Gulf-Stream-Based, Ocean-Thermal Power Plants

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This paper presents the results of an ongoing analytical study at the Univ. of Mass. for the design of major components and a total power system for Gulf-Stream-based ocean-thermal power of 400 Megawatts electrical net power output. On the basis of these studies, the ultimate power potential of a 15 mile wide by 550 miles long length of the Gulf-Stream extending from south of Miami, Fla. to Charleston, S.C. is estimated to be approximately 2 trillion kilowatt hours per year which could be transmitted to shore by undersea cables. Critical subsystems and components (such as heat exchangers, ocean-based hulls, and cold water inlet pipe) are identified, and the technical basis for their configuration and design is discussed. 90/10 copper-nickel alloy plate-fin heat exchangers, with propane flowing upward (evaporators) or downward (condensers) through small passages in the plates and sea water flowing horizontally between the plates, have been selected. The latest power system (Mark II) is based on a submerged, twin catamaran concrete hull configuration with hulls approximately 80 ft in diameter by 800 ft long. The evaporators are staggered serially in height in 6 tiers placed above the twin hulls which contain the condensers, turbines, pump, and other power cycle components. A cold-water inlet pipe of elliptical cross section (1500 ft long and with a hydraulic diameter of 87 ft) is hinged between the hulls with a gun-buckler type joint. Technical problems facing the deployment of such plants are summarized, and the latest cost estimates predict busbar power costs of approximately 15 mills per kilowatt hour.

Introduction

RECENT events involving the United States' supply of energy have instigated a series of long range energy policy studies, such as the Project Independence Study completed under the Federal Energy Administration.[1] As pointed out by the Solar Energy task force group of this study,[2] Ocean Thermal Energy Conversion (OTEC) has the long range potential to supply a significant amount of our nation's future energy needs. Such OTEC power plants, specifically designed to produce power for the Eastern coast, have been under investigation at the Univ. of Mass. This study supported by the National Science Foundation RANN Directorate, with the objective of determining the technical feasibility of the concept, and identifying those subsystems and components critical to the concept.[3-6] Such work defined a first total-system concept, the Mark I, shown in Fig. 1. More recent work has been concentrated on the definition and design of major subsystems and components for offshore power plants up to 400 Megawatts electrical net situated in the Gulf-Stream off the Eastern U.S. coast. In this paper the design of such systems, conceived as a float, semisubmerged, anchored, and as close as 15 miles to land, is discussed in detail.

Siting of ocean thermal power plants is by no means limited, however, to use of the Gulf Stream resource. Other studies supported by NSF[7-9] have and are considering many other sites, such as those in the tropical oceans. The tropical high seas will require considerably smaller and less expensive power plants per unit of productivity than will those in the Gulf Stream off the U.S. southeast coast. However such siting will also require seaborne transport of some "Transportable" energy product other than electricity or pipeline gas. The key reason for our concentration on Gulf Stream sites is the conviction that those near-shore waters have a large potential for practice of this natural solar driven process. This resource size may be approximated[10] by assuming an average Gulf-Stream flow rate of 33×10^6 cubic meters per second and an overall 1°F reduction in temperature along a 550-mile length (Fig. 2). This yields an available heat energy input to the power system of 690 trillion kilowatt hour per year. Using a realistic assumption of an average 1.5% overall energy conversion efficiency (including load factor) converts this resource to useful work at a rate of 10 trillion kilowatt hour per year. One estimate[11] of predicted U.S. electrical generation places the total at 3 trillion kilowatt hours per year in 1980. Siting studies suggest that as many as 550 power plants rated at 400 megawatts electrical each with an expected annual plant factor of 0.93, could be placed "in line" along this Stream. Such a line would convert 1.8 trillion kilowatt hours per year. There is a width of about 15 miles in which such lines of power plants could be located. Therefore it appears quite possible that the entire available 10 trillion kilowatt hours per hour could be extracted from this one rather small deployment site. Furthermore, oceanographic data indicate that this supply is reasonably constant. This eliminates the need for large energy storage subsystems common to the vast majority of other solar energy conversion systems which stop producing power when the sun sets. Thus, OTEC plants are able to serve easily as base load plants. Further work[12] underway, at the Univ. of Mass., detailing the exact size and yearly output potential of the Gulf Stream may change these estimates. The 10 trillion kilowatt hours per year is thought to be a good starting point for further discussion.

Other potential advantages to the siting of OTEC plants in the Gulf Stream are:

1) This resource is close to the U.S. coast, and, at any point along the proposed deployment area, electrical power can be transmitted to land via conventional undersea power cables.

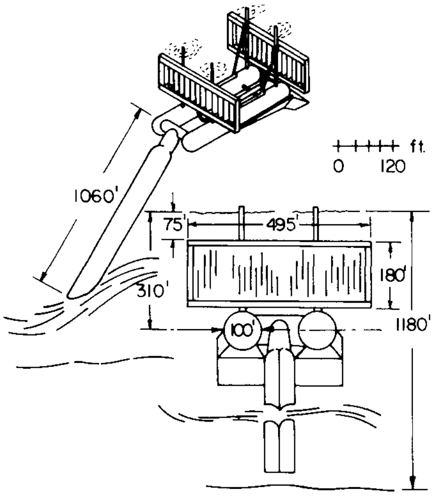

Fig. 1 Schematic of Mark I Gulf-Stream OTEC plant.

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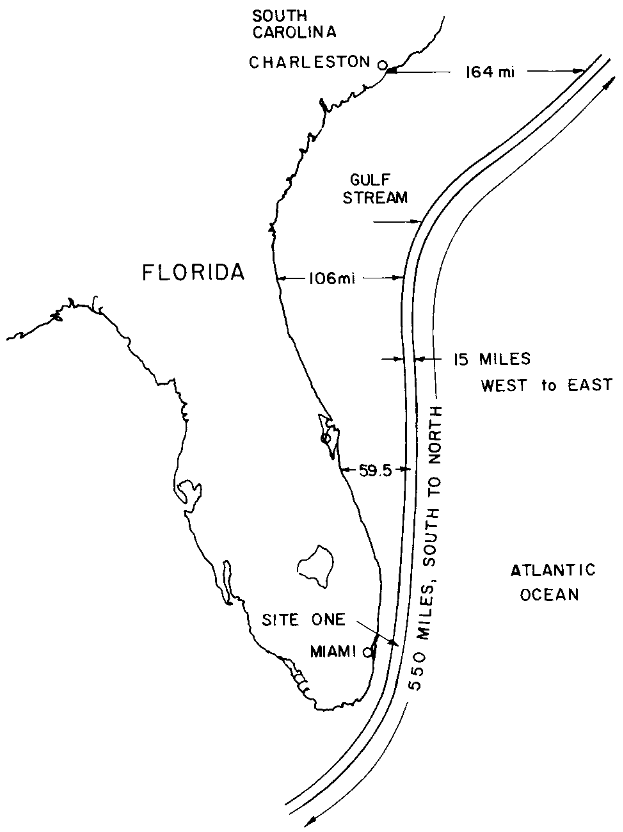

Fig. 2 Suggested Gulf-Stream sites.

There is also the possibility of completing an in-sea-bed pipeline hydrogen energy link to the entire Eastern seaboard. Although the planned deployment area is in international waters, rights to the resource are clearly established by existing U.S. policy regarding sovereignty over resources in and over our continental shelf and slope, to the extent that they can be exploited.

2) The Gulf Stream flows in this area with natural velocities up to 6.5 fps, and that velocity can be used to advantage. It has been shown that biofouling reaches a near-zero minimum in ocean water moving in excess of 2.5 fps. The flowing water could also be used as a natural pumping system on the evaporator side of the cycle. Another advantage is that the continued replenishment of the hot and cold water resource to a Gulf Stream plant allows the plant to be stationary.

3) There is a considerable body of scientific data pertaining to the characteristics of the Gulf Stream in this region, which can be used to advantage to specifically design the proposed systems.

On the other hand, a Gulf Stream site has some disadvantages, foremost of which is that, compared to sites nearer the Equator, the lower available temperature difference places some important thermal design disadvantages on the system. For example, a key parameter in this system is the size of the heat exchangers. Table 1 shows that a change in site ΔT from 32° to 45°F causes over a fourfold reduction in heat exchanger material volume. The 32°F site temperature difference is representative of the present design Site No. 1, (Fig. 2) which is 15 miles east of Miami.

Table 1 Effect of site conditions on heat exchanger material volume requirements for 400 MWe plant

Site	ΔT (°F)	Heat exchanger material requirements (ft^3)
Gulf Stream ($T_H = 77$°F, $T_c = 45$°F)	32	81,000
Tropical ($T_H = 82$°F, $T_c = 45$°F)	37	48,3000
Tropical ($T_H = 85$°F, $T_c = 40$°F)	45	24,9000

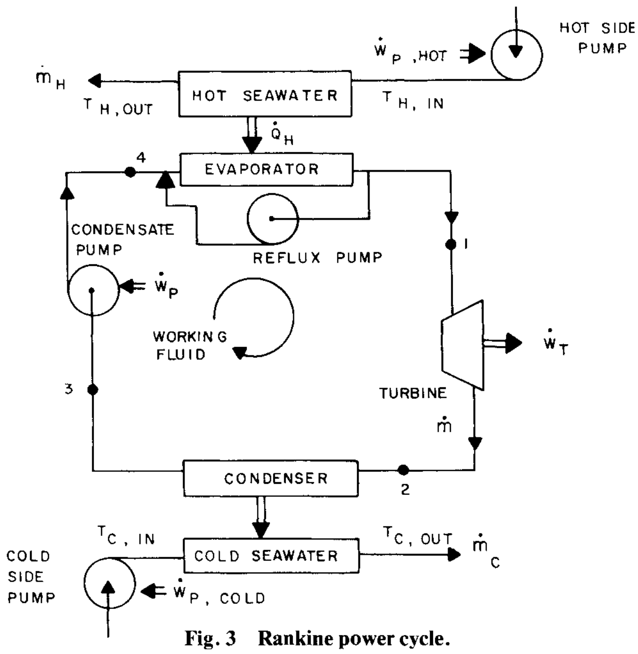

Fig. 3 Rankine power cycle.

Thermal Power System

The favored closed Rankine cycle is shown schematically in Fig. 3. It includes an evaporator recirculation pump subsystem to reduce evaporator size for a given output, and to facilitate off-design cycle control. The cycle working fluid now under consideration for the Gulf-Stream-based system is propane. It should be noted that ammonia is a superior working fluid on the basis of thermodynamic and heat-transfer properties, requiring smaller working fluid flow rates (and inventory), heat exchangers, and turbines. However, propane has been chosen chiefly on the basis of its compatibility with all the candidate materials for heat exchangers, including the selected 90/10 copper-nickel alloy. Propane also does not need the complex working-fluid-clean-up system that an ammonia system would require.

Because of the small thermodynamic driving potentials involved with a Rankine power cycle operating in the Gulf Stream, the analytical details of a complete system (especially the heat exchangers) become fairly involved, and require detailed mathematical models.[4,6] Although it is beyond the scope of this paper to consider the details of these analytical models, Table 2 summarizes the effects, for a typical cycle design, of various losses on an important design parameter, total heat exchanger size. Furthermore, the large number of variable cycle and component design parameters require a comprehensive parametric or optimization study before the physical size of a complete power package can be made. Such studies have been carried out and form the basis of the designs proposed in this paper.

As previously mentioned, the heat exchangers form a most important part of the thermal power cycle. Although the first system design (Mark I) featured banks of staggered tubes for

Table 2 Effect of cycle losses on heat exchanger size

Cycle configuration	A/A_{ideal} (cumulative)
1) ideal Ranine cycle (1°FΔTon each exchanger)	1
2) Add 90% turbine	1.12
3) Add 85% cycle pump	1.13
4) Add 5% pressure drop in evaporator	1.43
5) Add 5% pressure drop in condenser	2.09
6) Add 10% cold side pumping work	2.38
7) Add 10% hot side pumping work	2.79

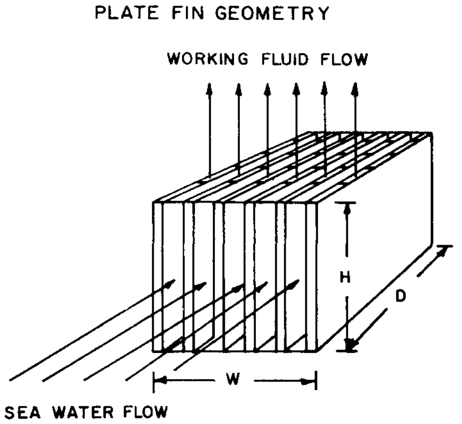

Fig. 4 Plate-fin heat exchanger geometry.

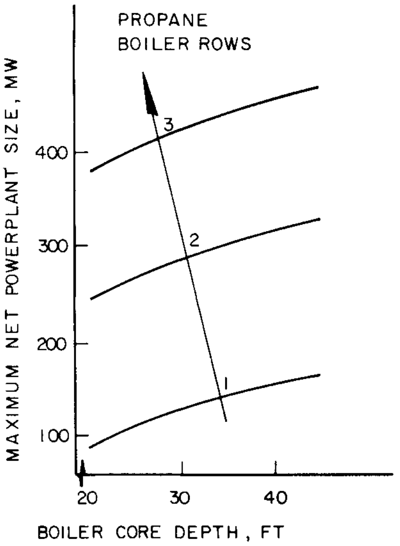

Fig. 5 Evaporator limitations of naturally-pumped system.

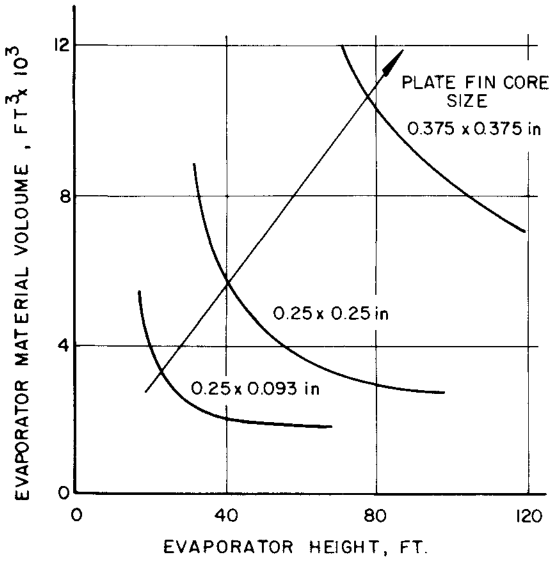

Fig. 6 Effect of passage size on plate-fin exchanger size.

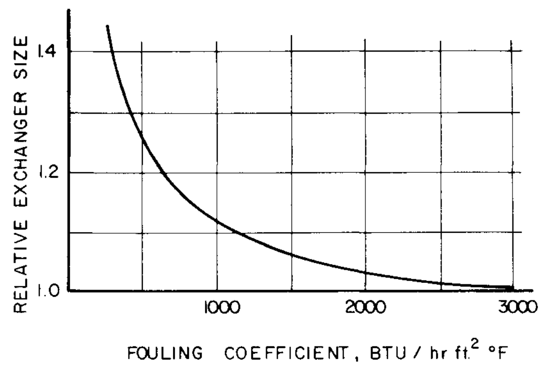

Fig. 7 Effect of fouling on heat exchanger size.

the evaporator and conventional shell-and-tube heat exchangers for the condensers, the latest designs are based on the use of plate-fin type heat exchangers, as shown in Fig. 4. The primary mode of heat-transfer on the working fluid side is forced convection (evaporation or condensation), and the working fluid flows upward in the evaporator and downward in the condenser. Due to the heat-transfer characteristics of the working fluid (even with small working fluid side spacing as low as ⅛ in. and internal fins for area augmentation) this side dominates the overall heat-transfer coefficient. Thus, the ocean side does not require fin augmentation and relatively large flow openings (with approximately ½-in. spacing) can be used on the ocean side. Obviously a few spacers will be needed on this side for mechanical assembly, but, in general, the ocean flow side on both the evaporators and condensers should be easy to clean, if necessary.

One major design option involves the choice between naturally pumped evaporators, using the kinetic energy of the Gulf Stream, and forced pumped evaporators. Based on the results of an analytical study modeling the flowfield of the Gulf Stream[13] the latest design feature banks of evaporators fitted with sea water pumps. Figure 5 shows that a major reason for this choice was the power plant size limitations imposed by natural pumping, restricted to only one "row" of evaporators across the flow.

The size of the working fluid passage involves another major design choice. Figure 6 shows that smaller passages are desirable from a heat exchanger materials requirement viewpoint (because of thinner wall requirements for pressure-proof exchangers and improved heat-transfer characteristics). It is also desirable to use smaller passages giving lower height evaporators (heights over 100 ft are undesirable because of higher temperature variations in the Gulf Stream and are also awkward to arrange) and minimum volume condensers (requiring smaller containment hulls). However, the smaller size passages (and wall thicknesses) will require more innovative manufacturing processes than would be the case with larger passages.

As previously mentioned, the desire to keep 90/10 copper-nickel, alloy an active candidate heat exchanger material, prompted the choice of propane as a working fluid. This alloy is expected to be inherently fouling-free in this application. The potential effects of ocean water fouling on both of the exchangers is shown in Fig. 7; fouling could easily require a 25% increase in heat exchanger size. In our OTEC plant design, provision has also been made for either trickle or batch treatment with chlorine to avoid fouling. The turbines and auxiliary fluid systems, also have been investigated in some detail.[14,15] Turbine size optimization has been the key factor in the power module sizing. Both the Mark I and Mark II baseline configurations have been based on the use of sixteen 25-megawatt electrical net power packages (32.5 megawatt electrical gross power per package).

Power Plant Configuration

The latest configuration, the Mark II design, is shown in model form in Fig. 8. This arrangement evolved from the earlier Mark I design, and is a 400 megawatt electrical-net-power-output power plant comprised of 16 identical 25-megawatt electrical-net power packages. A turbine and condenser are provided for each power package inside the hulls: a total of 18 evaporator modules (2 spares) are "shared" by the turbines.

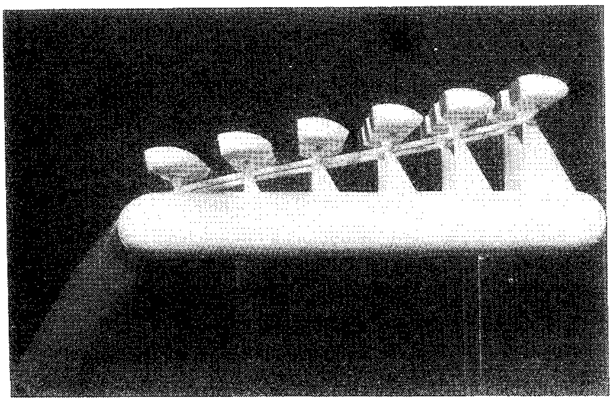

Fig. 8 Model of Mark II system.

The catamaran hull design uses reinforced-concrete, horizontal cylinders capped by hemispherical heads, and subdivided by flat bulkheads. The latest internal layout studies yielded an internal diameter of 80 ft. The naval architecture of the system must satisfy both submarine and normal surface ship requirements, and capabilities. Major submarine capabilities include the possession of submerged neutral buoyancy, and adequate main ballast system that will permit change from a surfaced condition to the neutrally buoyant submerged condition, and an adequate variable ballast system that will adjust to changes in the variable force vectors. Evaluation of the Gulf Stream data at Site 1 shows that daily variations in current induced drag are on the same order of magnitude as seasonal variations. Based on this information, the variable ballast system is able to compensate for current variations of up to 20% (corresponding to drag variations of about 40%). The depth of submergence of the power plant (nominally 220 ft below the sea surface) will be tuned to surface sea conditions, with considerable added depth available when hurricanes are predicted.

There have been 3 major internal layout variations of the Mark II design, each involving different circulating cold water paths. These in turn have influenced the shape and nature of the pressure hulls, the external shape of the power plant hull, and the type of connection between the hull and the cold water inlet pipe (CWP). In all of these arrangements 3 fu ental requirements have been met:

1) There are two "skins" or closures between the circulating water and the machinery compartments.

2) There is internal subdivision with bulkheads capable of holding to pressure hull collapse depth.

3) Any condenser core can be removed from an isolated condenser box, and a replacement core can be moved the length of the hull into the compartment in which required. (This requirement has the strongest influence on the diameter of the pressure hulls.)

A model of the second internal variation is depicted in Fig. 9. In this design the CWP was split about 100 ft forward of the hulls, and half of the circulating water was led up into the front of each hull via a gun-buckler-type swing joint. The flow was then directed outward into conical circulating water feed sponsons on each side of a hull, tapering away from the bow to the stern. Each condenser box took its circulating water from the sponsons via pressure-proof circular cross-section supply ducts. The cold water passed through each condenser box and then overboard through discharge ducts slanting down through the bottom of the cylindrical hull. The total of turns in the flow path starting from the bottom of the inlet pipes equaled 285°, giving a significant reduction in hydraulic losses from the initial inline flow path of the Mark I configuration. However, the circulating water supply and discharge ducts within the cylindrical pressure hulls constituted a doubling up of pressure-proof boundary at considerable expense that would not improve water-tight integrity, or damage control capability of the system.

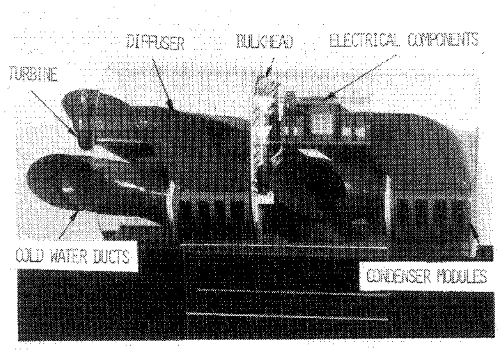

Fig. 9 Internal hull system (Mark II-Mod. 2).

The third internal arrangement scheme, presently most favored, places the CWP between the pressure hulls, terminating at the hulls with a gun-buckler-type swing joint which allows the circulating water to flow axially into the space between the hulls. From this pressure balanced flow space, circulating cold water enters each condenser box. The total of the turns in any one flow path is reduced to 225°, plus whatever discharge vector angle is used for the discharge from the hull. With this arrangement, studies suggest that a pressure hull i.d. of 80 ft, and a compartment length of 100 ft will be required. It also appears that there should be a pressure-proof bulkhead between each 2 power packages in line, and that (as shown in Fig. 9) the turbines and diffusers should be arranged to overlap so the most favorable turbine diffuser geometry can be obtained. Rather large diffusers have been specified because it is expected that they will add as much as 2% to turbine efficiency.

As shown in Figs. 8 and 10, the evaporators are tiered on the pressure hulls. Each evaporator module is divided into units with one hot-water pump taking a suction from the almost square discharge face of that unit. Each evaporator is built into its own structural framework, capable of holding it in place on the hull before submergence. Once the evaporators are fully immersed, each evaporator can be treated as a separate, neutrally buoyant, structurally independent unit, connected by valves and piping to both the liquid working fluid distribution mains, and the vapor collection headers. Any one unit can be isolated, broken free, surfaced, and towed away for repairs. It will also be possible to accomplish on-site evaporator repairs by surfacing the power plant to expose the bottom of the tier in which the faulty unit is located. The latter method may cause undue loss in power plant operating time if fault lies in the lower tiers. The support structure (concrete-encased steel) between the tops of the pressure hulls and the undersides of the evaporator units is rather massive because of its need to carry all of the in-air weight of the evaporators during construction, and prior to being dived on site.

Because gross fouling cannot be tolerated, the following control methods have been given consideration:

a) Use of sea water velocities in excess of 2.5 fps.

b) Use of inherently fouling-free material (90/10 copper-nickel alloy) on sea water sides.

c) Trickle-dose chlorination using some of the produced electricity to generate chlorine from sea water. The power drain to do this is appreciable but acceptable (assuming that 1/10 ppm dosage will suffice).

d) Soaking type chlorination.

e) Soaking with near-boiling sea water, prepared at the expense of some product electricity.

f) Mechanical brushing.

g) Any combination of 2 or more of the above.

The production-line biofouling control system will probably be chosen as a result of full-scale heat exchanger trials in situ.

Before mentioning the mechanical design aspects of the cold water pipe, it is important to note that this component,

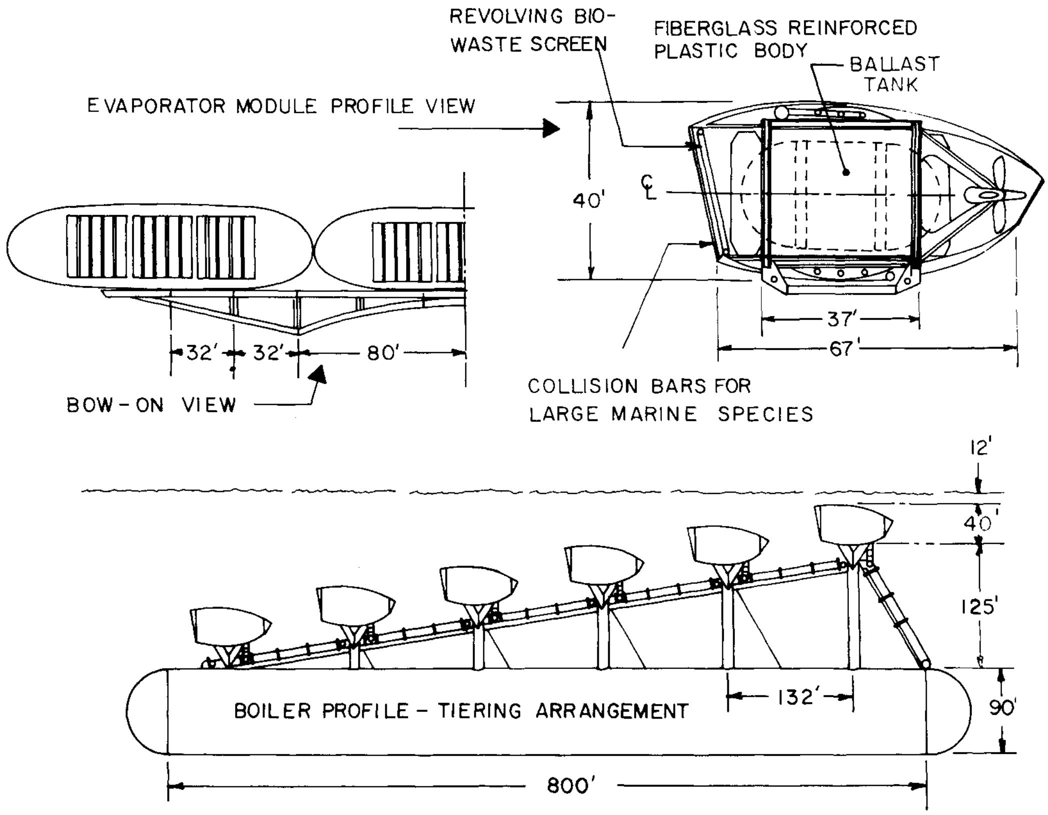

Fig. 10 Mark II evaporator layout.

because of its effects on the cold water parasitic pumping losses, can exert a sizable influence on the thermal cycle component requirements. This point is illustrated in Fig. 11, which shows relative heat exchanger requirements for a Mark II design plant (with a 1500-ft long CWP) as a function of CWP hydraulic diameter and internal sea water velocity. A small reduction in heat exchanger size is achieved once the internal hydraulic diameter exceeds about 80 ft. A hydraulic diameter of 86.5 ft (elliptical cross section with 95-ft major and 80-ft minor diameters) is presently used with a specified cold water flow velocity of 9 fps. This limiting value of internal velocity has been selected, based on the use of aluminum for the cold tube construction material.

The space between the cylindrical hulls in the third configuration version becomes a combination of circulating water delivery passage, main ballast tank, and free flooding void. Some consideration has been given to a design that would house the CWP inside this space for the journey to the operating site, then slide it out the front when on site. At the same time the gun buckler joint is being pulled and locked in the essential structure required to transmit mooring loadings. This might be possible for a 1000 ft long cold water pipe, though it might eliminate much of the main ballast tank structure that could be carried therein. It would certainly be possible for "telescoped" cold water inlet pipes.

Within each of the machinery spaces, on decks arranged in the upper compartments outboard of the turbine diffuser, there is ample room for the electrical equipment needed for either a dc electricity-in-cable umbilical, or for a complete hydrogen umbilical, using conventional electrolyzer components. In both cases, the turbines turn rotating ac machines and all of the power is rectified to dc . A 3-bus arrangement of electrical conductors has been used, and start-up/emergency diesel generators have been provided. If the power plant is to use the dc electricity umbilical, the rectifiers may be concentrated at the forward ends of the pressure hulls, so that the cable may go down the CWP. From here it is led down the tether to the anchor, and then the ashore cable. With a dc umbilical, sea return is placed, and the "electrode" required at the hull has been designed as a vertically suspended bare cable about 200 ft in length. For distances from shore less than 15-18 miles, it may be more economical to transmit the power directly as ac, with a 3-cable system. This eliminates the need for ac-dc, and dc-ac rectification.

Conclusions

Our work on the design of Gulf-Stream-based ocean thermal power plants is continuing and it is expected that future designs will reflect subsystem, component, and total system improvements. Obviously, economics dictate an important part in any power system study, and as Table 3 illustrates, we believe that the cost of such plants based on our current designs, would be no higher than $800/per kilowatt energy yielding a power generation cost of 15 mills/kilowatt hour. Furthermore, we believe[16] that such systems are economically competitive with conventional and nuclear-fueled power plants.

There are 2 key problem areas which must be totally overcome or the concept, regardless of ocean site, will fail. These are:

1) Corrosion: these power plants must survive in an ocean environment. They must exhibit longer maintenance free life

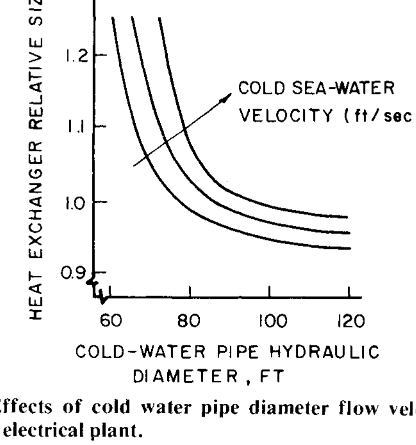

Fig. 11 Effects of cold water pipe diameter flow velocity for 400 Megawatts electrical plant.

Table 3 Estimated cost for Gulf Stream plant in dollars per kilowatt electrical

Component description	Cost ($/kWe)
Hull and concrete components	39
Cold water inlet pipe	63
Machinery	179
Heat exchangers	340
Electrical and transmission	82
Command and control	1
Auxiliary/life support	5
Outfit and furnishings	3
Sub-total	712
Present value of construction with interest and escalation	797[a]

[a] At a fixed charge rate of 15.5% and a plant factor of 0.93, busbar power cost would be of the order of 15 mills per kilowatt hour.

than have most ocean systems in the past, and this must be achieved at a reasonable cost. If materials, such as aluminum, are used, some corrosion may be acceptable, but the reduced capital costs of such material must offset the increase in maintenance costs. Under no circumstance, however, can situations involving dissimilar metals coupled in seawater be tolerated.

2) Biofouling: the warm waters of the Gulf Stream contain populations of fouling creatures; the cold water from near the sea bed will probably be relatively free from such populations. In both cases, the heat exchanger surfaces simply cannot tolerate biofouling encrustation; such growth will effectively stop the power plant. Nor can the hull or other exposed parts of the power plant accomodate excessive marine growth. Fouling must always be controllable and controlled.

We believe that the designs we have proposed for Gulf Stream sited plants can effectively overcome these problems. However, experimental work in the second of those areas is very much needed: those biofouling populations must be known and be quantified. Furthermore, our work has pointed out the following problem areas in which improved technical solutions must be sought:

1) While the necessary heat exchanger design theory has been developed, a breakthrough in working fluid side heat-transfer augmentation would give the economics of the process a significant boost. Also, low-cost fabrication techniques for the large heat exchangers must be demonstrated.

2) The turbines should achieve efficiencies of at least 90%. Our analytical studies indicate this is possible, particularly if power module size is dictated by optimum turbine size.

3) The supply of both hot and cold water at a site without disruption of the natural thermal layering of the ocean at that site must be achieved. Our analytical studies indicate that, with proper system design, this is possible for the Gulf Stream site.

4) The cold water pipe and connecting duct system required to convey the huge quantities of cold water and, at the same time, induce a minimum of hydraulic losses in the process, appears to be a major structural problem. Many, widely varying solutions have been suggested and need study.

5) The mooring and anchor system for a power plant located in the swift waters of the Gulf Stream will be larger and heavier than any ever used before. The probable motions and acceleration of these power plants in the ocean excited by wind-waves (including hurricanes) and currents must be understood in advance of hull design, and in advance of mooring system design.

6) These power plants offer an opportunity for large-scale use of steel reinforced concrete in the ocean, particularly for the submerged pressure hulls.

7) The energy umbilicals, whether for electricity or gaseous hydrogen will be large and expensive.

Practical, economic Ocean Thermal Power Plants can be designed and developed for the Gulf Stream site from the baseline created during the past 2½ yr. No breakthroughs are required. If some practical method of augmenting heat-transfer on the working fluid side can be demonstrated, it could make these power plants even more competitive, and would be a tremendous addition to the technology. But such a breakthrough is not mandatory to proceeding with development of this concept.

The sola ergy resource available to the contiguous 48 states via cable or pipeline using the ocean thermal differences process is huge. These power plants built by the hundreds and deployed in the Gulf Stream starting in as few as 6 yr could be the single most effective action this country could take to set right our energy industry, which is the key to all our industry. The idea of betting on systems with overall thermal efficiencies as low as 1.5 to 2%, is anathema to many of our power industry engineers. They must change their minds and join this effort. Our energy salvation lies only in our ability to put solar energy to work for us as soon as we can. We believe the Gulf Stream is the place to start, followed directly thereafter by tropical high-seas fleets. Steady, imaginative, shrewd engineering, step-by-step development, testing, rialing are all that need. A large-scale national program should be started at once.

References

[1] Federal Energy Administration, *Project Independence Report,* U.S. Government Printing Office, Nov. 1974.

[2] Federal Energy Administration, *"Project Independence Task Force Report-Solar Energy,"* U.S. Government Printing Office, Nov. 1974.

[3] McGowan, J.G., Connell, J.W., Ambs, L.L., and Gross, W.P., "Conceptual Design of a Rankine Cycle Powered by the Ocean Thermal Difference," *Proceedings of the Eighth Intersociety Energy Conversion Engineering Conference*, AIAA, pp. 420-427, 1973.

[4] McGowan, J.G., Heronemus, W.E., Connell, J.W. and Cloutier, P.D., "Ocean Thermal Difference Power Plant Design," *Journal of Engineering for Industry,* Vol. 96B, Nov. 1974, No. 4, pp. 1119-1129.

[5] Heroneumus, W.E., et al., "Research Applied to Ocean Sited Power Plants, Progress Report NSF/RANN/SE/GI-34979/PR/73/3, NTIS, PB-228067, Jan. 1973, Univ. of Mass., Amherst Mass.

[6] McGowan, J.G. and Connell, J.W., "Heat Exchanges Design for Ocean Thermal Difference Power Plants," *Journal of Engineering for Industry,* Vol. 97B, No. 3, Aug. 1975, pp. 1035-1045.

[7] Lavi, A., "Final Report: Solar Sea Power Project," Report: NF/RANN/SE/GI-39114/PR/74/6, Jan. 1975, Carnegie Mellon University, Pittsburgh, Pa.

[8] Douglass, R.H. and Bakstad, P., "Systems Aspects of Ocean Thermal Energy Conversion," AIAA Paper 75-615, April 1975, Los Angeles, Calif.

[9] Trimble, L.C. and Messinger, B., "Ocean Thermal Energy Conversion System Evaluation," AIAA Paper 75-616, April 1975, Los Angeles, Calif.

[10] Schmitz, W.J. and Richardson, W.S., "On the Transport of the Florida Current," *Transactions of the American Geophysical Union,* April 16, 1967, 48th Annual Meeting Proceedings, pp. 76-86.

[11] Dupree, W.G. and West, J.A., "United States Energy Needs Through the Year 200," U.S. Department of the Interior, Dec. 1972, Washington D.C.

[12] Kirchhoff, R.H., Magarella, P.A., McGowan, J.G., "Hydrodynamics of Gulf Stream Ocean Thermal Power Plants," ASME Winter Annual Meeting Paper, Dec. 1975, in *Ocean Engineering Mechanics,* 1975, pp. 1-27.

[13] Kirchhoff, R.H., McGowan, J.G., Connell, J.W., Seluk, D., "Hot Side Heat Exchangers for an Ocean Thermal Difference Power Plant," *Proceedings of Ninth Intersociety Engergy Conversion Engineering Conference,* American Society of Mechanical Engineers, pp. 354-361.

[14] Ambs, L.L. and Marshall, J., "Ocean Thermal Difference Power Plant Turbine Design," Technical Report NSF/RANN/SE/GI-34979/TR/73/17, Nov. 1973, University of Massachusetts, Amherst, Mass.

[15] Ambs, L.L. and Veenema, R.J., "Power Turbines for Ocean Thermal Energy Conversion Systems," ASME Winter Annual Meeting Paper, Dec. 1975, ASME Paper No. 75-WA/OCE-11, Houston, Texas.

[16] McGowan, J.G. and Heroneumus, W.E., "Ocean Thermal and Windpower: Alternative Energy Sources Based on Natural Solar Collection,"*Environmental Affairs*, Vol. 4., No. 4, Fall 1975, pp. 629-660.