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TITLE: STUDIES OF BIOFOULING IN OCEAN THERMAL ENERGY CONVERSION PLANTS

AUTHORS: J. G. Fetkovich
Carnegie-Mellon University
Professor of Physics
5000 Forbes Avenue
Pittsburgh, Pennsylvania 15213
(412) 621-2600 Ext. 484
Visitor of SNAME

G. N. Grannemann
Carnegie-Mellon University
Research Physicist
5000 Forbes Avenue
Pittsburgh, Pennsylvania 15213
(412) 621-2600 Ext. 8797
Visitor of SNAME

L. M. Mahalingam
Carnegie-Mellon University
Research Physicist
5000 Forbes Avenue
Pittsburgh, Pennsylvania 15213
(412) 621-2600 Ext. 360
Visitor of SNAME

D. L. Meier
Carnegie-Mellon University
Research Physicist
5000 Forbes Avenue
Pittsburgh, Pennsylvania 15213
(412) 621-2600 Ext. 360
Visitor of SNAME

F. C. Munchmeyer
University of Hawaii
Associate Professor of Mechanical Engineering
2540 Dole Street
Honolulu, Hawaii
(808) 948-7585
Member of SNAME

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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.

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 indicating technical, and probable economic feasibility (1). Second, the rapidly rising costs of traditional energy sources has increased the intensity of searches for 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 present almost half the cost of 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.

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 seawater. 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 as a function of 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 the INTRODUCTION 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.

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 might affect the instrumentation.

In these experiments we are faced with the necessity of measuring 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

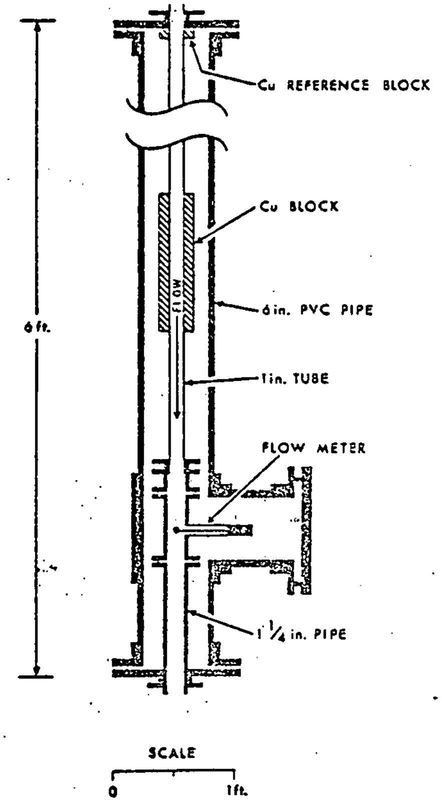

Figure 1. Schematic of test unit.

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 in the THEORY section of this paper.

Fig. 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.

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.

Fig. 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 electronics boxes, which in turn are connected to the beach by means of a 1500 foot submarine cable. This cable contains three power lines, three

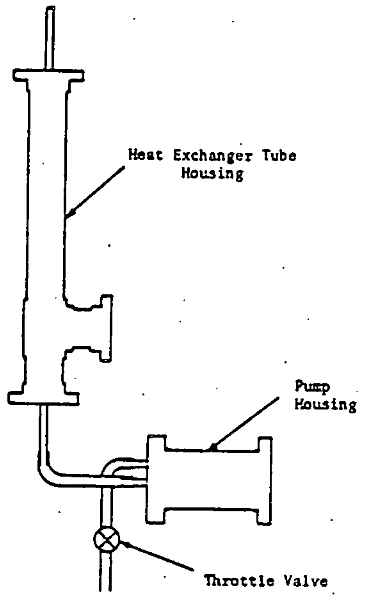

Figure 2. Test unit showing pump attached.

shielded pairs of wires for digital data transmission and three unshielded pairs for digital control signals.

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.

The appratus has been designed so that a given experiment (e.g., a chosen combination of tube alloy, 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.

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:

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

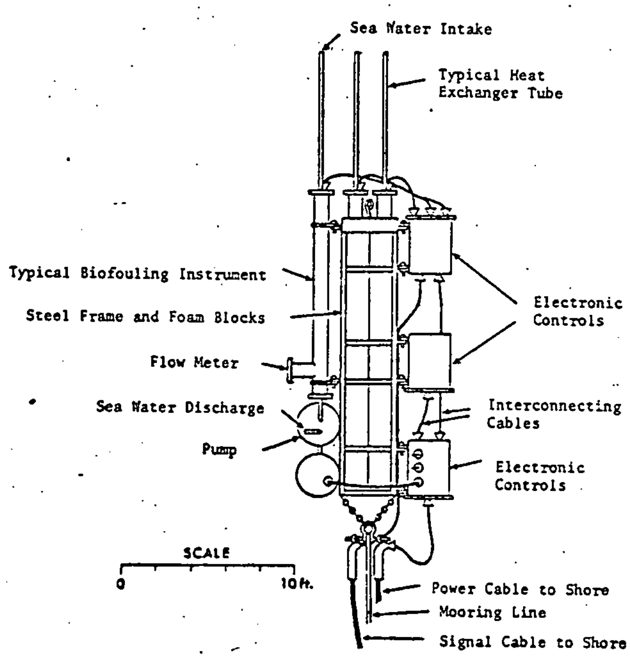

Figure 3. Test units and electronics attached to submarine buoy.

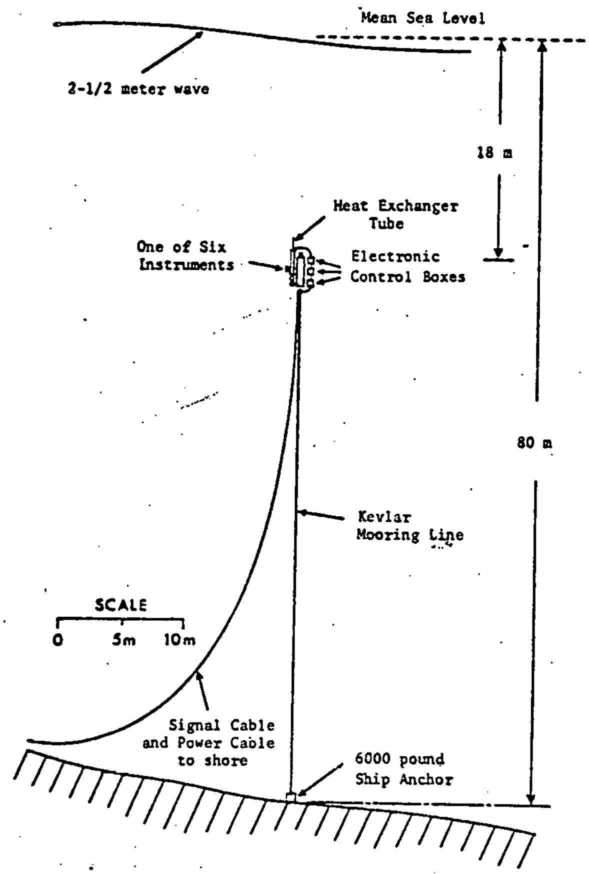

Figure 4. Mooring System.

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 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 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:

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

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,

$$\frac{dQ(t)}{dt} = C\frac{dT(t)}{dt} = -\pi DLh_{ideal}T(t) \quad .$$

The solution of this equation is:

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

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

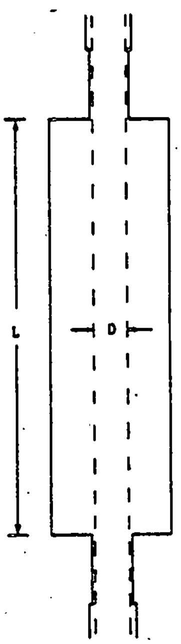

Figure 5. Thick-walled section of the tube.

$$\tau = \frac{C}{\pi DL} \frac{1}{h_{ideal}}$$

This time constant, therefore, provides a link with the heat transfer coefficient, at least in the ideal case.

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, assumption 1 must be removed so that the finite thermal conductivity of the thick-walled section is incorporated. The physical situation is then described by the time-dependent heat flow equation:

$$\alpha \frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r}\right) = \frac{\partial T}{\partial t} \quad . \qquad (2)$$

Here $\alpha = k/c\rho$, where c is the specific heat and ρ is the density of the thick-walled section. Assumptions 2 and 3 are retained in this improved model, so as to preserve the one-dimensional character of the problem. Eq. 2 was solved in closed form under two boundary conditions which are implicit in the model. The first is that no heat passes through the outer surface:

$$\left.\frac{\partial T}{\partial r}\right|_{r_2} = 0$$

where r_2 is the outer radius of the thick-walled section. The second is that all heat passes through the inner surface of the thick-walled section to the flowing water:

$$k \left.\frac{\partial T}{\partial r}\right|_{r_1} = hT$$

where r_1 is the inner radius (r_1=D/2). Thus, the heat transfer coefficient (h) enters the model through a boundary condition.

The solution of Eq. 2 is:

$$T(r,t) = \sum_{n=1}^{\infty} T_{on} Z_n(r) \; e^{-\alpha \lambda_n^2 t} \qquad (3)$$

where:

$$Z_n(r) = J_o(\lambda_n r) Y_1(\lambda_n r_2) - Y_o(\lambda_n r) J_1(\lambda_n r_2) \; .$$

Here J and Y are Bessel functions of the first and second kinds, respectively; r is an arbitrary radial position in the thick-walled section; λ_n is a constant which is evaluated by solving a transcendental equation involving h; and T_{on} is a constant which depends on the temperature distribution at t=0 and which also involves h.

Thus, the full solution of Eq. 2 is an infinite series, each term of which is an exponential with a different time constant ($\tau_n = 1/\alpha\lambda_n^2$). In practice, however, each τ_n is considerably smaller than the preceding one ($\tau_{n+1} < \tau_n$), and all terms become negligible compared to the first in a time of the order of seconds after the heat has been cut. Then Eq. 3 is reduced to:

$$T(r,t) = T_{01} Z_1(r) e^{-t/\tau_1} \quad .$$

This is of the same form as obtained in Eq. 1 for the ideal case (infinite thermal conductivity), except that the time constant is slightly different.

Since τ_1 depends on λ_1 which in turn depends on h, a relationship $h = h(\tau_1)$ can be obtained within this model by solving the transcendental equation for λ_1 with various values of h. An experimental determination of the time constant τ_1 is then effectively a measurement of h. But h determined in this way (designated $h_{uncorrected}$) must be larger than the true value of h because assumptions 2 and 3 do not permit heat leaks to the thin-walled section of the tube or to the air. Corrections for these effects are made by estimating the ratio of heat loss to the thin-walled section of the tube ($\dot{Q}_w$) to the main loss directly into the water ($\dot{Q}_o$), and the ratio of heat loss to the air ($\dot{Q}_a$) to $\dot{Q}_o$. The wall ratio is:

$$R_w = \frac{\dot{Q}_w}{\dot{Q}_o} = \frac{2}{L}\sqrt{\frac{kt}{h}}$$

where t is the wall thickness of the tube immediately above and below the thick-walled section. As shown in Fig. 5 the heat exchanger tube is machined down above and below the thick-walled section in an effort to make R_w small. R_w has a typical value of 0.05. Similarly the air ratio is given by:

$$R_a = \frac{\dot{Q}_a}{\dot{Q}_o} = \frac{h_a}{h} \frac{r_2}{r_1} \frac{Z_1(r_2)}{Z_1(r_1)}$$

where h_a is the coefficient for heat transfer from the outer surface of the thick-walled section to the air and Z_1 is the combination of Bessel functions as given in conjunction with Eq. 3. Because $h_a \ll h$, R_a is small, typically 0.01. In the expressions for R_w and R_a, $h_{uncorrected}$ is used for h. The final corrected value for h is then given by:

$$h_{corrected} = (1-R_w-R_a)h_{uncorrected} \ .$$

Implicit in the above description is that the "tube" and the "thick-walled section" are of the same physical piece of material. In actuality, the "tube" is a schedule 40 1" pipe (copper-nickel) and the "thick-walled section" is a copper cylinder which fits tightly around the pipe. The pipe region then will have values of k, c, and ρ which can be considerably different from those of the copper cylinder. These differences have been properly taken into account in solving the heat flow equation (Eq. 2) for this two-region case. The solution follows qualitatively as outlined for the single-region case above, with the actual calculations being, of course, more complicated. One striking result of the calculations is that the dominant time constant in the thermal decay (τ_1) is the same in both regions.

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 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 THEORY 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

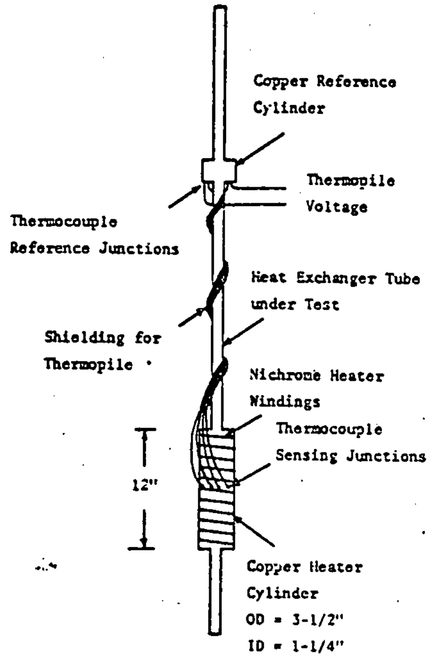

Figure 6. Tube with instrumentation.

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.

The thermopile voltage is sent via well-shielded lines into a DC amplifier with a gain of 10,000 and then into a digital voltmeter (DVM). The transfer of thermopile voltage from the DVM to paper tape is made upon command from a crystal-driven remote clock at intervals of 2, 4 or 10 seconds. In this way, a permanent and computer-compatible record of amplified thermopile voltage as a function of time is made during the course of a heat transfer coefficient measurement. Measurements were made at flow velocities ranging from 2.0 to 8.4 ft/sec.

A typical decay curve of amplied 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 an initial transient period, 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%.

Note that in this procedure no power

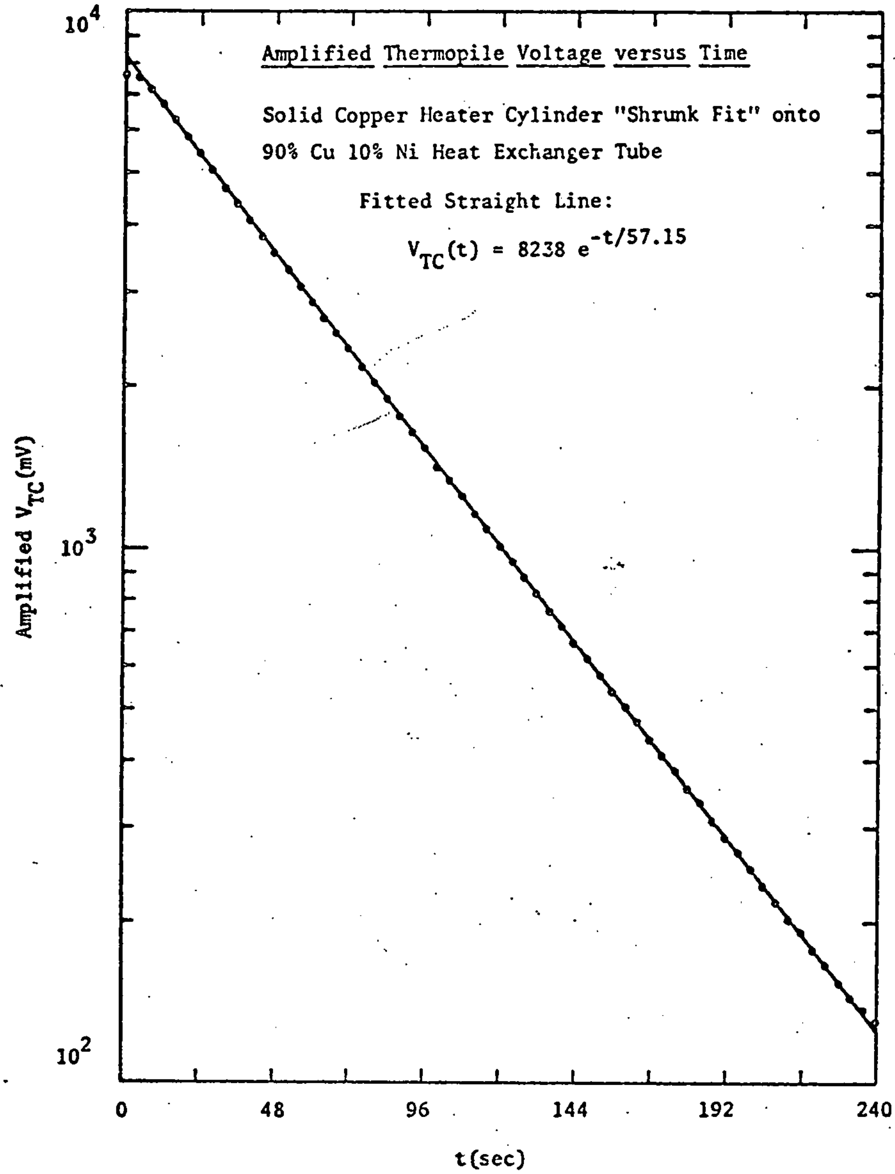

Figure 7. Decay of temperature (thermopile voltage) with time.

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 to v by:

$$h = \frac{160(1+0.012\ T_f)}{D^{0.2}}\ v^{0.8} \qquad (4)$$

where D is the inner diameter of the tube and T_f is the film, or laminar layer temperature (3). 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

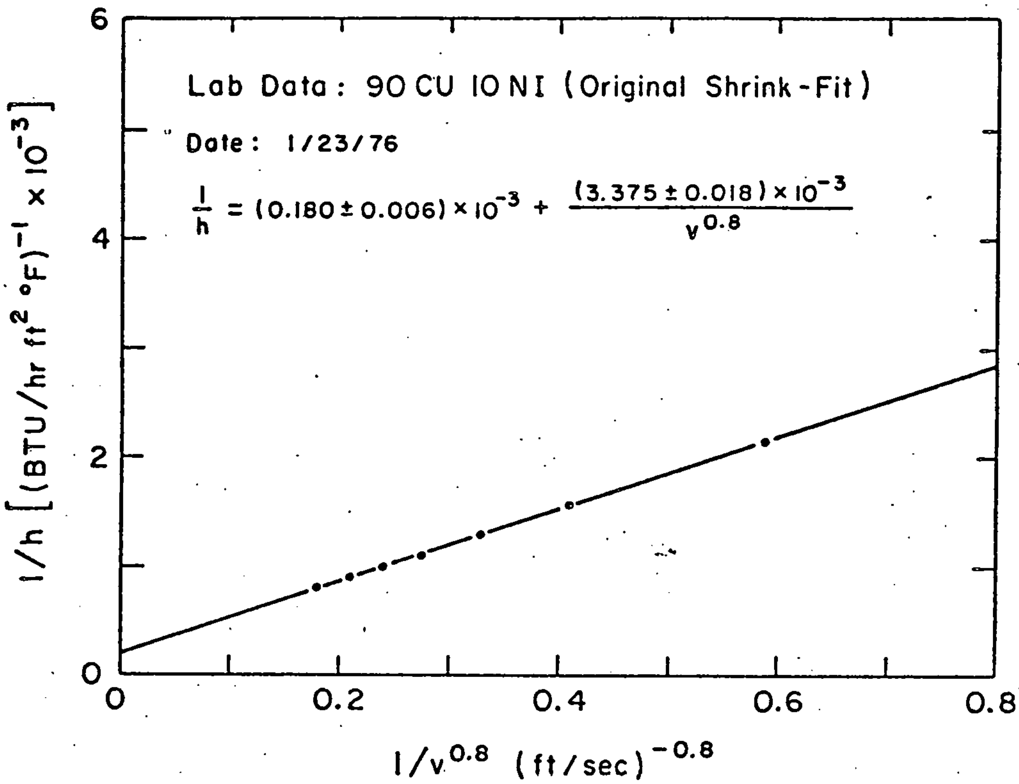

Figure 8. Dependence of the heat transfer coefficient on flow velocity.

the data yielded an intercept of (0.180±0.006) x 10^{-3} and a slope of (3.375±0.018) x 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. 4.

The non-zero intercept of the 1/h axis in Fig. 8 is due to a velocity-independent thermal resistance between the thermocouple locations and the turbulent region of the flowing water. Since the inside of the heat exchanger tube was cleaned prior to taking the data, this intercept 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 0.765±0.032.

PRESENT STATUS AND PRELIMINARY RESULTS

At present, nearly all of the components of the six units (Figs. 1 and 2) have been assembled and several of the units have been sent to Hawaii after extensive testing in the lab. The full complement of six units is expected to be on Hawaiian soil sometime during December, 1976. The selection of tube materials for the initial monitoring of biofouling build-up includes titanium, an aluminum alloy and a 90% copper - 10% nickel alloy. Each tube will be characterized by a particular flow velocity with the full range being 2 ft/sec to 10 ft/sec. Biofouling of the tube walls will thus be measured as a function both of tube material and of flow velocity. Means of inhibiting the fouling are to be implemented at a somewhat later stage of the research.

The electronics, submarine electronics housings, and the subsurface buoy (Fig. 3), as well as the mooring line and anchor (Fig. 4) are in hand. All the beach hardware (fenced-in shelter, power generator and electronics) are on hand and have been tested. The signal and power cable (Fig. 4), to be used to connect the system to the power source and control electronics on the beach, has arrived. With the successful deployment of the subsurface buoy and laying of the signal and power cable, the installation as described earlier in this paper will be complete.

During the process of attending to final details related to putting the buoy in place and laying the cable, most of the system components have been tested and preliminary data have been taken. This was done by using two units mounted on a boat which was anchored near the test site at Keahole Point. Warm surface water was pumped to each of the test units from a depth of 20 feet.

The first such data of July 13 and 14, 1976 is shown in Fig. 9. This 1/h versus $1/v^{0.8}$ plot is for an aluminum tube (alloy 6061-T6) and shows considerably more scatter than its laboratory counterpart. The plot of

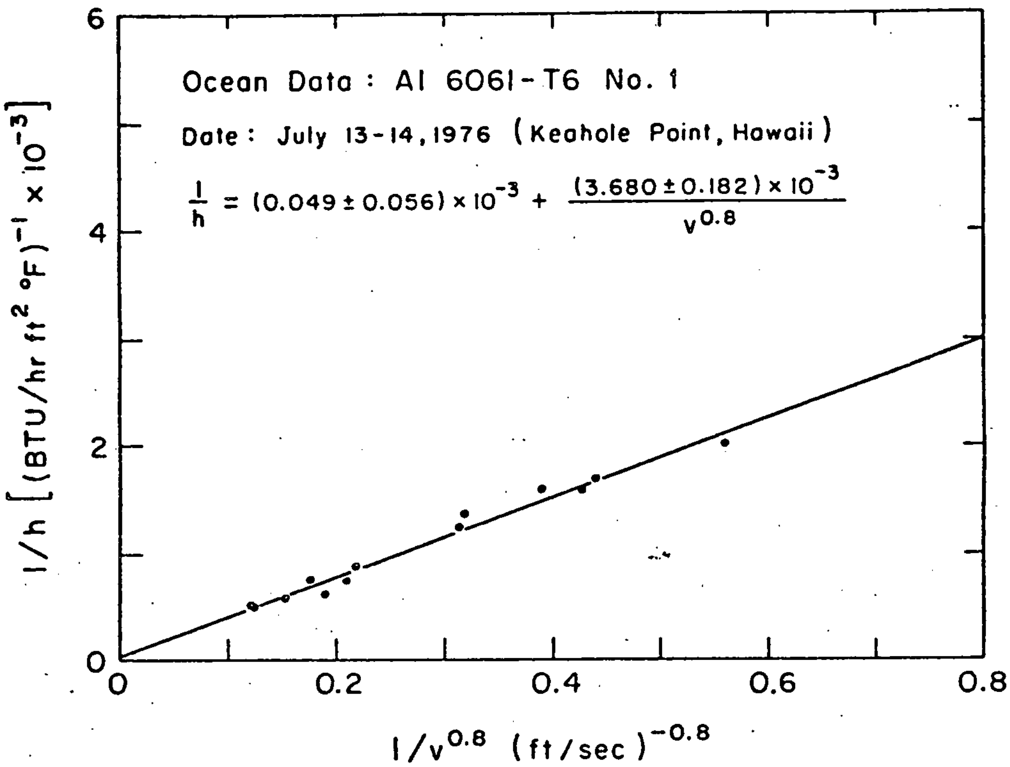

Figure 9. Baseline Ocean Data for an Aluminum Tube.

lab data for 1/h versus $1/v^{0.8}$ for this aluminum tube is not given in this paper but, in terms of scatter of the data, is quite similar to that of Fig. 8 for a 90% copper - 10% nickel tube. It is, of course, highly desirable to keep the scatter of the ocean data small so that small changes in heat transfer coefficient due to biofouling can be detected.

The scatter is due primarily to two effects. The first is the variation of flow velocity due to the rolling of the boat and the consequent motion of the intake hoses which carry the seawater to the test units. This effect will disappear when the test units are mounted on the subsurface buoy. The second, and dominant, effect is shown quite vividly in Fig. 10, where an extreme case of ocean behavior is illustrated.

There the temperature of the ocean water (T_W) passing through the aluminum tube is plotted as a function of time. During the same time interval the natural logarithm of the thermopile decay voltage is plotted. It is obvious that there is a strong correlation between the water temperature fluctuations and the deviation of the thermopile data from the fitted curve. The decay of Fig. 10 should be compared with that of Fig. 7 which was taken under laboratory conditions where the temperature of the flowing water was much more stable. It is evident that considerable uncertainty must be attached to the extraction of a heat transfer coefficient from the slope of the decay curve of Fig. 10.

The term "Window" in Fig. 10 refers to a certain interval of time early in the decay. It is the thermopile data within this window that is fitted to determine h. It should be pointed out that early in the decay the thermopile voltage is high and is less sensitive to water temperature fluctuations than the lower voltages later in the decay. In addition, the proper statistical weight attached to each decay point in the fit is proportional to the square of the thermopile voltage, so that earlier points are much more heavily weighted than later points. Yet this relatively close coupling between water temperature fluctuations and thermopile voltage deviations certainly decreases the precision of the measurement of h, as the data of Fig. 9 reveal.

There are essentially two approaches in dealing with this coupling problem. The first is to incorporate the water temperature fluctuations into the data analysis. This is unwieldy but feasible. A much more satisfactory approach is to remove the source of the coupling. The data of Fig. 10 are explained if the thermal time constant of the thermopile reference cylinder is smaller than the thermal time constant of the heater cylinder. Then the reference cylinder is able to respond to the decrease in water temperature more rapidly than the heater cylinder, and this relatively rapid response causes the thermopile data to pull away from the fitted curve. Nominally, the reference cylinder and the heater cylinder have the same time constant. That the time constants are not the same may be due to the differences between

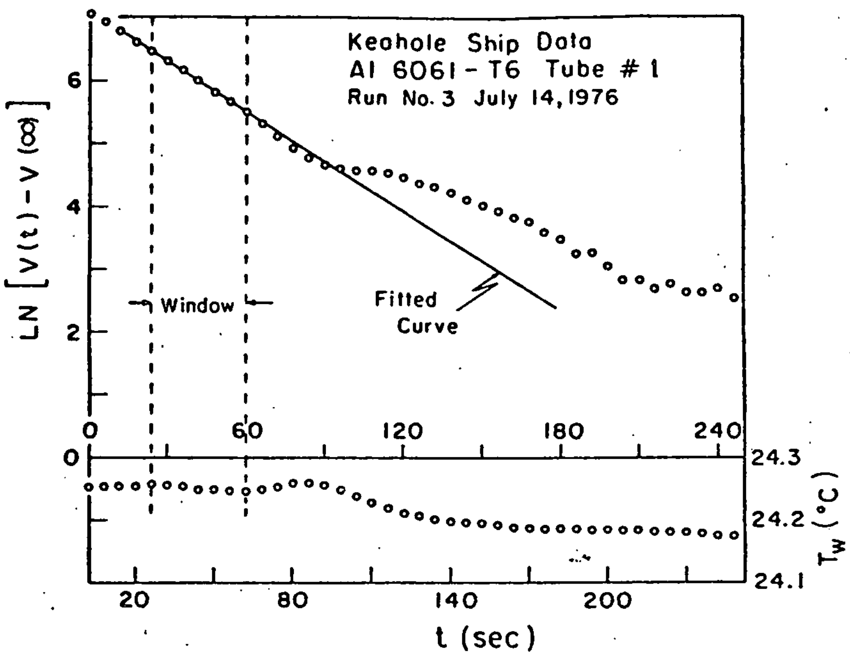

Figure 10. Correlation between Water Temperature Fluctuations and Deviations of the Thermopile Decay Data from the Fitted Curve.

the machined tube surface where the heater cylinder is clamped and the unmachined tube surface where the reference cylinder is clamped, to the fact that the tube wall above and below the heater cylinder is thin whereas above and below the reference cylinder it is normal thickness, to differences in potting the reference junctions and the sensing junctions of the thermopile, or to some other factors.

Now if the heater cylinder and reference cylinder had exactly the same time constant, fluctuations in water temperature would not affect the thermopile decay curve. A simple experimental technique has been developed for measuring the ratio of the time constant of the heater cylinder (τ_H) to the time constant of the reference cylinder (τ_R). Proposed methods for "tuning" the cylinders so that τ_H/τ_R is nearly 1 include varying the pressure by which the cylinders are clamped to the tube, removing mass from the reference cylinder, and adding mass to the reference cylinder by means of rings which would clamp to the circuference of the cylinder. (These methods may be used in conjunction with a thermal grease which serves to reduce considerably the contact resistance between the copper cylinders and the tube.) If a fine tuning with τ_H/τ_R=0.98 is achieved, abrupt changes in the ocean water temperature as large as 0.1°C would only show up on the thermopile as ~0.001°C peaks. This is quite satisfactory since the slight instabilities in the existing electronics limit the resolution of the thermopile circuit to ~0.001°C. No problems are envisioned in achieving this degree of tuning, and consequently the thermopile output is expected to be essentially decoupled from the variations in water temperature.

The results available at the time of this writing concerning the rate of biofouling in two aluminum (alloy 6061-T6) heat exchanger tubes are presented in Table 1. Each aluminum tube formed the heart of one of the two units that was mounted on a boat anchored at Keahole Point, as mentioned earlier. For the first tube (Table 1a), warm surface water was pumped at approximately 3 ft/sec from July 13, 1976 to September 22, 1976 except for a 16 day period beginning August 3 when the flow velocity was inadvertently changed to 4 ft/sec. Similarly, surface water was pumped through the second tube (Table 1b) at approximately 6 ft/sec from August 19, 1976 until September 26, 1976. The pumping was continuous except for a routine shutdown of the electrical generator for an oil change. The shutdown was for a period of a few hours every two weeks. When the heat exchanger units are in final position on the subsurface buoy, the electrical power system will be arranged so that the flow need never by interrupted.

The flow velocity, which is measured during a thermopile decay, has a typical RMS deviation of 0.1 ft/sec or ~3% at 3 ft/sec. This short-term variation is due largely to the rolling of the boat. Also present is a long-term drift toward smaller flow velocities as the filter on the tube inlet begins to clog, and as the inexpensive pump perhaps begins to age. These long-term drifts are counteracted by adjusting the flow control valve to maintain a flow velocity near the nominal value. In any case, before being recorded in Table 1, the resulting heat transfer coefficient at a given velocity was normalized to the velocity of the baseline data (July 13-17 and August 19) by the $v^{0.8}$

Table 1

a) **Nominal Flow Velocity: 3 ft/sec**
Material: Al 6061-T6 (Tube No. 1)

Date	$\langle h \rangle$ $\left(\frac{BTU}{Hr\ Ft^2\ °F}\right)$	$\langle 1/h \rangle$ $\left(\frac{Hr\ Ft^2\ °F}{BTU}\right) \times 10^5$	$\Delta\langle(\frac{1}{h})\rangle = R_f$ $\left(\frac{Hr\ Ft^2\ °F}{BTU}\right) \times 10^5$	$\frac{d}{dt}(R_f)$ $\left(\frac{Hr\ Ft^2\ °F}{BTU\ \ Week}\right) \times 10^5$	t_f (mils)
July 13...17, 1976	627.2±8.0	159.4±2.0	0.0	---	0.0
Aug. 10, 1976	608.8±7.5	164.3±2.0	4.9±2.8	1.23±0.7	0.21±0.12
Aug. 31, 1976	551.6±4.3	181.3±1.4	21.9±2.4	5.67±1.2	0.92±0.10
Sept. 15, 1976	478.8±3.3	208.9±1.4	49.5±2.4	13.14±1.62	2.08±0.10
Sept. 22, 1976	435.4±3.7	229.6±2.0	70.2±2.8	20.70±3.69	2.95±0.12

b) **Nominal Flow Velocity: 6 ft./sec.**
Material: Al 6061-T6 (Tube No. 2)

Date	$\langle h \rangle$ $\left(\frac{BTU}{Hr\ Ft^2\ °F}\right)$	$\langle 1/h \rangle$ $\left(\frac{Hr\ Ft^2\ °F}{BTU}\right) \times 10^5$	$\Delta\langle(\frac{1}{h})\rangle = R_f$ $\left(\frac{Hr\ Ft^2\ °F}{BTU}\right) \times 10^5$	$\frac{d}{dt}(R_f)$ $\left(\frac{Hr\ Ft^2\ °F}{BTU\ \ Week}\right) \times 10^5$	t_f (mils)
Aug. 19, 1976	1083.9±30.0	92.3±2.6	0.0	---	0.0
Aug. 30, 1976	1079.0±12.9	92.7±1.1	0.4±2.8	0.25±1.78	0.02±0.12
Sept. 13, 1976	963.1±11.1	103.8±1.2	11.5±2.9	5.55±2.02	0.48±0.12
Sept. 18, 1976	946.6±24.6	105.6±2.7	13.3±3.7	2.50±6.53	0.56±0.16
Sept. 26, 1976	848.0±14.9	117.9±2.1	25.6±3.3	10.79±4.35	1.08±0.14

factor of Eq. 4.

The evaluation of the degree of fouling of the tube on a particular day is made by carrying out 16 independent thermopile decay measurements. These 16 measurements are analyzed and generally 3 or 4 of the 16 are rejected as non-representative because of exceedingly large variations of water temperature or for some other reason. The remaining 12 or 13 measurements provide a statistical sample from which an average value of h and the RMS deviation on this average value can be calculated. This average value and the associated uncertainty are given in Table 1.

Also given in Table 1, along with uncertainties, are the overall thermal resistance (1/h), the increase in thermal resistance associated with the fouling layer ($\Delta(1/h) = R_f$), the rate of change of $R_f(dR_f/dt)$, and the thickness of the fouling layer (t_f). This thickness was estimated by assuming that the thermal conductivity of the fouling material is the same as that of water. The derivative dR_f/dt was evaluated numerically from the data and is therefore an estimate of the rate of fouling at a time midway between two successive data points.

In Fig. 11 is shown a plot of the thermal resistance of the fouling layer as a function of time. The fouling initially builds up rather slowly but as time goes on the build-up is more and more rapid. This limited quantity of data appears to be empirically described by a cubic relation between R_f and t. One might speculate that in the early stages of the experiment the fouling organisms have difficulty attaching themselves to the smooth tube wall. With time, however, the interior surface of the tube does begin to become fouled and the corresponding rougher surface allows attachment more readily.

The data of Fig. 11 give cause for some optimism concerning the biofouling problem in OTEC heat exchanger tubes, at least in the waters near Keahole Point, Hawaii. Although the fouling after 10 weeks is rather severe, it is quite modest for the first 3 weeks. This means, among other things, that the methods for inhibiting biofouling may be simpler and much less demanding than were once conjectured. It is important to realize, however, that these conclusions must be considered preliminary until confirmed by a repetition of the experiment.

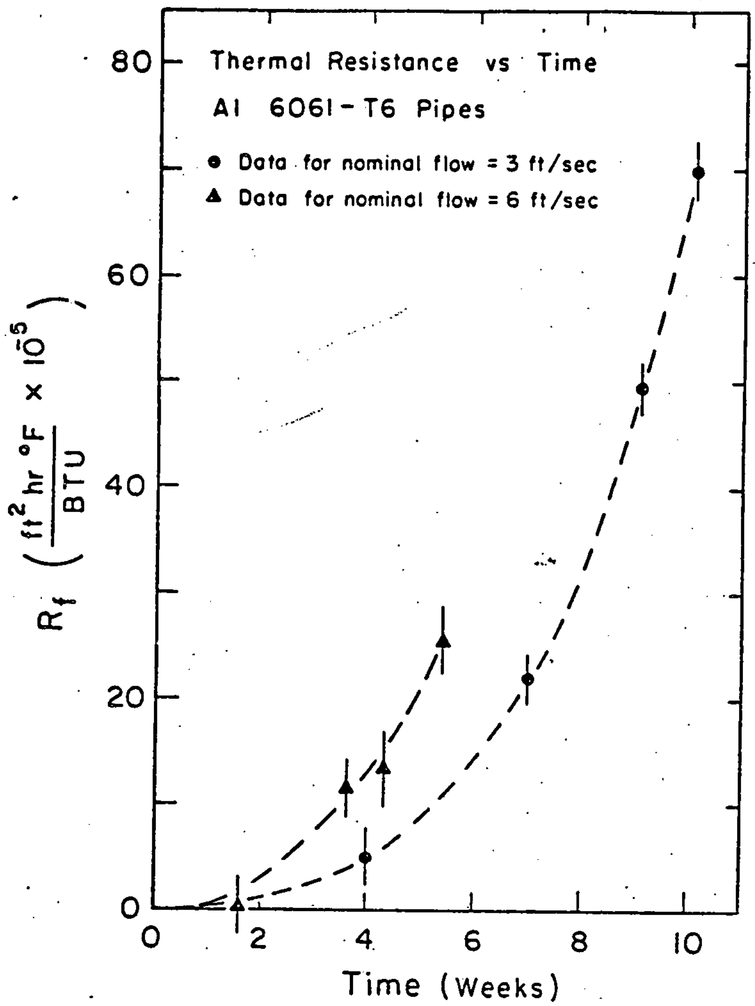

Figure 11. Thermal Resistance Due to Biofouling Layer as a Function of Time.

At the time of the conference all recent results on fouling rates of various materials at various flow rates, anti-fouling methods, metallurgical studies of corrosion of the tubes, and biological studies of the fouling organisms will be presented, along with a discussion of future work to be undertaken by this group.

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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 Baumeister and Lionel S. Marks, McGraw-Hill, 1966 (p. 4-100).