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STUDIES OF BIOFOULING IN OCEAN THERMAL ENERGY CONVERSION PLANTS

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ABSTRACT

Efforts to extract energy from the ocean's thermal gradients by means of closed-cycle Ocean Thermal Energy Conversion (OTEC) plants require very large heat exchangers. The seawater passing through these will be heated (or cooled) by only a few degrees. Under these conditions it is feared that biological fouling (biofouling) may seriously impede heat transfer unless appropriate measures are taken. There exists surprisingly little data on biofouling under conditions approximating those expected to exist in an OTEC heat exchanger. For these reasons we have undertaken a study of biofouling in simulated OTEC heat exchangers. Currently, the effect of fouling on the heat transfer coefficient is being investigated as a function of the material used and water velocity. Next, the effectiveness of several means of biofouling prevention will be determined. Later stages of the study will include investigation of variation with a number of other phenomena important to OTEC design and operation.

I INTRODUCTION

In recent years there has been growing interest in the extraction of energy from the ocean's thermal gradients by means of Ocean Thermal Energy Conversion (OTEC) plants. This increasing interest is due mainly to two causes. First, the efforts of a number of groups, especially of the pioneers Anderson, Heronemus, and Zener, have resulted in the delineation and solution of many of the critical problems, leading to detailed designs and cost estimates[1] indicating technical, and probable economic feasibility. Second, the rapidly rising costs of traditional energy sources has increased the intensity of searches of alternatives.

Most OTEC work is centered on closed-cycle systems in which heat exchangers are used to transfer heat between seawater and a working fluid. Because of the inherently low thermal efficiency of the system, large quantities of seawater must pass through the heat exchangers, which must therefore be very large and correspondingly costly. Indeed, the heat exchangers are expected to represent almost half the cost of

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the entire plant. This being so, it is clearly critical to ensure that the heat transfer coefficient is as large as possible.

A potentially serious problem is that when seawater bathes any surface, biological fouling (biofouling) is likely to occur, seriously degrading the heat transfer properties of the system. The severity of the effects of biofouling and the likelihood of avoiding or mitigating these effects at reasonable cost cannot at present be known with confidence. A program of research is needed to answer these questions.

It is well known that biofouling is sensitive to local conditions. The research program must therefore be designed to cover a sufficient range of all important variables to include most likely OTEC operating conditions. In particular, it is desirable at least to study fouling under varying conditions of location, depth, flow velocity, heat exchanger material, temperature change, pressure change, and antifouling method.

We have embarked on a program of research designed ultimately to look at all these parameters, and more. Here we wish to describe only our initial efforts, and to present some very preliminary results.

II INITIAL STUDIES

We have designed an "experimental unit" the essence of which is a simulated heat exchanger tube 1" in diameter and 8-1/2' in length. Attached to the tube is apparatus permitting the pumping of seawater through it at controlled velocities ranging up to at least 10 feet per second. Also attached is instrumentation permitting the measurement, at will, of water velocity and water temperature, as well as of the heat transfer coefficient between the tube wall and the flowing sea water. Since biofouling in OTEC heat exchangers is of importance primarily because of its effect on the heat transfer coefficient, we have designed the instrument to permit continual monitoring of this coefficient as a measure of fouling.

The unit is designed so that the tube itself may be removed and replaced by another of the same or a different material. With minor modification, a tube of different diameter and length may be used.

We have chosen to do our first studies at a location off the island of Hawaii near waters where conditions are good for the location of a plant. In these studies a battery of six units will be attached to a subsurface buoy about sixty feet below the surface. In practice, water from such depths would be passed through the evaporator of an OTEC plant, and so we are studying the problem of biofouling in the evaporator. (The water passing through the condenser of an OTEC plant would come from great depth, where biological activity is much less. However, other kinds of fouling may be severe in the condenser. We expect to study fouling in deep waters at a later stage.)

The first observations will be made on tubes of copper-nickel, aluminum, and titanium alloys. In these tubes the intensity of biofouling will be measured vs. water velocity. These observations can be done with the tube wall at temperatures ranging from 0-5°F above the water temperature (using an electric heater wound around the test section). The effect on fouling of gross variations of pressure change will be determined by changing the pump location from downstream to upstream of the tube. Manipulation of valves in the circuit will provide some finer degree of variation. After the principal features of the biofouling problem have been delineated by these initial studies, the other effects discussed in Sec. I will be pursued.

Concurrently with these measurements, various types of data will be taken on the surrounding ocean conditions such as temperature, salinity, dissolved oxygen, nutrient concentration, biomass, etc. Attempts will be made to correlate these data with the observed tube fouling.

III APPARATUS

The geometry of a test unit is shown in Fig. 1. The "1 in. TUBE" is the test subject. The heat transfer coefficient from the tube wall to the seawater flowing through it is measured using instrumentation attached to the "Cu Block" as described below. The flowmeter is a Ramapo type Mark V - 1-1/4-SFY, which determines the water velocity by measuring the force on a target immersed in the flow. To protect the instrumentation from seawater, it is contained in a housing constructed of standard schedule 80 6" PVC pipe with associated flanges and a tee. The seawater seals were carefully designed and tested to be vacuum tight. The housing is filled with dry nitrogen to prevent condensation which affects the instrumentation.

In these experiments we are faced with the necessity to measure heat transfer coefficients under rather confining conditions. The apparatus is remote, submerged many feet deep in the ocean for long periods, and inaccessible for weeks, months or longer. For one thing, this means that we are unable to calibrate the thermometers or power meters routinely. Another important constraint is that the temperature difference (ΔT) between the tube wall and the water must be kept small if the biota being studied are not to be affected by the measurement. In order to mitigate these difficulties, we have developed a novel method of measuring h which requires no measurement of power, and no calibration of the thermometer. The method is capable of high precision. It is described below.

Figure 2 shows the complete unit with attached pump (inside the housing). The pump is positioned downstream of the test unit initially, so that the potential fouling organisms are not subjected to the stresses of passing through the pump before going through the test section.

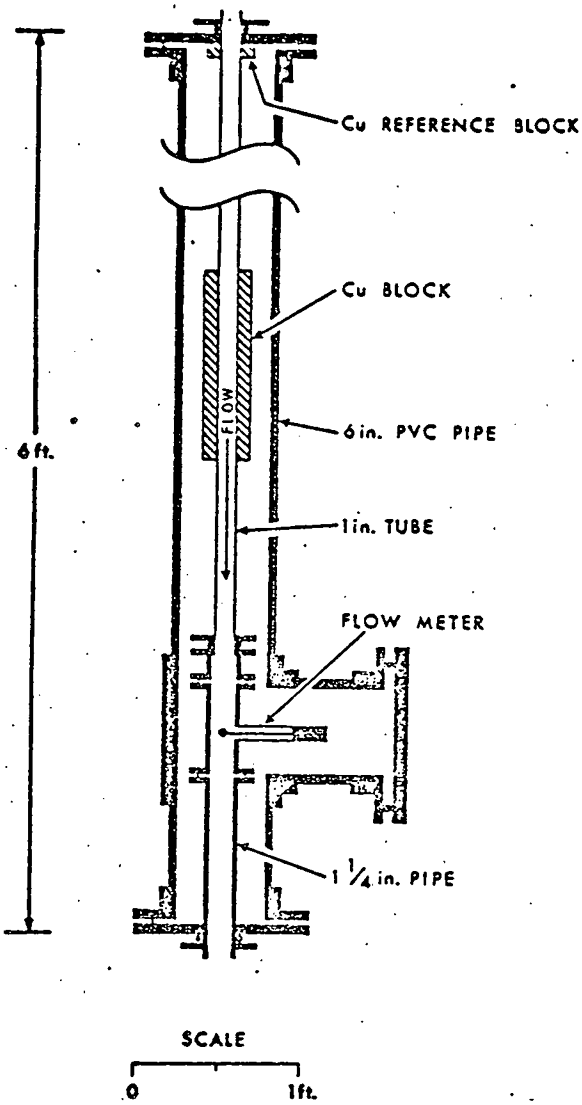

Figure 1. Schematic of test unit.

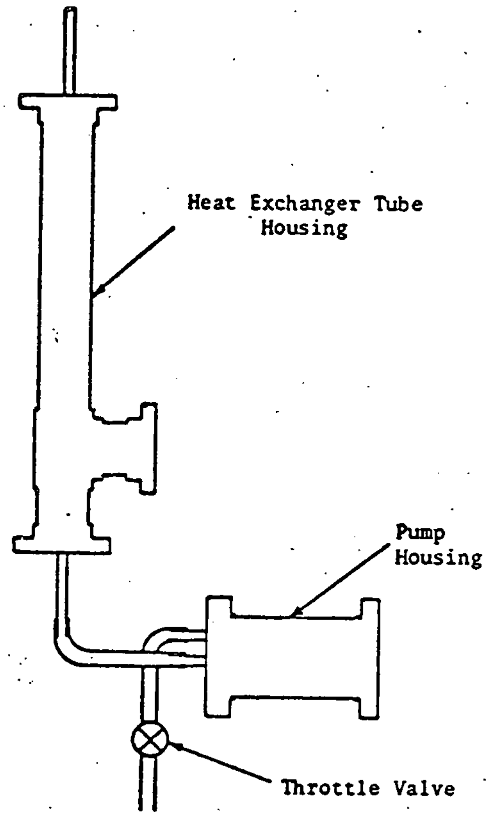

Figure 2. Test unit showing pump attached.

The pump chosen is the Eastern Model MDH-25 which has a capacity sufficient to allow us to study fouling at velocities over ten feet per second in the one inch tube. It is a plastic-impeller, magnetic-drive pump which should not suffer any corrosion problems. The throttle valve is adjusted to control the flow velocity.

Figure 3 shows how up to six test units may be attached to a submarine buoy at a depth of about 60 feet. Submarine cables (underwater connectable) connect the instrumentation in each unit to vapor-tight

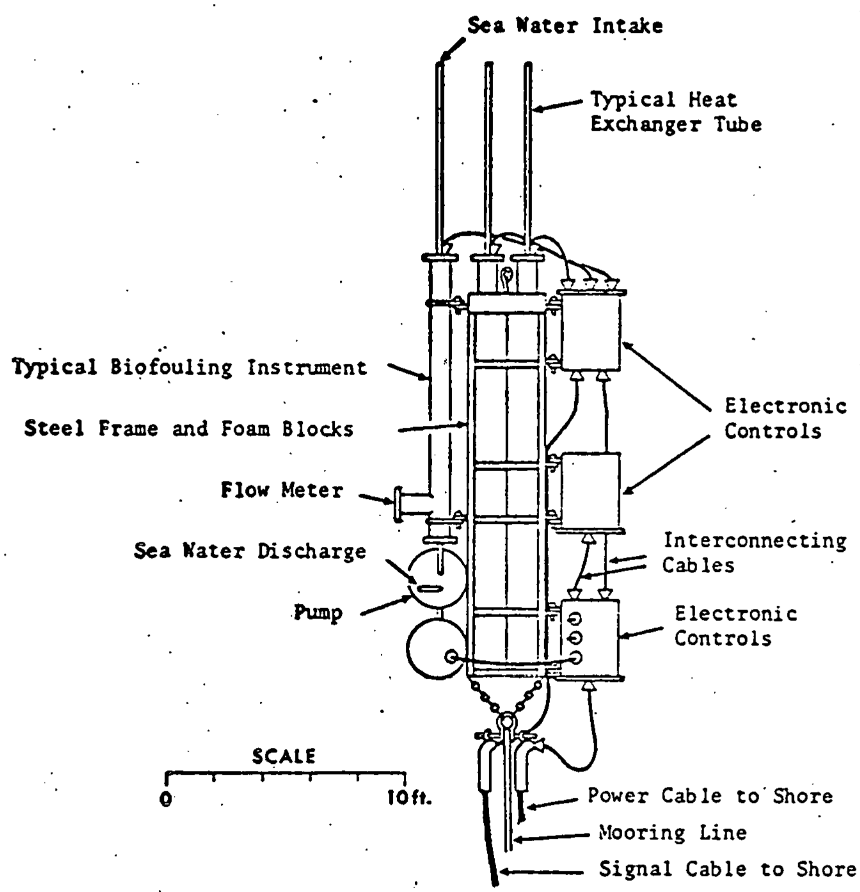

Figure 3. Test units and electronics attached to submarine buoy.

electronics boxes, which in turn are connected to the beach by means of a 1000 foot three-wire submarine power cable and a signal cable containing six shielding pairs.

The submarine electronics boxes contain power supplies, control electronics and multiplexing electronics to allow us to run the experiments remotely from the beach. The system is designed to permit us, at will, to turn off any one or all pumps, to turn on and off the copper-block heater, at either of two power levels, in any unit, and to sense simultaneously, the copper-block temperature, the flow velocity, the heater power, and the seawater temperature in any unit. In addition, we can measure any of a number of other parameters associated with antifouling equipment to be added later.

In Fig. 4 is shown the mooring arrangement at Keahole Point off the Kona coast of Hawaii.

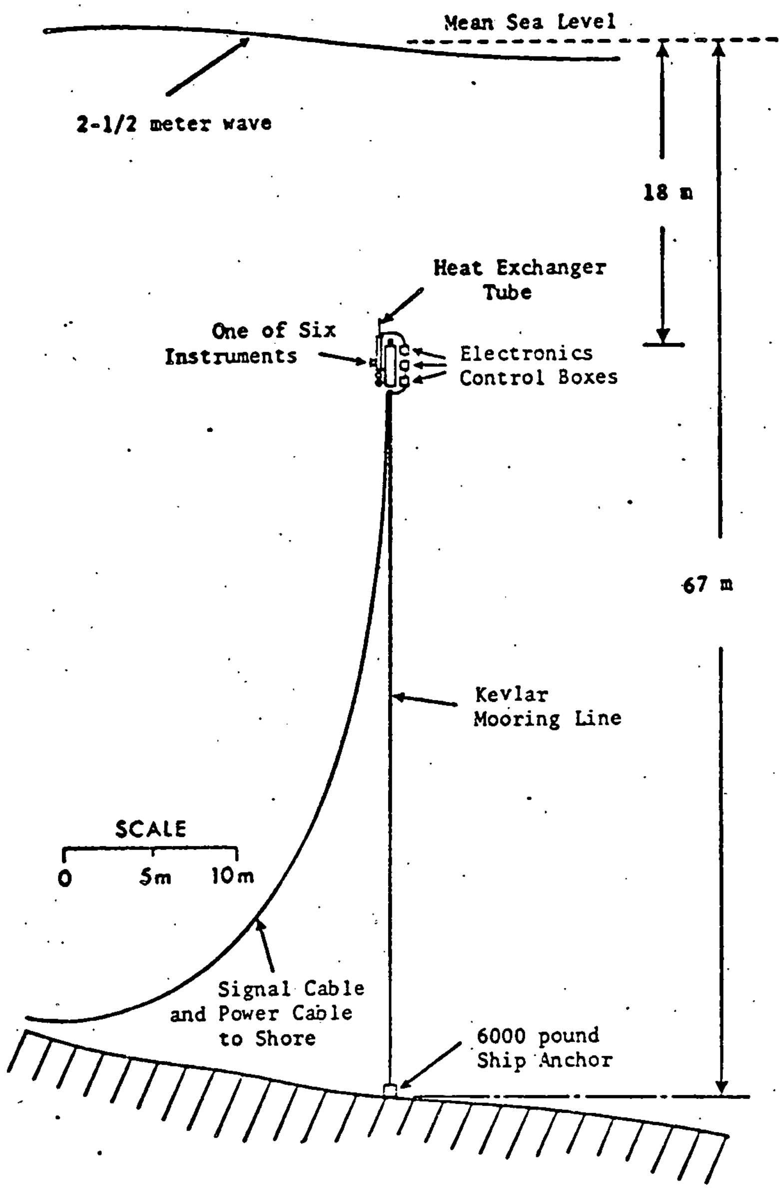

Figure 4. Mooring system.

The apparatus has been designed so that a given experiment (e.g., a chosen combination of tube allow, flow velocity and fouling inhibitor) can be continued uninterrupted for months or even years. At any time during such an experiment, we may monitor the degree of fouling without disturbing the conditions of flow. When an experiment is to be terminated, a diver will disconnect the unit from the buoy and swim it to the surface (it is made neutrally buoyant). It is a simple matter to remove the tube from the test unit (for subsequent biological and corrosion studies) and to replace it with another. The unit is then swum down to the buoy and reconnected.

IV THEORY

Consider the physical situation in which a fluid (water in our case) is flowing through a heat exchanger tube whose inner wall is at a temperature T above that of the bulk fluid. Because of this temperature difference, heat will pass through the inner wall of the tube (surface area A) at a rate which is given by:

$$\frac{dQ}{dt} = -hAT .$$

The magnitude of h is determined both by the properties of the laminar layer of the fluid and by the extent to which the inner surface of the tube is fouled.

Suppose that the tube wall thickness is standard except for a test section where the wall thickness is significantly increased, as shown in Fig. 5. Further, suppose that heat is being supplied by some unspecified source to this thick-walled section which has inner diameter D and length L. Then, under steady state conditions, heat is passed from the thick-walled section along three paths. The first and dominant path is through the inner surface of area πDL to the flowing water, the second is through the thin-walled sections of the tube to the flowing water, and the third is through the outer surface of the thick-walled section to the surrounding medium (air). If at time t=0 the heat source is cut, then the heat which had been stored in the thick-walled section will flow out along the three paths. The rate of this flow is sensitively related to h. This relationship can be found under the following three conditions which define the ideal case:

1) The thermal conductivity (k) of the tube material is infinite;
2) The heat flow through the thin-walled section of the tube is zero;
3) The heat flow through the outer surface of the thick-walled section is zero.

Given conditions 2 and 3, all heat must flow radially from the thick-walled section to the flowing water. The dimensionality of the problem is thereby reduced from three to one. Condition 1 implies that the temperature of the entire thick-walled section is uniform. Now if T(t) is the temperature difference between the wall and the flowing water at some time t, then the heat content of the thick-walled section is given by:

$$Q(t) = CT(t) ,$$

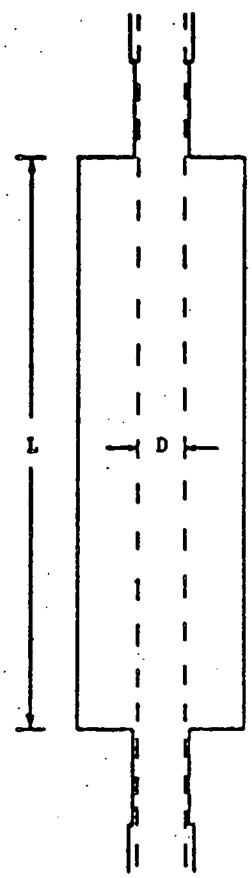

Figure 5. Thick-walled section of the tube.

where C is the total heat capacity of the thick-walled section. After the heat is cut off at t=0, the rate of heat loss from the thick-walled section is determined only by the rate of heat transfer to the water, since no other mechanism is permitted by conditions 2 and 3. Thus,

$$\frac{dQ(t)}{dt} = C\,\frac{dT(t)}{dt} = -\pi DLhT(t)\ .$$

The solution of this equation is:

$$T(t) = T_o e^{-t/\tau}\ . \qquad (1)$$

Here T_o is the steady state temperature at t=0 and τ is the time constant for temperature decay which is given by:

$$\tau = \frac{C}{\pi DLh} \qquad (2)$$

This time constant, therefore, provides a measure of the heat transfer coefficient.

The above analysis is quite useful in illustrating the method and in developing a physical feeling for the problem. However, in order to make the model more realistic, conditions 1-3 must be replaced by more realistic ones. When this is done, the solution for temperature vs. time is found to be much more complex than that given in Eq. 1. However, it is found to reduce to the form of Eq. 1 after a brief (<1 sec. in our geometry) transient period. The relationship between τ and h is also found to be slightly different from that given in Eq. 2. The true relationship, for a geometry with cylindrical symmetry, is easily calculated by means of a computer program written for the purpose.

V LABORATORY TESTS

Before using the method described above for monitoring h in field studies, we carefully tested it, and the apparatus, in controlled laboratory conditions.

The experimental apparatus is diagrammed in Fig. 6. A solid copper cylinder was "shrunk fit" onto a 90% Cu 10% Ni heat exchanger tube. This technique was chosen for the first unit in order to realize the cylindrical symmetry assumed in the theory. (This requirement was relaxed somewhat for subsequent copper cylinders which were designed to clamp onto the tube and be demountable.) The copper cylinder thus forms the "thick-walled section" to which frequent reference was made in the previous section.

A nichrome heater with a resistance of 52 Ω was wound around the outside of the copper cylinder. The temperature difference between a central location in the copper cylinder and the flowing water is measured by means of a thermopile consisting of 11 iron-constantan thermocouples. The sensitivity of the thermopile is 0.314 mV/°F, and temperature differences were measured with a precision of ±0.001°F. The reference junctions of the thermopile are epoxied into a copper cylinder which is in good thermal contact with the flowing water, while the temperature-sensing junctions are epoxied into the copper heater cylinder ("thick-walled section"). These sensing junctions are located midway between the inner and outer radii of the copper heater cylinder. A power dissipation of 124 watts in the nichrome heater windings resulted in a temperature rise at the thermocouple locations of 2.1°F above that of the water which was flowing with a velocity of 7.0 ft/sec. Measurements were made at flow velocities ranging from 2.0 to 8.4 ft/sec.

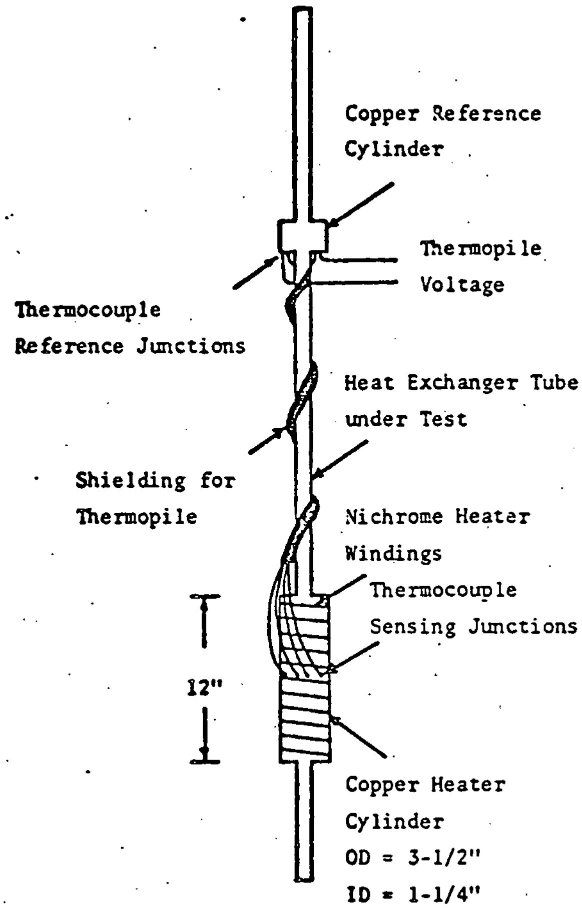

Figure 6. Tube with instrumentation.

A typical decay curve of amplified thermopile voltage as a function of time is plotted on a semi-log scale in Fig. 7. These data were taken at a flow velocity of 5.06 ft/sec. After a short time the decay becomes highly linear and remains so for a time in excess of four time constants. In this region the relation between thermopile voltage (V_{TC}) and time (t) is given by:

$$V_{TC}(t) = V_{TC}(0)e^{-t/\tau} .$$

The data are properly weighted and then are fitted to the above expression in order to extract the time constant (τ). From this fit a value of τ is obtained with an estimated uncertainty of -0.5%.

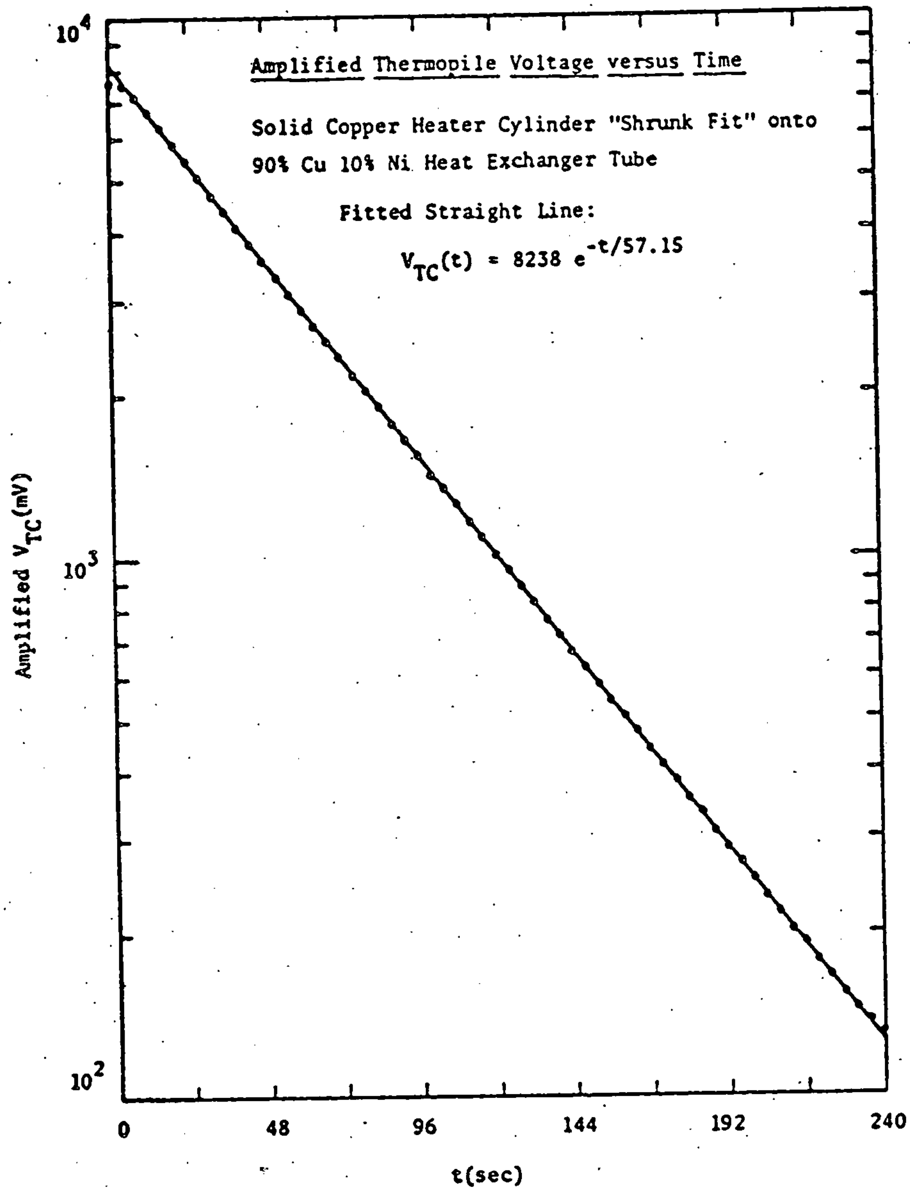

Figure 7. Decay of temperature (thermopile voltage) with time.

Note that in this procedure no power measurements are made and the thermopile need not be calibrated. The only assumption regarding the thermopile is that its output voltage varies linearly with temperature difference and that its calibration, whatever it may be, remains constant only for the duration of a single decay (~10 min).

The results of the theoretical analysis of the decay permit the conversion of the measured time constant to a heat transfer coefficient. The overall precision to which h can be measured is estimated to be better than 1%.

Conventional wisdom tells us that the heat transfer coefficient for the laminar layer of flowing water varies as $v^{0.8}$. This result follows from a combination of dimensional analysis, correlations of existing data, and analogy with theoretically tractable situations.[2] In the case where the fluid is water flowing inside clean tubes, the physical properties of the fluid are lumped into a temperature dependent term, and h is related by v by:[3]

$$h = \frac{160(1+0.012\ T_f)}{D^{0.2}} v^{0.8} \tag{3}$$

where D is the inner diameter of the tube and T_f is the film, or laminar layer temperature. Thus, the thermal resistance due to the laminar layer of the water (1/h) should vary as $1/v^{0.8}$.

A plot of 1/h versus $1/v^{0.8}$ for v from 2.0 to 8.4 ft/sec is given in Fig. 8. This set of data is representative of several sets which were taken on the 90 Cu 10 Ni heat exchanger tube. A linear least squares fit to the data yielded an intercept of $(0.4143 \pm 0.0048) \times 10^{-3}$ and a slope of $(3.634 \pm 0.014) \times 10^{-3}$. The data were taken at temperatures near 70°F, and for purposes of comparison were normalized to 70°F by the temperature dependent factor of Eq. 3.

The non-zero intercept of the 1/h axis in Fig. 8 is due to a thermal resistance between the thermocouple locations and the turbulent region of the flowing water which is independent of flow velocity. Since the inside of the heat exchanger tube was cleaned prior to taking the data, this is interpreted as being due to a thermal contact resistance between the "shrunk fit" copper heater cylinder and the outer wall of the heat exchanger tube.

The exponent of v was also permitted to be a parameter in fitting the data of Fig. 8. The fitted value for the exponent was found to be 0.79 ± 0.02.

VI PRESENT STATUS AND PRELIMINARY RESULTS

At present four of the six units (Figs. 1 and 2) have been assembled and tested. The last two should be ready in September 1976. The electronics, submarine electronics housings and the subsurface buoy (Fig. 3), as well as the mooring line and anchor (Fig. 4) are in hand. We are now waiting for the delivery of the signal and power cable (Fig. 4) to be used to connect the system to the power source and control electronics on the beach. We are hopeful that the cable

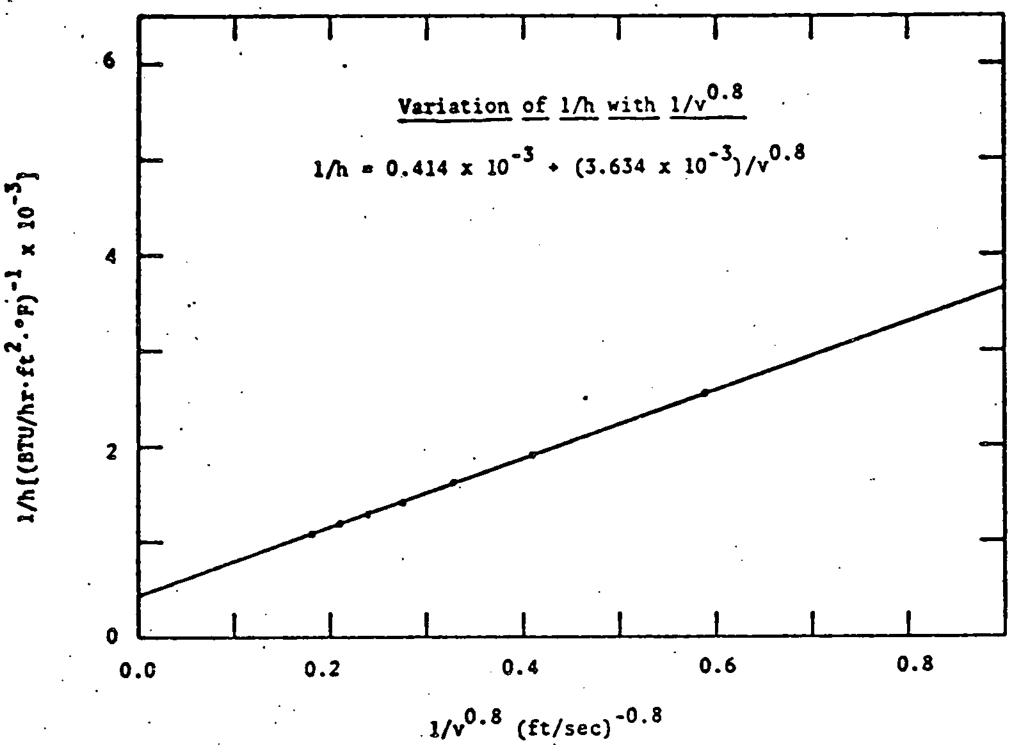

Figure 8. Dependence of the heat transfer coefficient on flow velocity.

will be in hand by September. All the beach hardware (fenced-in shelter, power generator and electronics) are on hand and have been tested.

While waiting for the cable, we have tested most of the system components and are taking preliminary data. This is being done by using two units (numbers 2 and 3) mounted on a boat. Each unit is connected to a pump (downstream) and a 2" reinforced PVC flexible hose (upstream). The other end of the hose hangs overboard in the sea and allows water to be pumped through the test unit from depths up to about 50 feet.

Data are not yet available from units 2 and 3. However, unit 1 has been operating for some weeks in a similar fashion from the state pier at Makapuu Point, Oahu. This unit contains a 1" diameter 9-10 Cu-Ni tube. It has had seawater running through it continuously since March 7 (except for a four day period in late March). Most of this time was spent in testing the system and getting some bugs out of the electronics. The water velocity was usually in the range 4-5 feet per second, except occasionally when it was varied up to 10 feet per second for tests.

Some recent results are shown in Fig. 9.

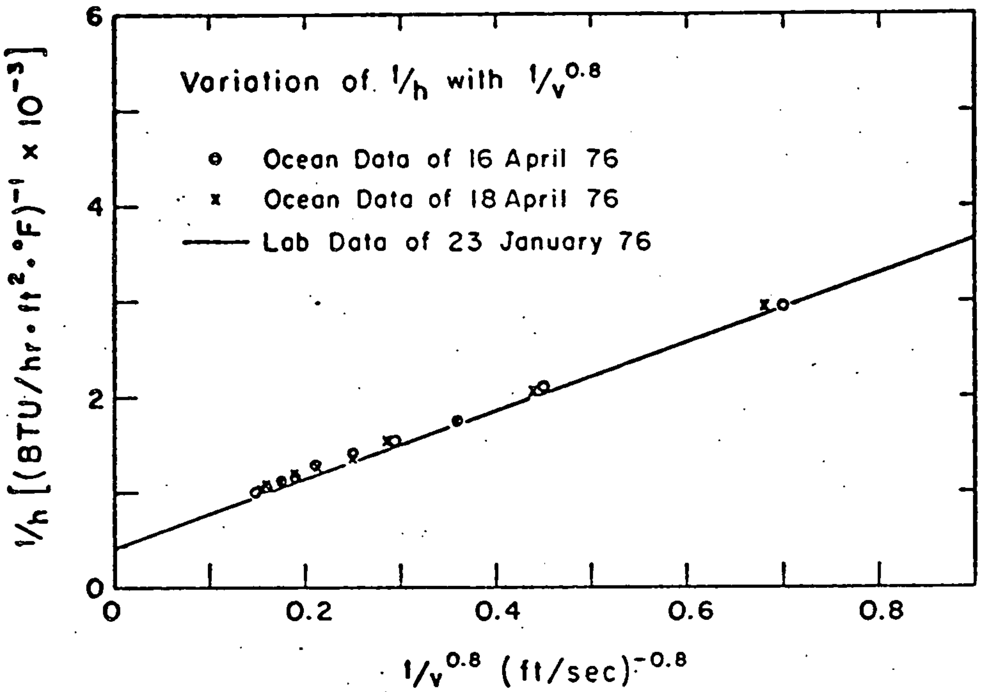

Figure 9. Variation of 1/h with $1/v^{0.8}$.

The experimental points represent data taken on April 16 and 18 when h was measured as a function of water velocity. If all is well, we should get a straight line when we plot 1/h vs. $v^{-0.8}$. That we do get the theoretically expected line with a clean tube was confirmed with careful measurements in the laboratory. The laboratory data (from Fig. 8) are represented by the curve in Fig. 9.

The data indicate that there has been a slight decrease in h corresponding to a fouling factor of (13±2) x 10^3 BTU/hr ft^2°F. The error on this figure was determined from the rms deviation of the experimental points from the best fit straight line.

Caution should be exercised in accepting these preliminary results. The data must be carefully checked before confidence is placed in them.

VII ACKNOWLEDGMENTS

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REFERENCES

1. "Ocean Thermal Energy Conversion, Research on an Engineering Evaluation and Test Program," TRW Systems Group, One Space Park, Redondo Beach, Ca. 90278; "Ocean Thermal Energy Conversion (OTEC), Power Plant Technical and Economic Feasibility," Ocean Systems, Lockheed Missiles and Space Co., Inc.

2. Heat Transmission, William G. McAdams, 3rd ed., McGraw-Hill, 1954 (p. 219).

3. Standard Handbook for Mechanical Engineers, Theodore Banmeister and Lionel S. Marks, McGraw-Hill, 1966 (p. 4-100).