Oceanography, Oregon State University, Corvallis. April 1974.

Fair, G. and J. Geyer. Elements of Water Supply and Waste Water Disposal Systems. New York: John Wiley and Sons, Inc. 1971.

Fairbank, Morse & Company. Hydraulics Handbook. 3rd edition. Kansas City: Fairbank, Morse & Company. 1959.

Hill, N.E. The Sea, Vol. 2. New York: Inter Science Publishers. 1965.

Hirayamo, K. and R. Hirano. "Influences of High Temperature and Residual Chlorine on Marine Phytoplankton," Marine Biology 7:205-213. 1970.

Ingle, R.M. Summary of Florida Commercial Marine Landings--1968. Florida State Board of Conservation, 1970.

Joyce, E.A., Jr. The Commercial Shrimp of the Northeast Coast of Florida. Professional Paper Series #4, Florida State Board of Conservation, 1965.

Moe, M.A., Jr. A Survey of Offshore Fishing in Florida. Professional Paper Series #7, Florida State Board of Conservation. January 1963.

Murry, J. and J. Hjort. The Depths of the Ocean. New York: Stechert-Hoffman Service Agency, 1965.

Raymont, J. Plankton and Productivity in the Oceans. Oxford, England: Pergamon Press, 1967.

Sonnichsen, J., et al. A Review of Thermal Power Plant Intake Structure Designs and Related Environmental Considerations. National Technical Information Service, 1973.

Sverdrup, H.V., et al. The Oceans: Their Physics, Chemistry and General Biology. Englewood Cliffs, N.J.: Prentice-Hall, Inc., 1942.

Thorson, Gunnar. Life in the Sea. New York: World University Press, 1971.

U.S. Department of the Interior. Design of Small Dams. U.S. Government Printing Office, 1965.

ANNEX 1

CIRCULATION AND RECIRCULATION CONSIDERATIONS

There are two questions which should be answered with respect to circulation and flow around the sea solar power plant. These are: 1) Will the presence of this plant and its induced flows significantly alter the general circulation patterns, and if so, what area will this change affect? And 2) Will the proximity of discharge ports to intakes induce recirculation patterns to form around the plant?

Some perspective of size of the flows will shed light on the first question. The Florida Current has a average net transport of 2.6×10^7 m^3/sec, or 7×10^8 ft^3/sec, spread over a 50 mile width (Sverdrup, 1942). The power plant will intake 25×10^3 ft^3/sec through each of two intakes, as shown in Figure A-1, and uses two separate discharges. Depths associated with each activity are noted on the left of that figure.

To compare these two flows we must consider the area over which the plant intake flows could possibly hold any influence. With a vertical intake an appropriate horizontal maximum influence width would be on the order of twice the diameter of the upper, and 3 to 3 1/2 times the diameter of the lower intake on each side of the respective intake. These differing ratios are a result of the differing natural current levels and differing intake velocities at the two depths. Widths of influence are then 5 x D, or 5 x 75 = 375 ft, for the warm water intake, and 8 x D, or 8 x 50 = 400 ft, for the cold water intake. For convenience we may assume this influences 1/12 mile total width, or 440 ft. The transport of the Florida Current over this same 1/12 mile = $(1/12)(7\times10^3/50)$ = 1.5×10^6 ft^3/sec. Thus the combined plant flow, $7.5 \times 10^3 \times 2 = 15 \times 10^3$ represents only 0.1% of the natural flow through this 1/12 mile area. Said differently, this means that 99.9% of the water flowing through this 1/12 mile wide sector will not be entrained into the plant. Certainly more water than this will be influenced, and diverted, as a result of the plant flows, but only those waters passing through the plant will undergo, or cause, any major shift in circulation patterns. Thus we may tentatively conclude that plant induced currents will not have a significant effect on normal current patterns outside the immediate vicinity of each intake and discharge.

Data gathered for this study will only point toward a solution to the question of recirculation. Sonnichsen (1973) has suggested that 60 feet of vertical separation between intake and discharge is sufficient to prevent recirculation for conventional plants on lakes. Compared to that situation, here the flows are much larger than in a conventional plant, which would require greater vertical separation. However, natural dominant currents are also present here which tend to reduce required separation. Finally, these are vertically oriented flows directed away from each other. This also would reduce required separation.

To what extent these factors counteract each other is not known, and is an appropriate subject for further research. Should the increased flow criterion predominate, the 70 ft vertical separation between warm water intake and warm water discharge may not be sufficient to prevent recirculation pattern development. There lies an additional advantage of the angled traveling mesh screen for the warm water

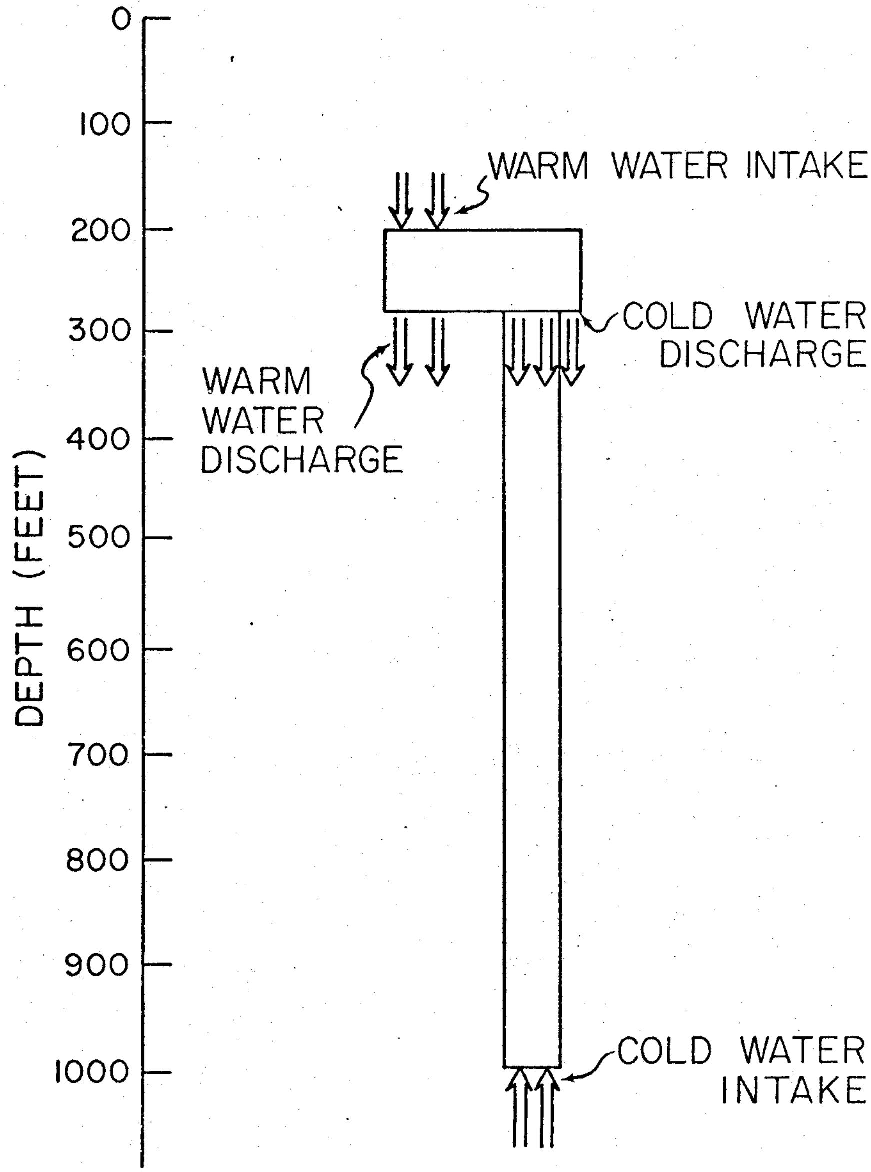


FIGURE A-1. Plant Flow Patterns and Elevations intake. This screening system would provide an additional 40 ft of vertical separation between these flows.

A second solution to this problem would be the insertion of horizontal stabilizer plates on the sides of the plant between the warm water intake and discharge. While these plates would increase the drag forces and weight loads for the mooring and anchoring system, they would provide additional isolation between the warm water intake and discharge ports.

With 700 ft of separation between the cold water intake and discharge, there will be no recirculation into the cold water intake. Natural current forces will provide adequate separation. Further, as the cold water discharge port is horizontally displaced downstream from the warm water discahrge and is discharging colder, heavier water, recirculation from the cold water discharge to the warm water intake will not occur if recirculation from the warm water discharge to the warm water intake can be prevented.

Thus it appears that recirculation will not present a serious problem. However the sensitivity of this plant to temperature fluctuations which could be caused by recirculation indicate further study in this area is warranted.

SEA SOLAR POWER PLANT - BIOFOULING CONSIDERATIONS

by

Thomas Heinecke

ABSTRACT

Possible undesirable affects of biofouling on a offshore, submerged structure are studied; the types of marine life to be expected, probable growth rates of this life, and state-of-the-art prevention techniques are considered.

Only limited attachment and growth is expected to occur once the structure is at the selected site, in a submerged position. If construction occurs at a nearshore site, biofouling will probably occur. If the structure spends extended time on the surface for repairs, etc, problems may occur.

Paints or coating are available with protection capabilities up to and exceeding three years. Electrolysis techniques are available for protection of smaller, more sensitive equipment.

SEA SOLAR POWER PLANT - BIOFOULING CONSIDERATIONS

by

Thomas Heinecke

INTRODUCTION

For this report, biofouling can be considered the attachment and growth of marine organisms on exposed materials. This usually causes a performance and maintenance problem for structures placed in a water environment. These "structures" may range from ocean going vessels to timber pilings. For vessels, a six-month growth of barnacles can cause up to a 40% increase in drag or fuel consumption (Starbird, 1973). A large tanker may have 15 tons of marine growth within a few years time. Pilings can be completely destroyed, if not protected, within short periods of time by marine borers.

Our particular concern is for the effect of biofouling on the Sea Solar Power Plant. Physically it is a large, submersible structure, to be constructed in a nearshore environment, towed to a chosen site and moored. The structure will be capable of controlling its buoyancy characteristics. The two most immediate problems that would be caused by biofouling are:

1. Increased drag forces causing increased demands on the mooring and anchoring systems.

2. Increased weight, affecting the flotation capabilities of the structure.

Less obvious, but significant problems that would also be encountered are:

1. Increased corrosion (crevice and pitting).

2. Increased headloss within piping systems.

3. Reduction of heat transfer characteristics.

4. Loss in general mechanical/structural efficiency.

This report will approach the biofouling problem through the following areas of concern:

A. The types of biofouling flora and fauna expected at the plant.

B. The critical marine environments for biofouling growth.

C. The probable organism growth rates, and relative importance of the various organisms.

D. The present techniques used to prevent or discourage growth.

E. Expected unique problems for the proposed power plant.

TYPES OF FLORA AND FAUNA EXPECTED

Naturally the particular types of marine life that would cause problems would vary from site to site. This report will be concerned with types of marine organisms which seem to generally cause the more serious biofouling problems. All are found generally throughout the oceans. Again, the occurance is a function of the water environment, (temperature, salinity, light, turbidity, current, food available, the attachment surface for sessile organisms, etc) typical of many fouling, sessile (attached) organisms is the characteristic of having more than one form. Different physical forms usually occur for the following stages:

1. Free swimming or floating (nauplii stages)

2. Period of settlement (cyprid stages)

3. Final growth and maturity (adult stage)

The problem flora and fauna in a common order of time sequence attachment onto a structure, are as follows (Lovegrove & Robinson, 1966):

1. Slime, algae and bacteria (Enteromorpha and ectocarpus) Physical form - brown and green weeds.
2. Hydroids
3. Tubeworms
4. Barnacles
5. Sea squirts (Tunicates)
6. Mussels

Slime (bacterial and algal growth) can be on most marine structures in a matter of a few days (Haderlie, 1968 and Dolgo Dovskava, 1969). This type of growth, sometimes considered minor, can however affect drag coefficients on ships and therefore, would affect our structure. This type of growth can get to be several centimeters long. Another related problem is that this growth creates an environment attractive to other organisms (mainly food).

Hydroids, usually considered to be littoral marine forms, have been found growing in depths up to 7000 meters. Most of these organisms reach a maximum length of 20 centimeters, but some have been found up to two meters in length.

Tubeworms are usually the first major animal to attach and start adult growth. They may be found on structures in less than two months.

Approximately next in time are the barnacles. Three of the more common species in U.S. waters are *Balams Balanoides*, *Eliminius Modestus* and *Tubularia Larynx*, (over 100 species of barnacles cause fouling problems). Growth, in ideal conditions, can be up to a few centimeters in height in less than a year. 240 feet is the approximate depth limit for many barnacles.

Tunicates or sea squirts, (a leathery-texture shapeless animal) and mussels (marine bivalve mollusks) also may cause problems. As most of the literature available considers the algal and bacterial growth, the hydroids, and the barnacles of greater importance, the emphasis in this report will be on these particular organisms.

CRITICAL MARINE ENVIRONMENTS

The majority of the studies accomplished thus far state that the nearshore environment is by far the "better" environment for the organisms mentioned earlier. Some tests by NCEL of the Coast of California in greater than 6,000 feet of water found only hydroids and slime on test panels after 36 months (Muraoka, 1970). Shallow water (less than 200 feet) is also desirable (Ray, 1959). Basically the shallow, nearshore environment provides the best environment for the organisms.

Water currents are also important in providing the proper environment. In steady currents exceeding about 1.6 knots (approximately three ft/sec), most sessile organisms have difficulty in attaching to an object (Starbird, 1973). Barnacles, once attached however, can withstand currents up to 12 knots before being dislodged.

Basically, the fouling organisms causing the majority of the problems live and grow best in:

1. Shallow, nearshore environments (a major exception is the goose barnacle, *Lepas*)

2. Euphotic zone

3. Currents less than 3 ft/sec.

PROBABLE GROWTH RATES, RELATIVE IMPORTANCE OF THE VARIOUS ORGANISMS

This section will primarily be a compilation of several reports, including Haderlie (1968), Muraoka (1970), Walters (1966), and Thompson (1967). The intent is to indicate which marine organisms can cause major biofouling (and corrosion) problems at given depths and distances offshore. Also included will be the average growth rates and sizes of various organisms. This material has been condensed into Table 1.

Most of this information comes from the California coast, with a bit from England. Generalizing we can see that barnacles, tubeworms, and mussels are most predominately found in shallow water, nearshore environments, growing in terms of one to two centimeters high within months, and with 100% coverage of the exposed structure possible within 1 1/2 years. In deeper offshore waters, test panels showed a lack of barnacle and mussel growth (these panels were not exposed to a nearshore environment at any time). They did show a 100% algal and bacterial coverage, and a 50% hydroid coverage within two to three years.

The majority of the reports emphasized the nearshore environment for the following reasons:

1. Nearshore is considered to be the "worst case" location.

2. It is assumed that nearshore characteristics could be related to deep water conditions.

3. Nearshore, shallow-water testing is less expensive.

4. Common collection sites (buoys, platforms, etc.) are generally in the near shore area.

It appears that once a site for a Sea Solar Power Plant is identified, on site studies should be made to pinpoint local biofouling characteristics.

The relation between biofouling and corrosion rates were considered by Muraoka (1970) and will be mentioned here. Comparing heavily fouled shallow water test panels to deep water lightly fouled panels, the corrosion rate of steel and aluminum was two to four times greater on the heavily fouled panels. The lightly fouled panels were corroding

TABLE 1. GROWTH RATE OF ORGANISMS ON METALS AT VARIOUS DEPTHS AND LOCATIONS

Organism	Location	Depth	Near Shore	Off Shore	Individual Growth Rate	No. of Organisms
Algae, Bacteria	Calif.	50'-100'	X		30cm long/mo.	100% coverage/8 mos.
Algae, Hydroids	Calif.	25'	X			100% cov./70 days
Hydroids, Anenomes	Calif.	60'	X			100% cov./18 mos.
Barnacles	Calif.	100'	X		13mm high/3 mos.	10 organ./mo./sq.ft.
Tubeworms, Barnacles, Mussels	Calif.	25'	X		½" dia./ 2/3 yr.	100% cov./1 yr.
Barnacles	England		X			1 organ./sq.cm./day
Barnacles, Mussels	Calif.	60'	X			90% cov./18 mos.
Barnacles	Calif.		X		14gms/day/sq.meter	
Hydroids, Slime	Calif.	6800'		X		100% cov./3 yrs.
Bacterial Slime	Calif.	6000'		X		100% cov./3 yrs.
Hydroids, Anenomes	Calif.	200'		X		50% cov./21 mos.
Barnacles, Mussels	Calif.	6800'		X		0% cov./3 yrs.
Barnacles	Calif.	200'		X		0% cov./21 mos.

at about twice the rate of the control panels, (in a sterile sea water environment). Therefore biofouling does contribute to corrosion (primarily by enhancing crevice and pitting corrosion).

PRESENT TECHNIQUES USED TO PREVENT OR DISCOURAGE GROWTH

It should be apparent that the prevention of biofouling is an important design consideration. A few of the possible techniques that could be used to prevent or retard biofouling are:

1. Protective Coatings.
2. Clorination
3. Increased Temperature
4. High Water Velocity Environment
5. High Frequency Sound

The last three are used in special cases (such as piping, small mechanical devices, etc.). The first two methods will be considered here in more detail.

Coating Systems: Coating systems of toxic paints, coal tars, epoxy are in common usage for protection of ships and other marine structures. Some of the more common naval coatings are listed in Table 2.

Much of the antifouling paints (80% in 1972) are copper based. As copper is leeched out of the paint, it prevents bacterial and algal growth. Table 3 shows the minimum concentration in seawater for various compounds to inhibit the growth of algae.

Coating systems are available to effectively prevent biofouling, and of course are dependent on the structural design and metals chosen.

Clorine Generation: Another technique used to prevent biofouling is the electrolytic generation of chlorine. This method is normally used in localized, sensitive areas where any growth would be detrimental to the use of the area, and where coatings might also affect usage (radio antennae, radar, etc.). Anodes and cathodes are placed at close intervals (two to ten feet), either as point sources, or as strips, and D.C. or A.C. currents are induced. A thin layer of chlorine at the surface of the metal is sufficient to inhibit growth. In approximately two knot currents, two ppm to ten ppm free chlorine at the source seems to give about 0.1 ppm to one ppm protection, depending on the surface shape, turbulence, etc. Continuous treatment of one ppm prevented all fouling during a testing program (Lovegrove & Robinson, 1966) along the coast of Miami, Florida with down to 0.25 ppm residual chlorine preventing fouling. The dosage need not be toxic to the organisms to prevent attachment. Just creating an unsatisfactory environment (called exomotive dosage) is sufficient. For example, in a free chlorine concentration of 0.02 to 0.05 ppm, after several hours, mussels detach and leave, but are not killed by that dosage. The purpose however is to prevent initial attachment. For example, barnacles once attached do not leave, and if killed still leave an attached shell.

The electric current can be applied in an intermittent or continuous manner. It has been found (Lovegrove & Robinson, 1966) that twice the current for 1/2 the time length can be more effective. One problem with intermittent treatment (even up to eight hours on), is that some organisms (mussels and anemones) may just close up and wait. Using A.C. current seems to be at least 40% more effective, besides preventing major deposits on the cathode. Tables 4 and 5 give some indication of dosage and effectiveness of this method.

Free chlorine would be produced at the anode and hydrogen at the cathode. Other possible secondary reactions near the anode are:

TABLE 2. SHIP COATING SYSTEMS

Coating	Effective Life Against Biofouling	Recoating Procedure
Coal Tar	1 year	Dry Environment
Vinyls	2 to 3 years	"
Epoxy	2 to 3 "	"
Chlorinated Rubber	2 to 3 "	"
Fiberglass	2 to 3 "	"
Bitumous Metal Pigmented	1 year	"
Zinc Silicates	several years*	"

*Corrosion resistant application only.

TABLE 3. MINIMUM CONCENTRATIONS OF COMPOUNDS TO INHIBIT ALGAL GROWTH

Compound	Concentration (ppm)
Arsenic	15 to 17
Phenarsazine Cloride	0.8
Copper Chloride	7 to 10
Cuprous Thiocyanate	7
Mercury	4 to 5
Phenyl Mercury	1
Dibutyl Lead Diacetate	2
Tributyl Tin Oxide	< 1
Tributyl Tin Fluoride	< 1

1. Hypochlorite Ions
2. Chloramine
3. Chlorate Ions
4. Iodide, Bromide, etc.

Other possible reactions at the cathode include:

1. Magnesium Precipitation
2. Production of Calcium Hydroxides

TABLE 4. LENGTH OF TIME TO KILL ALL ORGANISMS, CONTINUOUS DOSAGE

Organism	Dosage (ppm)		
	10	2.5	1
Anemones	4 days	6-8 days	15+ days
Mussels	5	5	12-15
Barnacles	3	4	5-7

TABLE 5. LENGTH OF TIME (DAYS) TO KILL ALL ORGANISMS, INTERMITTENT DOSAGE

Organism	Period of Treatment (hrs/day)				
	1	2	4	8	24
Anemones	10+	10+	10+	10+	4
Mussels	10+	10+	10+	10+	5
Barnacles	10+	10+	10+	5	3

To produce the concentration of chlorine required, currents on the order of 200 milliamps down to 50 milliamps are required, depending upon the size and shape of the area to be protected, currents, etc. From the testing program previously mentioned, 200 milliamps protected an area of about five square feet and 50 milliamps, an area of about three square feet, for about 1/2 year (approximate length of the test). The voltages ranged from about six to twelve volts, or an energy requirement in the order of 0.1 to 0.4 watts per square foot. Even though protection seems to be good, energy requirements could get quite high if a large structure was to be entirely protected in such a manner. If the entire Sea Solar Power Plant structure was to be protected in such a manner, using some approximate sizing, the energy requirements for continuous protection would be approximately 500 kilowatts. Additional cost considerations would include installation and maintenance of the cathodes and anodes. (This cost may be partially offset by cathodic protection provided for the structure against corrosion.)

EXPECTED PROBLEMS FOR THE PROPOSED POWER PLANT

As stated before, the nearshore shallow-water environment is the environment most likely to cause biofouling. Our structure will probably be fabricated in a nearshore environment, and during that period (approximately 1 year), would be exposed to all forms of biofouling. (This could be significantly reduced by using a fresh water construction site.) Either each wetted portion of the structure would have to be individually protected as it was built, or a rather large maintenance project would result before plant movement to its final site.

The most likely method of short term protection for this construction phase would be use of some coating system. Any parts to be protected by electrolysis would have to use an external source of power, or not be installed until the final operational site. Again, if the construction stage spends at least a few weeks in salt water before towing to sea, biofouling could be a problem.

When the plant is at its final site, it will be located at a depth of 200 to 270 feet. This is just at the depth where biofouling becomes less of a problem (bottom edge of the euphotic zone in many areas). At this depth rate of attachment and growth would be slower. However, this is somewhat a function of how well the protection has prevented biofouling from occuring at the nearshore construction site, as the structure may bring its own community of organisms with it. None of the studies encountered looked at such a possibility. Assuming no community and a good protection system, new biofouling at that depth and location should be slight.

The structure will also be in a surfaced position approximately once every 14 days for crew replacement and routine maintenance. Even though the structure is at an offshore location, attachment of organisms, especially goose barnacles, could occur during these surfaced periods.

The most likely organisms to attach and grow at the offshore (submerged) site would be algae, bacteria, and hydroids, unless other organisms were brought to the site.

CONCLUSIONS

1. Possible biofouling organisms include algae, bacteria, hydroids, barnacles, mussels, and tubeworms.

2. The biofouling organisms can increase drag coefficients by 40%. They can also create buoyancy problems from the added weight.

3. Growth rates may range from 100% barnacle coverage and up to a centimeter high within a few months, to only algal and bacterial slime after a few years, depending upon the environments.

4. Protection against biofouling for the nearshore phase (the worst condition) and for the offshore site will be necessary. It will probably consist of a coating system with some production of chlorine.

REFERENCES

Dolgo Dolskava, M.A. "Biological and Physico-chemical Factors Influencing Effectiveness of Antifouling Paints", Corrosion Engineering, 1969.

Haderlie, E.C. "Marine Boring and Fouling Organisms in Open Water of Monterey Bay, California", A Thesis, U.S. Navy Post Graduate School, Monterey, California, 1968.

Lovegrove, T. and T. Robinson. "Prevention of Fouling by Localized Chlorine Generation", University of Miami, 1966.

Muraoka, T.J. "Relation Between Marine Fouling and Corrosion Rate of Carbon Steel and Aluminum Alloy at Surface and at 6,000 Ft." Naval Civil Engineering Lab Technical Report No. 681. NCEL, May, 1970.

Naval Ships System Command. Preservation of Ships in Service. Technical Manual Chapter 9190. U.S. Government Printing Office, 1970.

Ray, D.L., ed. Marine Boring and Fouling Organisms. Seattle: University of Washington Press, 1959.

Coating Systems Guide for Hull, Deck and Superstructure. T & R Bulletin 4-10. The Society of Naval Architect and Marine Engineers, 1967.

Starbird, E. "Friendless Squatters of the Sea", National Geographic. 144(S): 622-633, November, 1973.

Thompson, J.C. "Effects of Deep and Shallow Ocean Environments on Construction Materials", Naval Civil Engineering Lab Technical Report, 1593. NCEL, 1967.

Walters, A.H. Biodeteriorization of Materials. London: Elsevier Publishing Co., 1966.

SOLAR SEA POWER PLANT

ENVIRONMENTAL IMPACT - AQUACULTURE POTENTIAL

by

William G. Mercer

ABSTRACT

The regions of natural upwelling, representing about one tenth of one percent of the total ocean surface, are producing approximately one half of the world's fish supply. By bringing deep nutrient laden water, containing nitrogen and phosphorus, from approximately 1000 feet depth, up to the solar sea power plant to be utilized in the condensors and then released, an artificial upwelling is caused in the surrounding area.

It is the utilization of this area of upwelling, and the resulting increase of biological productivity associated with it, to provide a substantial source of fish protein with the minimal amount of cost incurred, which this study investigates.

By providing an area of protein available for human consumption, as well as a source of energy; such a plant concept could provide a step toward solutions of the global shortages of food as well as energy.

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INTRODUCTION

The economic feasibility of a solar sea power plant, such as the one under study depends upon the utilization of all aspects related to its operation. Because of the artificial upwelling created by bringing nutrient laden water up from a depth of approximately 1000 feet, the possibility of aquaculture as a resulting by-product of the basic operation has great potential.

While the present population of the world is increasing at a rate of three percent per year, food production is only increasing at a rate of 2.7 percent; with food shortages presently being realized throughout the world. What is needed is a low cost food supplement rich in high quality protein. At the present time biological life in the oceans is the least tapped protein source of unknown abundance.

The major objective of this report is to investigate the possibility of utilizing the artificial upwelling as a means of a protein base in the form of aquaculture.

PRESENT FORMS OF AQUACULTURE

Aquaculture has been practiced at a very low level throughout most of the world, especially in the United States and Europe. One of the major reasons for this is that these countries have traditionally been what could be considered agricultural countries, with relatively little interest in fish consumption, and therefore, fish farming or aquaculture. Another reason for the low level of aquaculture to date in the western nations is that the people have had fastidious tastes and therefore, rarely eat other than a very small portion of the numerous edible seafood types available.

In contrast to this, the people of Japan consume a greater variety of seafood than any other country in the world. Accordingly, they also have the highest degree of aquaculture developed to date.

Closed Water Aquaculture:
Aquaculture in closed waters is a costly business resulting in a costly product. In order to be advantageous it must usually be conducted by producing food to be bought with a minimum amount of processing and transportation, or to produce such high

demand products as oysters, mussels, clams or shrimp, which the customer is willing to pay for the expenses incurred. In these cases large-scale controlled aquaculture can be achieved in restricted lagoons and estuaries, where the migratory fish, as well as the fertilization of the water with essential nutrients, can be retained.

Offshore Pumping to Onshore Ponds:
In this method of aquaculture, cold deep offshore water is pumped up through large diameter pipes, and then run into lagoons, tanks or ponds where it will stimulate the photosynthetic process by providing the necessary nutrients. The resulting phytoplankton can then be utilized by organisms higher up the trophic chain. Such a study was conducted in 1969 by O.A. Roels, R.D. Gerard, and A.W.H. Be, of the Lamont-Doherty Geological Observatory of Columbia University (Costlow, 1971). This study was conducted at an experimental station at St. Croix, where water was pumped to shore from a mile offshore where the water depth was approximately 1000 meters.

The results of this study showed that at least 35 microgram-atoms nitrate nitrogen per liter seawater were present at the nutrient maximum at the St. Croix station. They assumed approximately 65% of the nitrogen content in the water was in the form of nitrate nitrogen and therefore, assuming a 10% efficiency of conversion of phytoplankton to a high priced secondary consumer, they could produce 3.3 tons of secondary producer per year in 5,000 sq.ft.

Offshore Pumping and Open Sea Aquaculture:
This is the type of aquaculture which is basically under study in this project. By this method water is pumped up from well below the surface (approximately 1000 ft. in our study) and then dispersed in the upper layer of water penetrated by the sunlight, where the artificially upwelled nutrients may be utilized in an area of high photosynthesis.

Due to technological advances this form of aquaculture is the most recent to be experimented with and therefore, the least amount of reference exists for this as compared to the other types of aquaculture.

UTILIZATION OF AQUACULTURE WITH THE SOLAR SEA POWER PLANT

Applicable Forms of Aquaculture:
As can be seen in prior sections of this report, the general nature of the Solar Sea Power Plant would be an artificial upwelling of deep nutrient laden water, and therefore, either offshore pumping to onshore ponds or to open sea could be utilized. However, in this case, where the prototype plant would be located approximately sixteen (16) miles off of the coast of Miami, Florida, offshore pumping and open sea aquaculture is the type chosen as the most beneficial to study.

The rate of primary production is known to be highly variable. John H. Ryther (1969) states that nearly 90% of the ocean is essentially a biological desert. The open oceans produce a negligible fraction of the world's fish catch at present, and have little or no potential for yielding more in the future. Upwelling regions, on the other hand, which total no more than approximately one-tenth of one percent (0.001) of the ocean surface produce about half of the world's fish supply. The other half is produced primarily in coastal waters and the few offshore regions of comparably high fertility.

Photosynthesis is carried on mainly in the upper layer of solar light penetration where single cell organisms are produced, whereby larger marine growths are produced, step by step according to different trophic levels. The larger the plant cells at the beginning of the food chain, the fewer the trophic levels that are required to convert the organic matter into useful form. When this life dies the remains settle slowly to the ocean depths where they break down and disintegrate. The residues of these remains then are carried, both in solution and as particles back to various areas of upwelling. In these upwelling areas the food chains are potentially shorter than in non-upwelling areas as the organisms produced are large enough to be directly utilized by man from trophic levels very near the primary producers. This is due to the fact that the phytoplankton in these upwelling regions are of large size and that many of these species are colonial in habit.

For example, the eight most abundant species of phytoplankton in the upwelling region off Peru (one of the most fertile upwelling regions of the world) are: *Chaetoceros socialis*, *C. debilis*, *C. lorenzianus*, *Skeletonema costatum*, *Nitzshia seriata*, *N. delicatissima*, *Schroederella delicatula*, and *Asterionella japonica* (Reeve, 1968).

Much of these phytoplankton can be readily eaten by large fishes without special feeding adaptation. Also many of the clupeoid fishes (sardines, anchovies, etc.) have specially modified gill rakers for removing the larger species of phytoplankton from the water. Therefore, there exists a one or two step food chain between phytoplankton and man in most areas of upwelling, as compared with at least a three step food chain between phytoplankton and man in typical coastal waters.

The already observed advantages in areas of natural upwelling seem to point out the great potential in the study of artificial upwelling as would be seen in the utilization of aquaculture with the solar sea power plant under study. With the shortages of food, as well as energy, which are seen today, such a system may not provide the final answers, but would definately be concentrating efforts in the right direction.

Major Factors of Open Sea Aquaculture:

One of the major considerations for the utilization of an optimal aquaculture venture as a by-product of the solar sea power plant is the level at which the nutrient laden water brought up from approximately 1000 ft. is discharged. The maximum depth in most areas to which sunlight penetrates for the production of phytoplankton and zooplankton is roughly 300-350 feet. In our prototype, the discharge level has been established at approximately 270 feet below the surface. Therefore, the utilization of the artificially upwelled waters will be less than optimal.

Another major consideration would be the economic feasibility of conducting an aquaculture project at an offshore power plant. Due to the extremely high costs of both installation and maintenance it would be impractical to have containments which are customary to normal methods of aquaculture. These containments, such as pens, nets, or rafts, would require too much capital expenditure to be considered for use in this side by-product of the actual solar sea power plant.

Rather than having a confined area of certain forms of aquaculture, the power plant could provide areas of approximately concentric shape whereby an optimal zone would be established at which biological life could prosper in proper conditions (i.e. temperature, salinity, dissolved oxygen, effects of biofouling, predators, and areas have the highest amounts of usable nutrients). The exact levels of these various parameters would be dependent upon the specific combinations produced on the existing ambient conditions with the addition of the discharged deep water.

The types of products which could exist and be utilized by the artificial upwelling of the solar sea power plant under study would be those which best utilize the diatoms present around such a system and then in turn be beneficial to man with the least amount of trophic chain manipulations. Such creatures would be shrimp, sardines, and possibly anchovies. The possibility of oysters was studied, but rejected as they would require physical equipment such as rafts, and could be produced more efficiently closer to shore in warmer waters.

Since our system would not have any confinements for those products being raised, the problem of disease control would be minimized. Those animals not strong enough to fight the current could not remain in the high productivity zone surrounding the solar sea power plant. The larvae and other very small creatures produced by the adult size organisms would also be carried out of our system, but other adult size organisms would be attracted and therefore, a fairly stable ecosystem could be established.

The problem presented by periodic plant shut down for crew charge and maintenance should not present any serious detrimental effects to the proposed aquaculture system because the amount of down time is not sufficiently long to cause break up of the ecosystem. The system would have to be down for a period of time closer to a week in order to do substantial damage to the dependent ecosystem. If such a long shutdown period was to occur, it is very possible that fish would be effected by cold shock which in turn could inhibit feeding and predator avoidance or even cause death directly.

Although it is felt that the solar sea power plant under study would provide a system which would induce an artificial form of upwelling in which there would be increased biological productivity with a minimum amount of contamination, it is possible that the system, acting as an attractant to fishes and other animals, could present other problems. If impingement at the intake and entrainment of small organisms into the cooling system is density-dependent, the effect of attraction of biological activity could prove to be detrimental. However, since the nutrients, which will be brought up from the deep water by artificial upwelling, must be in the upper layer of water for a period of time before being utilized by higher trophic chains, and because the power plant is located in a relatively strong current; it is felt that the zone of highest productivity and increased biological activity, and thus the highest concentration of fish, will be in an area away from the plant itself.

Other factors must also be considered in the system under study. Although the artificial upwelling will cause a substantial increase in the amount of the principal nutrients nitrogen, phosphorus, and carbon, and thus, provide a richer habitat for diatoms, the changed water temperature may cause a shift in species composition favoring less desirable green and blue-green algae. The actual degree to which this type of algae would flourish would again depend upon the actual location sight of the solar sea power plant. In the case of our model, being located approximately sixteen miles off of the coast of Miami, Florida, with the deep cold water brought up from upwelling being discharged at a temperature of approximately 50°F, this should not present any significant problem.

CONCLUSIONS

The presence of the solar sea power plant, located off of the coast of Miami, Florida, should provide a beneficial environment for increased biological activity for the trophic chains discussed. Since, in our case, the current velocities are significant, and the direction is relatively stable, an area will be created which will attract fish and other forage animals. The plume will provide a more constant and suitable area for growth, especially during the colder months of the Fall and Winter.

It is noted that production is not equivalent to potential harvest since it will also be eaten by other top-level carnivores, besides man, such as birds, squid, tuna, and other natural predators.

Since no confining equipment, such as nets and pens, would be used, there would be no cost of maintenance, and not only would such a system provide a source of fish protein for human consumption, but it would provide a good indicator to be utilized to insure the level at which the solar sea power plant effects the immediate environment.

This type of noncontained aquaculture should provide an area of substantially increased biological production at a minimal cost, and therefore, be considered advantageous to the installation and design of a solar sea power plant, such as the one under study.

REFERENCES

Behrman, Daniel. The New World of the Oceans. Boston: Little, Brown and Company. 1970.

Chapman, W.M. Potential Resources of the Ocean. Serial Publication 89-21, 89th Congress, Government Printing Office, Washington, D.C. 1965.

Costlow, John D., Jr. Fertility of the Sea. Vol. 2. New York: Gordon and Breach Science Publishers. 1971.

Duxbury, Aly C. The Earth and its Oceans. Reading Massachusetts: Addison-Wesley Publishing Company. 1971.

Eppley, Richard, et al. Biological Aspects of Offshore Nuclear Power Plants. Power Siting Workshop, Rockville, Maryland. 1973.

Firth, Frank E. The Encyclopedia of Marine Resources. New York: Van Nostrand Reinhold Company. 1972.

Hull, Seabrook. The Bountiful Sea. Englewood Cliffs, New Jersey: Prentice Hall, Inc. 1969.

Othmer, Donald F., and Roels, Oswald A. Power, Fresh Water, and Food from Cold, Deep Sea Water. Science 182: 121-125.

Ryther, John H. Photosynthesis and Fish Production in the Sea. Science 166: 72-76, October, 1969.

Reeve, M.R. Symposium Marine Food Chains. Aarhul, 1968.