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MODELING OF PUMPING COLD DEEP NUTRIENT-RICH SEAWATER FOR MARICULTURE AND NUCLEAR POWER PLANT COOLING APPLICATIONS

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Abstract—A simple theoretical model is developed to assess the technical feasibility of pumping cold deep nutrient-rich seawater for mariculture and nuclear power plant cooling. Simulations for two locations in eastern Taiwan have shown that replacing warm surface seawater by cold deep seawater as nuclear power plant coolant is technically feasible for pipes with diameters larger than 150 cm and intake levels deeper than 400 m. The gravitational force-induced flow system is also examined and shown not to be suitable for the nuclear power plant cooling application. However, it is shown that both systems are suitable for mariculture applications. The required pumping powers at different pipeline conditions are also presented in the paper, for design purposes.

NOMENCLATURE

W_p	Pumping work per unit mass
P	Pressure
L	Pipeline length
D	Inside diameter of pipeline
R_h	Hydraulic radius
Re	Reynolds number
Pr	Prandtl number
Pe	Peclet number
H	Intake level
Y	Depth
X	Pipeline distance from intake
P_a	Atmospheric pressure
Z	Density-gradient-head loss
$\dot{W}_p$	Pumping power to the seawater flow in the pipe
T	Local fluid temperature inside the pipe
C_p	Specific heat of seawater
U	Overall heat transfer coefficient
T_s	Temperature distribution of seawater
T_∞	Seawater temperature at infinite depth.
$\dot{W}_{net}$	Net power output of original power cycle
T_B	Boiling temperature of the power cycle
T_C	Condensation temperature of the power cycle
Q_B	Heat added to the boiler of the power plant
T_{s0}	Surface seawater temperature
T_e	Deep seawater temperature at the outlet of the pipeline
T^*_C	New condensation temperature of the power plant
$\dot{W}^*_{net}$	New power output
G	Net power gain due to deep cold seawater cooling
B	Ground-hole depth
ν	Fluid *kinematic* viscosity inside the pipe
g	Gravitational acceleration
h	Gravitational height
ρ	Fluid density

f	Friction factor
e_v	Loss factor
ρ^*	Dimensionless density distribution
w^*_p	Dimensionless pumping work
n	Friction parameter $= (v^2/gD^3)(L/H)$
m_e	Mass flowrate in the pipe
k_f	Thermal conductivity of fluid
μ	Viscosity of seawater
β	Temperature curve constant
t	Dimensionless seawater temperature in the pipe
y	Dimensionless depth
t_s	Dimensionless deep seawater temperature at the outlet of the pipe
r	Dimensionless ground-hole depth.

1. INTRODUCTION

THE IDEA of using the temperature difference between the warm surface water of the tropical oceans and the deep cold layer returning from the arctic regions to run a Rankine-cycle heat engine to generate electricity was first proposed by d'Arsonval in 1881 and was first demonstrated by Claude in 1930. However, few research activities had been stimulated around the world until the energy crisis. In addition to its low-temperature characteristic, ocean deep water is the largest nutrient reservoir for marine life. Warmed seawater coming from a depth below 700 m could be used for the growth of algae, shellfish, crustacea, and seaweed. The growth rate has been proved to be much faster than in nature (Othmer and Roels, 1974). An experimental station has been in operation on the north coast of St Croix, one of the U.S. Virgin Islands, near Puerto Rico (Othmer and Roels, 1976). Three polyethylene pipelines, each 1800 m in length and with a 6.9-cm inside diameter, supply water from an 870-m depth at a rate of 150 l/min. This water is warmed by being drawn up through the small pipes, so that it is satisfactory for mariculture applications. One of the experiments showed a 27-fold increase over unicellular algae grown in water from the surface. Considerably rapid growth is also found for clams, European oysters, bay scallops, and lobsters etc.

The combined applications utilizing cold deep seawater were proposed by Othmer (1976). He suggested a number of engineering alternatives which first use the cold deep seawater for the OTEC system and then use the waste seawater to grow seafoods.

When pumping cold seawater from the deep sea through a long pipeline, pumping power is consumed in overcoming the flow friction loss and the density–gradient–head loss. The latter results from the density difference between the seawater in the pipe and the seawater outside the pipe, because all the water in the pipe is approximately at the density of the intake level whereas the density of the water outside the pipe decreases all the way to the sea surface. On the other hand, when drawing cold seawater from the ocean depths through the warmer water up to the surface, the cold seawater will be warmed and the temperature rise will have a large impact on system performance. A great deal of theoretical work has been performed recently to study the technical and economical feasibility of extracting cold deep seawater for power generation (Griffin, 1975; Watt *et al.*, 1977; Zener, 1977). However, little has been mentioned in detail about the engineering aspects of pumping cold deep seawater for OTEC, mariculture or other applications.

Due to some unknown reasons, diffuse upwelling and nutrient-rich layers at shallow depths are found at certain offshore areas near eastern Taiwan. Considerably higher nutrient content is expected at deeper layers. Water in these layers can be pumped up to the seashore for mariculture applications after exposure to the atmosphere for oxygen saturation and warming. The low-temperature nature of the deep seawater can also l e used for nuclear power plant cooling, to promote thermal efficiency or for OTEC applications. To understand this, a simple analytical model for a straight pipeline stretched directly from the seashore into the deep-sea region has to be developed to evaluate the pumping power for drawing deep seawater up to shore and the temperature rise due to warming-up during pumping, as well as the resultant effect on the feasibility of nuclear power plant cooling (see Fig. 1). To reduce the maintenance problem of the pumping facilities, a second kind of piping arrangement is proposed and examined. This pipeline is similar to the first kind except that the outlet of the pipeline is open to a deep ground-hole at the seashore, so as to maintain an elevation between the sea surface and the outlet of the pipeline and thus induce a gravitational driving force to draw up deep seawater. It is apparent that the induced flowrate will be a function of pipeline diameter, intake level, and elevation difference between the pipeline outlet and the sea surface. The calculation results presented in the present paper can be used for design purposes. For nuclear power plant cooling applications, a simulation will be carried out for a nuclear power plant at the two locations on the eastern coast of Taiwan to study the engineering feasibility of replacing the warm surface seawater coolant by cold deep seawater.

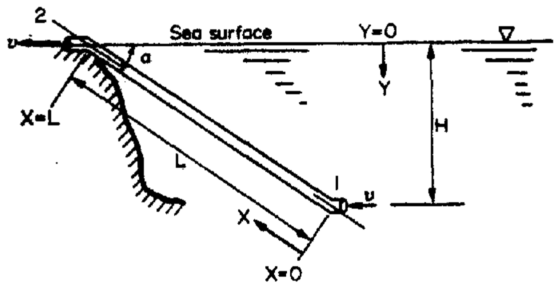

FIG. 1. Basic pipeline configuration.

2. PUMPING DEEP SEAWATER THROUGH PIPELINE

The basic pipeline model considered in the present analysis is shown schematically in Fig. 1. For steady incompressible fluid flow through a pipeline, the Bernoulli equation can be used to deal with the momentum balance:

$$\Delta(v^2/2) + g\Delta h + \int_1^2 dP/\rho - W_p + \sum_i [f(L/R_h)v^2/2]_i + \sum_i (e_v v^2/2)_i = 0, \tag{1}$$

where v is the average velocity of the fluid inside the pipe, h is the gravitational height, P is the fluid pressure, ρ is the fluid density inside the pipe, W_p is the pumping work to the fluid per unit mass, g is the gravitational acceleration and L is the pipe length. The fifth term of (1) represents the sum of all sections of the pipeline. f is the friction factor depending on the Reynolds number and can be expressed for smooth pipes as, according to Moody (1944),

$$f = \begin{cases} 16/\mathrm{Re} & \mathrm{Re} < 2100 \\ 0.0791/\mathrm{Re}^{0.25} & 2100 < \mathrm{Re} < 20{,}000 \\ 0.046/\mathrm{Re}^{0.2} & 20{,}000 < \mathrm{Re} < 10^7, \end{cases} \tag{2}$$

where Re is defined as vD/ν, D is the inside diameter of the pipe and ν is the kinematic viscosity of the fluid. R_h in (1) is the mean hydraulic radius. For a circular pipe, $R_h = D/4$. e_v is the loss factor due to fittings, valves, and bends etc.

To simplify the analysis, the following assumptions are made:

(1) Seawater is incompressible. The cross-section of the pipeline is uniform and the pipe is straight all the way from the seashore down to the offshore deep sea region, as shown in Fig. 1. Therefore the average velocities of the seawater at the intake and at the outlet of the pipeline are same, i.e. $\Delta_v = 0$, and the loss factor can be ignored, i.e. $e_v = 0$.

(2) The seawater in the pipeline is all at the density of the intake level. Therefore, $\rho_1 = \rho\Delta = \rho_H =$ constant, and

$$\int_1^2 \mathrm{d}P/\rho = (P_2 - P_1)/\rho_H. \tag{3}$$

(3) The outlet of the pipe is at sea level, and the pipe is circular.

(4) Only one straight pipe is considered in the present analysis, so that

$$\sum_i [f(L/R_h)\, v^2/2]_i = f(L/R_h)v^2/2. \tag{4}$$

From above assumptions, (1) can be written as:

$$W_p = gH + (P_2 - P_1)/\rho_H + f(L/R_h)v^2/2. \tag{5}$$

Since the seawater is pumped into the atmosphere at the outlet, the outlet pressure P_2 is equal to the ambient pressure P_a, i.e. $P_2 = P_a$. Thus the intake pressure P_1 can be evaluated by the hydrostatic relation:

$$P_1 = P_a + \int_0^H \rho\, g\mathrm{d}Y = P_2 + \int_0^H \rho\, g\mathrm{d}Y, \tag{6}$$

where ρ is the seawater density distribution with respect to depth outside the pipe. Y represents the depth from sea surface. Substituting (6) into (5), we get

$$W_p = gH - \int_0^H (\rho/\rho_H)g\mathrm{d}Y + f(L/R_h)v^2/2. \tag{7}$$

Normalization of (7) yields

$$w_p^* = 1 - \int_0^1 \rho^* \, \mathrm{d}\, y + (v^2/2g)(L/H)f/R_h, \tag{8}$$

where

$$w_p^* \equiv W_p/gH,\ y \equiv Y/H \text{ and } \rho^*(y) \equiv \rho(Y)/\rho_H\,.$$

Also we define a new parameter Z as $Z = 1 - \int_0^1 \rho^* \, dy$. Here Z represents the density-gradient–head loss depending on the density distribution. Since there exists thermohaline circulation at deep sea region, the variation of density with depth is generally not so smooth that a single profile $\rho(Y)$ can be approximated. Therefore, instead of direct integration, a finite difference summation is used to calculate the density–gradient–head loss, i.e.

$$Z = 1 - \sum_i \bar{\rho}_i^* \, \Delta\, y_i, \tag{9}$$

where $\bar{\rho}_i^* = (\rho_i^* + \rho_{i-1}^*)/2$, and $\Delta y_i = y_i - y_{i-1}$, $i = 1, 2, \ldots$ Since the density distributions are different in different oceans and in different seasons, Z is actually a regional and seasonal function determined from measurements. Substituting (2) for friction factor f into (8), we obtain

$$w_p^* = \begin{cases} Z + 32\, n\mathrm{Re}, & \mathrm{Re} < 2100 \\ Z + 0.1582\, n\mathrm{Re}^{7/4}, & 2100 < \mathrm{Re} < 20{,}000 \\ Z + 0.092\, n\mathrm{Re}^{9/5}, & 20{,}000 < \mathrm{Re} < 10^7, \end{cases} \tag{10}$$

where $n = (\Delta^2/gD^3)(L/H)$. Equation (10) shows that the required pumping work to the fluid is a function of Z, n, and Re as plotted in Fig. 2. And, n is a factor depending on the viscosity of seawater and the geometry of the pipeline.

The variation of Z-values on the eastern coast of Taiwan ranges from 0.001 to 0.002 for depths down to 1000 m. This is small, but cannot be arbitrarily ignored. If the flowrate is very small, such as in mariculture applications, Z becomes dominant. For a high flowrate such as in nuclear power plant cooling applications, it may be neglected compared with the frictional loss.

From (10), the total pumping power $\dot{W}_p$ for a given mass flowrate of deep seawater m_c from depth H can be evaluated by the relation:

$$\dot{W}_p = m_c g H w_p^*. \tag{11}$$

3. HEATING LOSS DURING PUMPING

When pumping the cold deep seawater through the pipeline, the water will be warmed due to heating through the pipe wall by the warmer outside seawater. A simple analytical model will be developed to predict the thermal loss.

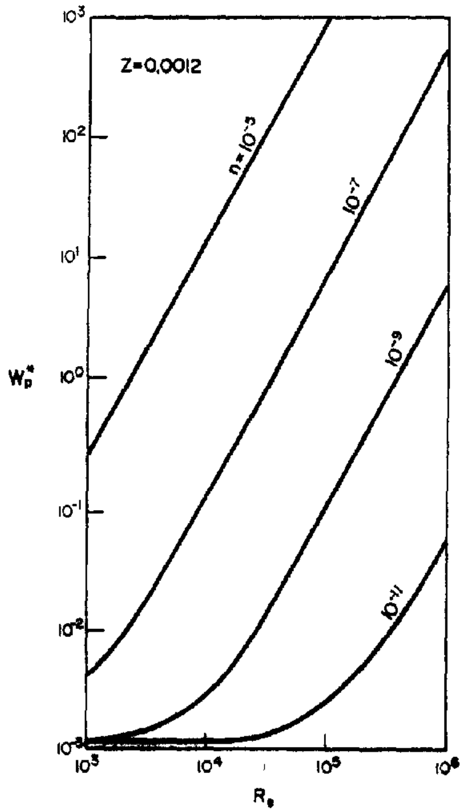

FIG. 2. Pumping work vs Reynolds number.

By energy balance to a differential volume of seawater inside the pipe dL, the following equation can be derived:

$$m_c C_p \mathrm{d}T = U \pi D(T_s - T)\mathrm{d}L, \tag{12}$$

where C_p is the specific heat of seawater, T is the mean bulk seawater temperature at L, U is the overall heat transfer coefficient across the pipe wall, T_s is the temperature distribution of seawater with respect to depth, and L is the position of the intake. From Fig. 1, the transformation of the coordinate yields

$$\mathrm{d}L = -(L/H)\mathrm{d}Y. \tag{13}$$

Substituting (13) into (12) and rearranging, we obtain

$$\frac{\mathrm{d}T}{\mathrm{d}Y} = -\frac{4L}{k_f \mathrm{Pr}\mathrm{Re}}\left(\frac{L}{H}\right)[T_s(Y) - T(Y)], \tag{14}$$

where k_f is the thermal conductivity of seawater inside the pipe, Pr ($\equiv \mu_f C_p / k_f$) is the Prandtl number of the seawater. Equation (14) can be solved if the temperature distribution of seawater, $T_s(Y)$, known.

In many oceans, the temperature variations with depth are so smooth that an exponential function (15), can be used to fit the measured data.

$$T_s(Y) = T_\infty + [T_s(0) - T_\infty] (\exp - \beta Y), \tag{15}$$

where β is a constant depending on location and season, and T_∞ is the temperature at infinite depth. Normalization of (14) in conjunction with (15) will give

$$\mathrm{d}t/\mathrm{d}y - \varepsilon t = - \varepsilon t_s \tag{16}$$

where

$$t \equiv \frac{T_s(0) - T(Y)}{T_s(0) - T_s(H)},$$

$$t_s \equiv \frac{T_s(0) - T_s(Y)}{T_s(0) - T_s(H)} = E [1 - \exp (- \beta Hy)],$$

$$E \equiv \frac{T_s(0) - T_\infty}{T_s(0) - T_s(H)},$$

$$y \equiv Y/H,$$

$$\varepsilon \equiv 4LU/k_f\mathrm{Pe},$$

$$\mathrm{Pe} = \text{Peclet number} = \mathrm{PrRe}.$$

Solving (16) with boundary condition, $t(1) = 1$, we obtain the fluid temperature distribution inside the pipe,

$$t(y) = E - \frac{\varepsilon E \exp (- \Gamma y)}{\Gamma + \varepsilon} + \left[1 - E + \frac{\varepsilon E \exp(- \Gamma)}{\Gamma + \varepsilon}\right] \exp [\varepsilon(y - 1)], \tag{17}$$

where $\Gamma \equiv \beta H$. From (17), the outlet temperature of cold deep seawater drawn through a straight pipeline can be determined:

$$t(0) \equiv t_0 = E + (1 - E) \exp (- \varepsilon) - \frac{\varepsilon E}{\Gamma + \varepsilon} \{1 - \exp[- (\Gamma + \varepsilon)]\}. \tag{18}$$

The unity value of t_0 represents no heating loss during pumping which takes place in perfectly insulated pipe, i.e. $U = 0$, or $\varepsilon = 0$. The value of t_0 is always less than 1 and thus indicates the amount of coldness of the deep seawater remaining after pumping out of the sea surface. Therefore the heating loss during pumping can be readily examined from the values of t_0. The factor ε has the physical significance of the overall heat transfer across the pipe wall, and Γ represents the effect of seawater temperature distribution on the heating loss. β or Γ is zero for a constant-temperature sea and approaches infinity for an abrupt change of temperature at the sea surface. Equation (18) is plotted in Fig. 3 to show the variation of t_0 with Γ and ε.

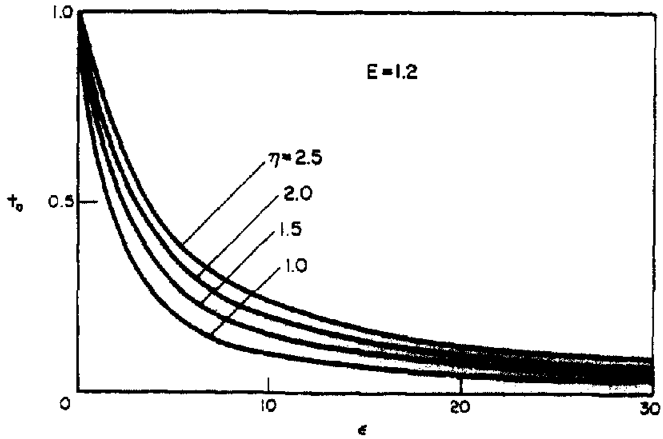

FIG. 3. Deep seawater temperature at the outlet of the pipeline.

4. COOLING OF NUCLEAR POWER PLANT BY COLD DEEP SEAWATER

In drawing cold deep seawater through a long pipeline, power is needed to overcome the friction and density-gradient losses, and the seawater is warmed up by the warmer seawater outside the pipe all the way up to the sea surface. The possibility of using this warmed deep seawater as a cooling medium for a power plant depends upon the flowrate and pipeline designs such as length, diameter and insulation material of the pipe.

Consider a power plant originally cooled by surface seawater. According to the principle of thermodynamics, the replacement of cold deep seawater as coolant would reduce the condensation temperature of the power cycle, thus promoting thermal efficiency. As a first order approximation, one can assume that the thermal efficiency η is proportional to the Carnot efficiency, i.e.

$$\eta = \dot{W}_{net}/Q_H \propto 1 - T_C/T_H \,, \tag{19}$$

where $\dot{W}_{net}$ is the net power output of the power plant, Q_H is the heat added to the boiler, T_C is the condensation temperature, and T_H is the boiling temperature. For simplicity, we assume that replacement with cold deep seawater as coolant will lower the condensation temperature by an amount of $T_{s0} - T_0$, where T_{s0} is the surface seawater temperature, and T_0 is the deep seawater temperature at the outlet of the pipeline which can be evaluated by (18). The new condensation temperature T_C^* then becomes

$$T_C^* = T_C - (T_{s0} - T_0), \tag{20}$$

and the new thermal efficiency η^* becomes

$$\eta^* = \dot{W}^*_{net}/Q_H \propto 1 - T^*_C/T_H \,, \tag{21}$$

where $\dot{W}^*_{net}$ is the new output power. Here, we also assume that the new higher output power can be delivered by a properly renewed turbine generator and all the components in the power plant can be reconstructed to meet the power output increase due to lowering condensation temperature. Therefore, the ratio of the power outputs will be

$$\dot{W}^*_{net}/\dot{W}_{net} = 1 + (T_{s0} - T_0)/(T_H - T_C). \tag{22}$$

The net power gain by deep seawater coolant G is therefore the difference between the power increase, $\dot{W}^*_{net} - \dot{W}_{net}$, and the required deep seawater pumping power, $\dot{W}_p$,

$$G = (\dot{W}^*_{net} - \dot{W}_{net}) - \dot{W}_p \quad . \tag{23}$$

Substituting (22) into (23), we obtain

$$G = [(T_{s0} - T_0)/(T_H - T_C)]\, \dot{W}_{net} - \dot{W}_p \quad . \tag{24}$$

From (24), it can be seen that the net power gain by deep seawater cooling is dependent on the outlet temperature of the pipeline, the highest and the lowest temperatures, the net power output in the original power cycle, and the pumping power consumed in drawing deep seawater through the pipeline. Associated with the previous pumping power calculations, the net power gain can be evaluated.

5. APPLICATIONS FOR MARICULTURE AND NUCLEAR POWER PLANT COOLING ON THE EASTERN COAST OF TAIWAN

Taiwan is a small spindle-shaped island located in the western Pacific ocean. Preliminary investigaions indicate that the east coast of Taiwan is very similar to the Virgin Islands in both meteorological and oceanographical conditions. Therefore, the applications of cold deep seawater on the eastern coast for mariculture and nuclear power plant cooling will be investigated according to the previous modeling.

Two locations on the eastern coast are selected for the present study. The results of the physical oceanographic measurements in the spring season at these locations are shown in Tables 1 and 2 (National Taiwan University, 1977). The seawater temperature distributions are first least-square fitted to (15) with the assumption that T_∞ is 1°C. The values of the exponents β are then determined and shown in Figs 4 and 5. The density distributions are plotted in Fig. 6.

5.1 *Pumping power evaluations*

From (10) and (11), we know that the pumping power required for drawing deep seawater through a pipeline depends on the intake level, the total length and diameter of the pipeline, and the flowrate. The simulation is performed for pipe diameters D = 7.5, 10, 15, 50, 100, and 150 cm. The first four diameters are suitable for mariculture applications

TABLE 1. OCEANOGRAPHIC DATA AT STATION 30 (22° 45′N, 121°17′E, OFFSHORE DISTANCE: 19 km)

Depth m	Temperature °C	Density g/l
0	27.2	1022.1
10	26.9	1022.3
20	25.9	1022.6
30	25.8	1022.6
50	24.4	1023.1
75	23.5	1023.4
100	21.0	1024.1
125	20.4	1024.3
150	18.8	1024.7
200	15.9	1025.4
250	14.3	1025.7
300	12.8	1025.9
400	10.3	1026.3
500	8.5	1026.6
600	7.2	1026.8
700	6.3	1027.0
800	5.6	1027.1
1000	4.0	1027.2

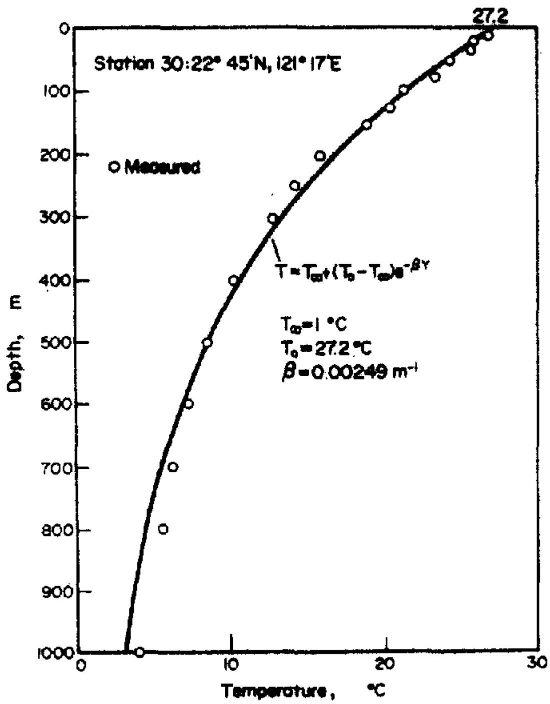

FIG. 4. Temperature distribution at Station 30.

TABLE 2. OCEANOGRAPHIC DATA AT STATION 31 (23°45′N, 121°46′E, OFFSHORE DISTANCE: 10km)

Depth (m)	Temperature (°C)	Density (g/l)
0	25.5	1022.9
10	25.3	1023.0
20	25.2	1023.1
30	24.7	1023.2
50	23.9	1023.5
75	22.3	1024.0
100	20.3	1024.6
125	19.0	1024.9
150	17.7	1025.2
200	15.6	1025.7
250	14.1	1025.9
300	12.9	1026.1
400	10.8	1026.6
500	8.5	1026.9
600	7.0	1027.0
700	6.0	1027.2
800	5.3	1027.2
1000	4.3	1027.4

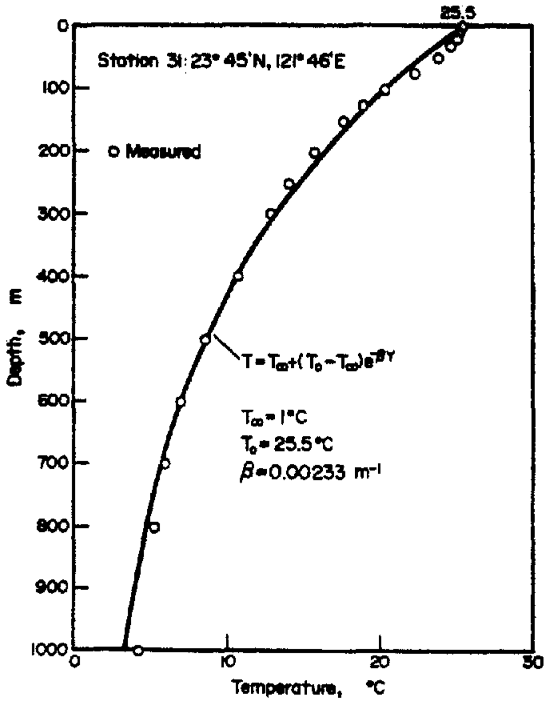

FIG. 5. Temperature distribution at Station 31.

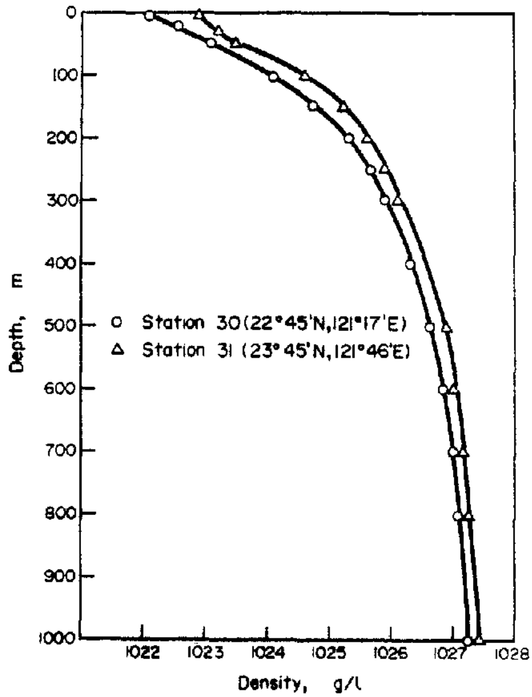

FIG. 6. Density distribution.

and the rest are suitable for power plant cooling. The intake levels H are taken as 400, 600, 800, and 1000 m for the present calculation. The results are shown in Figs 7–12.

It is seen that the required pumping power for each diameter is independent of the intake level at high flowrates. This is due to the fact that the slopes of pipelines in the present simulations are smaller than 0.10, therefore the pumping power is, from (11), approximately independent of the intake level. For low flowrates, the density-gradient-head loss becomes dominant compared with the frictional loss, and therefore the pumping power depends upon the intake level.

The calculations show that the pumping power is less than 100 kW for pipe diameters smaller than 15 cm and mass flowrates lower than 100 ton/h which is just in the range of mariculture applications. To reduce the pumping power in nuclear power plant cooling applications the pipeline diameter has to be larger than 50 cm.

5.3 *Heating loss during pumping*

Equation (18) gives the outlet temperature of cold deep seawater through a pipeline if the overall heat transfer coefficient U is determined. In general, the overall heat transfer coefficient can be evaluated by the following equation:

$$U = 1/\left[1/h_i + \ln(r_0/r_i)/2\pi k_p + 1/h_0\right], \tag{25}$$

where

h_i = convective heat transfer coefficient at pipe inner wall,
h_0 = convective heat transfer coefficient at pipe outer wall,
r_i = inside radius of pipe,
r_0 = outside radius of pipe,
k_p = thermal conductivity of pipe material.

The convective heat transfer coefficient at the inner wall can be evaluated by the equation for forced-convection (Dittus and Boelter, 1930),

$$\mathrm{Nu}_i = \begin{cases} 4.36, & \mathrm{Re} < 2100 \\ 0.023\,\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}, & 2100 < \mathrm{Re} \end{cases} \tag{26}$$

where Nu_i is the Nusselt number defined as $h_i D/k_f$. The heat transfer at the outside wall is determined by the tidal current which varies from location to location in the sea and is very difficult to measure. However, the pipeline is usually laid out along the sea bathymetry so that the effect of tidal currents can be neglected and the heat transfer at the outside wall is dominated by free convection. Since the slope of the pipeline is in general very small, the following empirical equation for free convection on a horizontal cylinder can be approximated to evaluate the heat transfer coefficient at the outside pipe surface (Churchill and Chu, 1975):

$$\mathrm{Nu}_0^{1/2} = 0.60 + 0.387\left\{\frac{\mathrm{GrPr}}{[1 + (0.559/\mathrm{Pr})^{9/16}]^{16/9}}\right\}^{1/6}, \tag{27}$$

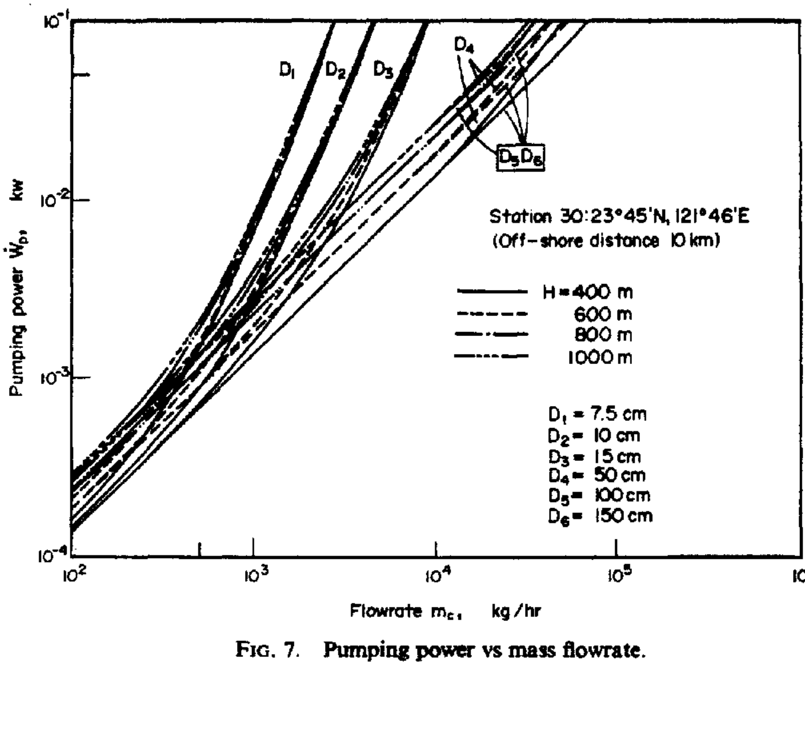

FIG. 7. Pumping power vs mass flowrate.

for $10^{-5} < \mathrm{GrPr} < 10^{12}$, where Gr = Grashof number based on film temperature, and Nu_e is the Nusselt number defined as $h_e D_e / k_f$.

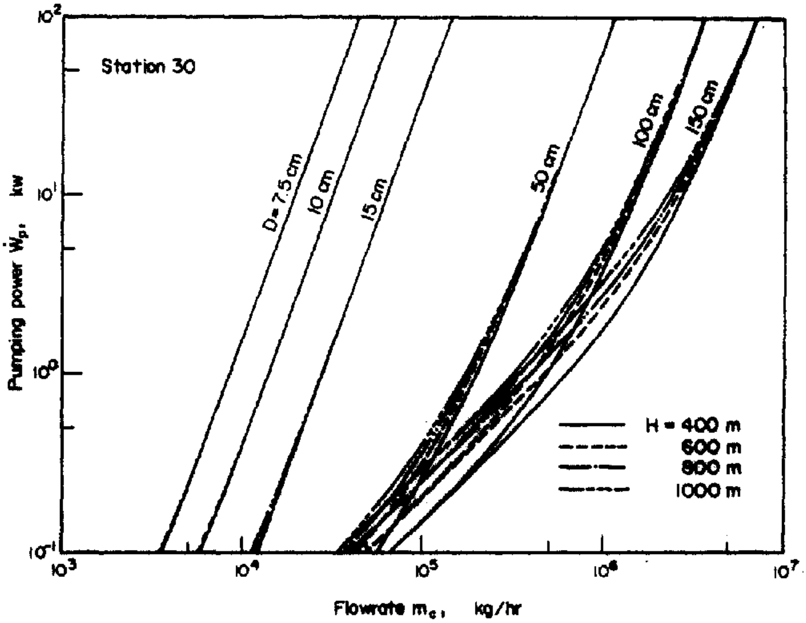

FIG. 8. Pumping power vs mass flowrate.

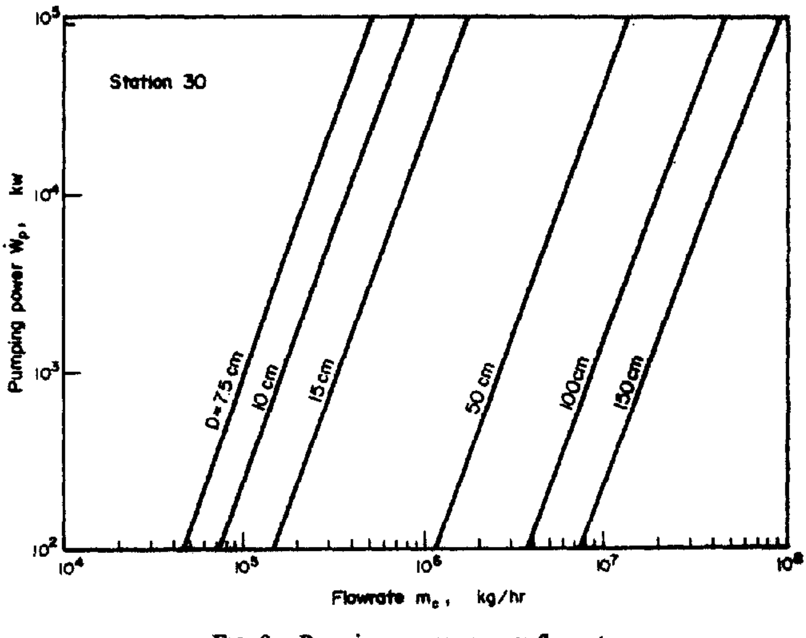

FIG. 9. Pumping power vs mass flowrate.

If the pipe is not well-insulated, e.g. bare metal pipe, the thermal resistance through the wall is negligible compared with the convective heat transfer, and therefore

$$U \approx h_i h_o/(h_i + h_o). \tag{28}$$

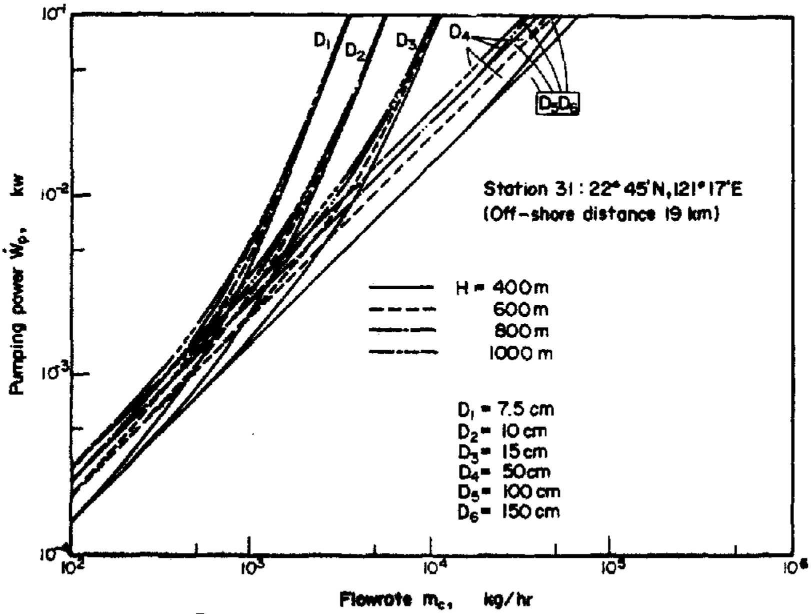

FIG. 10. Pumping power vs mass flowrate.

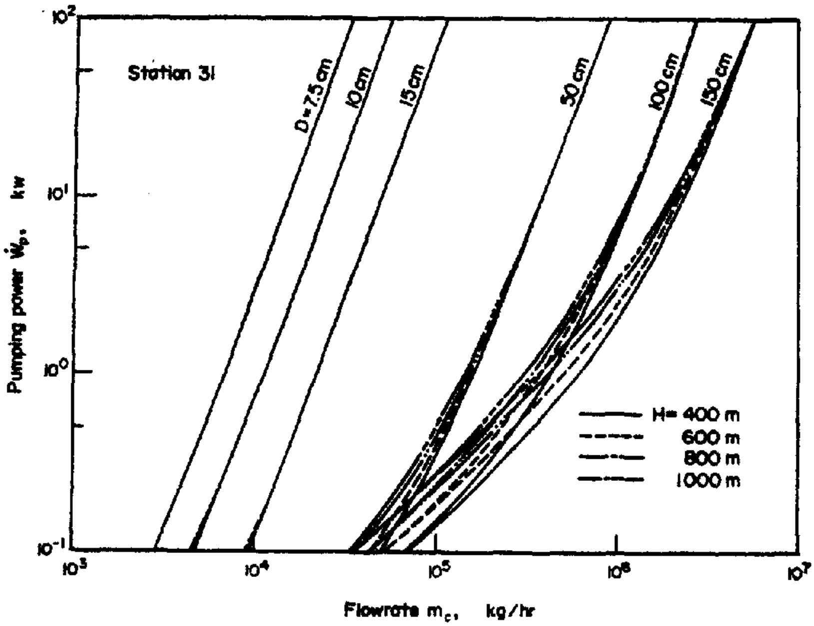

FIG. 11. Pumping power vs mass flowrate.

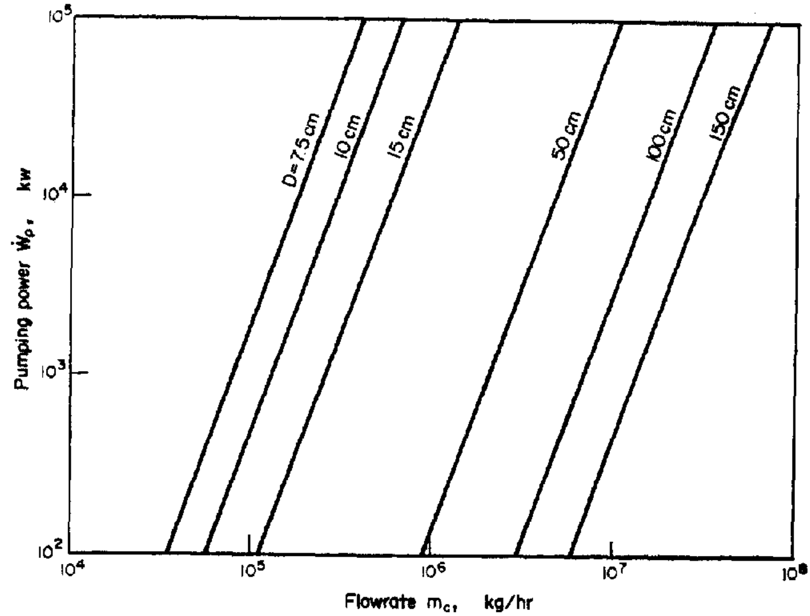

FIG. 12. Pumping power vs mass flowrate.

Associated with the previous data, the temperature rise is calculated for pipe diameter $D = 7.5$, 10, 15, 50, 100 and 150 cm, and the intake level $H = 800$ and 1000 m. The results are shown in Figs 13 and 14.

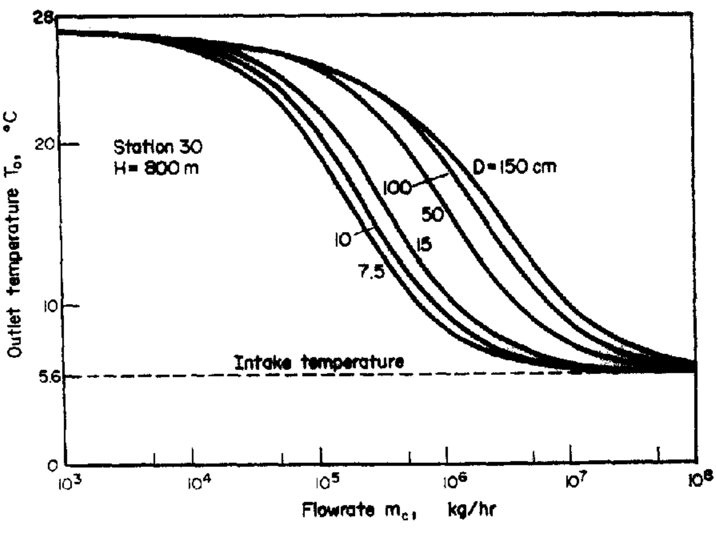

FIG. 13. Outlet temperature vs mass flowrate.

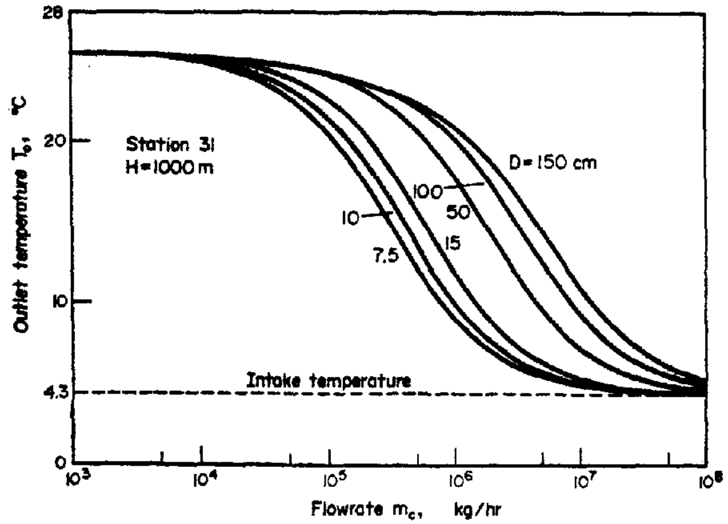

FIG. 14. Outlet temperature vs mass flowrate.

For a well-insulated pipe, the heat resistance through the insulation layer is so high that the overall heat transfer coefficient U approaches zero, and the outlet temperature of the deep seawater is the same as the intake temperature as shown in Figs 13 and 14. It is seen that the cold seawater will be considerably warmed up as the flowrate is too small. High flowrate could reduce the heating loss, but the pumping power would be too large to be applicable.

5.3 *Feasibility of nuclear power plant cooling applications*

Using (24) and previous simulation results for pumping power and heating loss, the feasibility of nuclear power plant cooling applications can be assessed. The nuclear power plant selected for the present simulation is the unit now installed at Chin Shan in northern Taiwan. This power plant now delivers a net output $\dot{W}_{net}$ of 635 MW at the highest cycle temperature T_H of 288 °C, the condensation temperature T_C of 47 °C, and the cooling water flowrate m_c of 3.3 × 10^7 kg/hr.

The net power gain by deep seawater cooling G is calculated for two limiting cases: well-insulated pipeline and bare metal pipeline. Tables 3 and 4 show the variations of net gain with pipe diameter and intake level. It can be seen that a reasonable positive gain can be obtained only for pipes with diameters larger than 150 cm and intake levels deeper than 400 m.

6. GRAVITATIONAL FORCE-INDUCED FLOW SYSTEM

To eliminate the maintenance problems of the pumping facilities, a new piping arrangement is proposed and examined. This pipeline is similar to the first kind except that the outlet of the pipeline is open to a deep ground-hole at the seashore, thus maintaining an elevation between the sea surface and the outlet of the pipe to induce a gravitational driving force to draw up deep seawater (see Fig. 15).

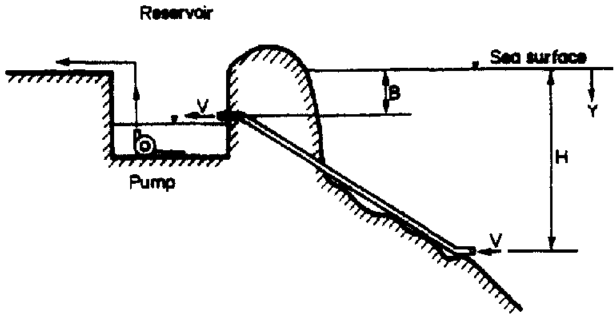

FIG. 15. Gravitational force-induced flow system.

Applying the Bernoulli equation, (1), to the modified pipeline model with the same conditions, the momentum equation can be derived:

$$\begin{cases} Z - r + 32\,n\mathrm{Re} = 0 & \mathrm{Re} < 2100 \\ Z - r + 0.1582\,n\mathrm{Re}^{7/4} = 0, & 2100 < \mathrm{Re} < 20{,}000 \\ Z - r + 0.092\,n\mathrm{Re}^{9/5} = 0, & 20{,}000 < \mathrm{Re} < 10^7. \end{cases} \tag{29}$$

The induced flowrate can thus be determined in terms of r, Z and n. Since the energy equation remains the same as in the previous case, the outlet temperature t_0 can be derived:

$$t_0 = E - \frac{\varepsilon\, E \exp(-\,\Gamma\, r)}{\Gamma + \varepsilon} + \left[1 - E + \frac{\varepsilon\, E \exp(-\,\Gamma)}{\Gamma + \varepsilon}\right] \exp\,[\varepsilon(r - 1)], \tag{30}$$

TABLE 3. NET POWER GAIN (MW) BY DEEP SEAWATER COOLING AT STATION 30 (22°45′N, 121°17′E)

H (m) / D (cm)	Bare metal pipe			
	400	600	800	1000
7.5	-1.1850×10^7MW	-1.1852×10^7MW	-1.1864×10^7MW	-1.1881×10^7MW
10	-2.9787×10^6	-2.9787×10^6	-2.9816×10^6	-2.9865×10^6
15	-4.2540×10^5	-4.2540×10^5	-4.2582×10^5	-4.2640×10^5
50	-1.2756×10^3	-1.2684×10^3	-1.2655×10^3	-1.2634×10^3
100	-8.9888	-1.6730	2.5100	6.3890
150	30.294	37.580	41.879	45.833
	Well-insulated pipe			
H (m) / D (cm)	400	600	800	1000
7.5	-1.1851×10^7MW	-1.1852×10^7MW	-1.1864×10^7MW	-1.1881×10^7MW
10	-2.9787×10^6	-2.9787×10^6	-2.9816×10^6	-2.9865×10^6
15	-4.2540×10^5	-4.2540×10^5	-4.2582×10^5	-4.2640×10^5
50	-1.2741×10^3	-1.2669×10^3	-1.2641×10^3	-1.2620×10^3
100	-6.1697	1.1780	5.1560	9.0150
150	34.300	41.647	45.667	49.588

TABLE 4. NET POWER GAIN (MW) BY DEEP SEAWATER COOLING AT STATION 31 (23°45′N, 121°46′E)

H (m) / D (cm)	Bare metal pipe 400	600	800	1000
7.5	-2.2493×10^7MW	-2.2487×10^7MW	-2.2477×10^7MW	-2.2487×10^7MW
10	-5.6539×10^6	-5.6513×10^6	-5.6503×10^6	-5.6516×10^6
15	-8.0756×10^5	-8.0705×10^5	-8.0698×10^5	-8.0705×10^5
50	-2.4629×10^3	-2.4533×10^3	-2.4483×10^3	-2.4466×10^3
100	-58.205	-49.673	-45.273	-42.641
150	16.893	25.077	29.515	32.329

H (m) / D (cm)	Well-insulated pipe 400	600	800	1000
7.5	-2.2493×10^7MW	-2.2487×10^7MW	-2.2477×10^7MW	-2.2487×10^7MW
10	-5.6539×10^6	-5.6513×10^6	-5.6503×10^6	-5.6516×10^6
15	-8.0756×10^5	-8.0705×10^5	-8.0698×10^5	-8.0705×10^5
50	-2.4609×10^3	-2.4506×10^3	-2.4456×10^3	-2.4439×10^3
100	-53.884	-44.598	-40.418	-38.204
150	22.919	32.182	36.349	38.576

where $r \equiv B/H$. A calculation was carried out at Station 30 to check the induced mass flowrate and the outlet temperature. Figure 16 presents the variations of induced flowrate with the ground-hole depth and the intake level for each pipe diameter. Figures 17 and 18 show the outlet temperature of the deep seawater. It is seen that, to obtain a high flowrate for nuclear power plant cooling, the diameter of the pipe should be larger than 150 cm and the depth of the ground-hole should be deeper than about 75 m. This will not, in practice, be economically feasible. However, for low flowrate applications such as mariculture, the use of a gravitational flow system is possible as the required ground-hole will not be too deep.

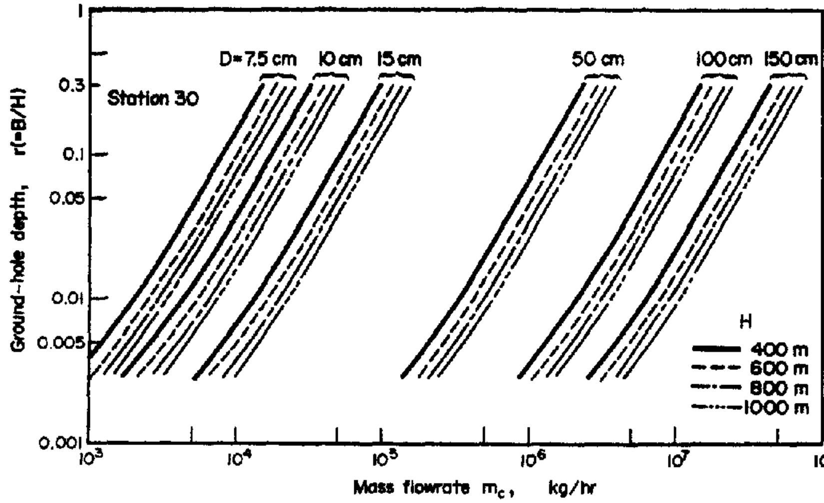

FIG. 16. Induced flowrate vs hole depth.

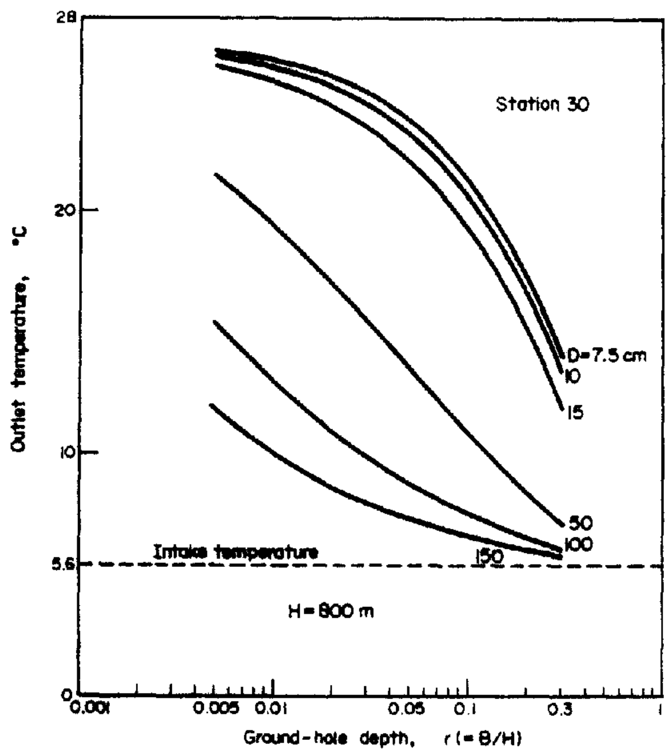

FIG. 17. Outlet temperature vs hole depth for gravitational system.

7. DISCUSSION AND CONCLUSION

A simple theoretical model is developed to assess the technical feasibility of pumping cold deep nutrient-rich seawater for mariculture, and for nuclear power plant cooling. The model assumes that the pipeline is straight and laid out directly to the intake level in the deep sea region. However, the bathymetrical map of Taiwan shows that there is a very narrow shelf along the east coast. Generally the continental slope drops off to 1000 m within 10 km from the coast. At several points off the coast, the 1000 m contour is even closer, within 2 km. Therefore a underwater straight pipeline cannot be constructed to reach the deep sea region. Instead a stepwise pipeline, as shown in Fig. 19, is more likely to be built in practical applications. Thus the previous calculation models should be modified by successive stepwise calculations using equations of similar form. Under these conditions, the pumping work equation becomes, from (1),

$$W_p = \sum_{i=1}^{n} [g\Delta h_i + \int_{L_i} dP/\rho_i + f_i(L_i/R_h)_i v^2/2 + e_{vi}v_i 2/2], \quad (29)$$

and the temperature variation at the ith section becomes

$$dT_i/dX_i = (U_i \pi D_i/m_c C_p)(T_s - T_i), \quad i = 1, 2, \ldots, \quad (30)$$

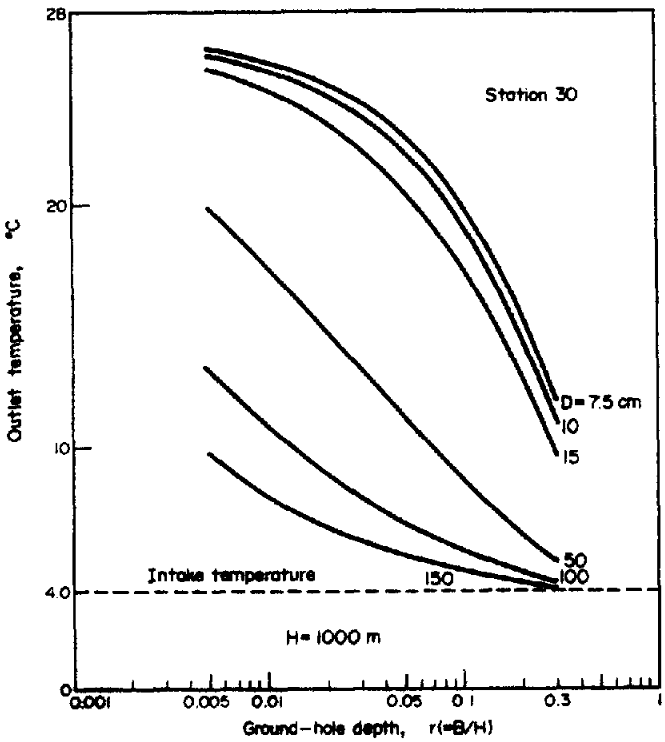

FIG. 18. Outlet temperature vs hole depth for gravitational system.

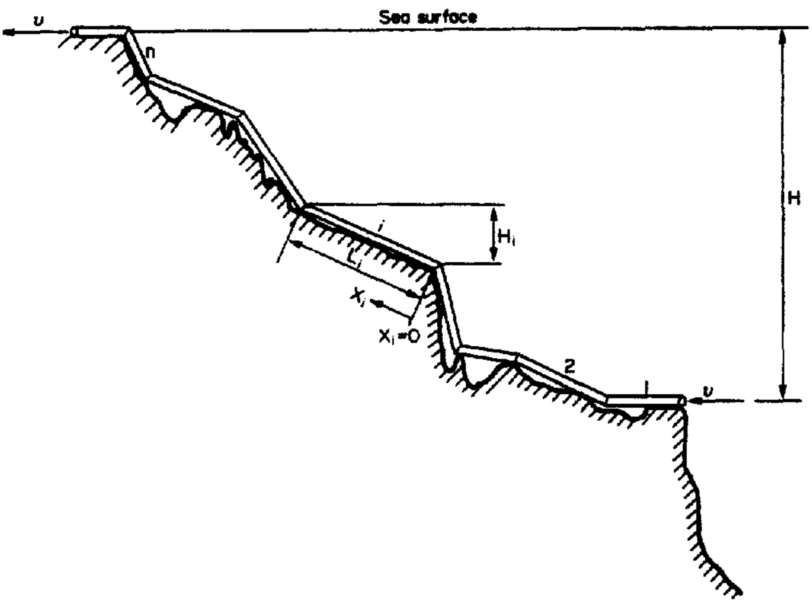

FIG. 19. Real piping configuration.

with the boundary conditions:

$$\text{for } i = 1,\ T_1(0) = T_H \text{ at } X_1 = 0$$
$$\rho_1(0) = \rho_H \text{ at } X_1 = 0.$$

The detailed computations can be accurately achieved when the bathymetrical survey around the application site is completed.

The present analysis for two locations off the east coast of Taiwan and for two kinds of piping models leads to the following conclusions:

1. The pumping power required to draw seawater from the deep sea is always less than 100 kW (134 h.p.) for pipe diameters smaller than 15 cm and mass flowrates lower than 100 ton/hr, which is suitable for mariculture application.

2. Replacing warm surface seawater by cold deep seawater as nuclear power plant coolant is technically feasible for pipes with diameters larger than 150 cm and intake levels deeper than 400 m.

3. The gravitational force-induced flow system is not applicable for nuclear power plant cooling applications, but is applicable for mariculture applications.

4. For accurate predictions, the bathymetrical survey around the application site should be completed and the successive stepwise computation for a real pipeline geometry is required for design work.

5. For further practical applications, an economical feasibility investigation is necessary.

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