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A Study of a Power System by Utilizing Natural Circulation

Effective Utilization of Heat Sources

(1st Report, Principles and Theoretical Performance Analysis)*

by Koji Akagawa**, Terushige Fujii**, Yutaka Ito****,

Seiichiro Amano*****, and Tetsuo Oka****

A power generating system utilizing natural circulation is proposed in order to use a large quantity of low-temperature heat effectively. First, the dimensionless performance characteristics of the output were obtained by a simple approximated theoretical analysis on the assumption of no slip between liquid and vapor, and constant friction losses over the tube length. Next, a more detailed theoretical analysis was made, taking the above assumption into account, and was applied to a system of actual size, using R-113 as a working medium. The effects of each parameter on the performance were shown. Furthermore, the influence of local heating on the output was discussed.

1. Introduction

In the situation of energy problems in the present time a large quantity of low-temperature and low-density energy resource should be utilized. There are energy resources⁽¹⁾⁽²⁾: (i) natural energy such as sun, ocean, and low temperature in cold weather; (ii) exhaust heat and drainage in factories, and (iii) biological energy. Concerning the effective utilization of low-temperature energy, Rankine cycle plant using a low saturation temperature fluid as the working medium (Fluorocarbon, Florinor 85, and the like), which utilizes the heat exhausted from factories and diesel engines, and the power generation using the ocean temperature difference energy (the working fluid of NH_3 , R-22, and the like), and further the power generation using the concentration difference energy have been studied, and some of them have been operated as practical plants⁽¹⁾⁽³⁾⁽⁴⁾.

The purpose of this study is to develop a power generation system utilizing the low temperature energy in the range of 50 to 100 °C. The principle of the system is to generate the power by utilizing the driving force of the natural circulation owing to the difference of the specific weight between liquid and vapor, using the low-saturation temperature fluid. The distinctive features are that it is possible to utilize the low-quality and large quantities of the heat re-

source, which have not been utilized, and there is no need of external power equipment such as pumps and compressors used in the conventional power plants, and it is simple and economical in construction. The driving of principle of the natural circulation system is similar to that in conventional natural circulation boilers, the air-lift pump⁽⁵⁾⁽⁷⁾, the boiling air-lift pump⁽⁸⁾, and the heat-driven pump⁽⁹⁾, but the present system differs from the other systems for power generation.

In this report the principle of the system is proposed and a generalized expression of the performance using an approximate analysis, and a more detailed analysis taking into account the characteristics of the friction loss, and slip ratio in two-phase flow are presented. An experiment to prove the performance conducted by a test plant (the evaporating tube of 37.1 mm ID and 5 m length, and the maximum output 30 W), and the results will be described in subsequent reports.

2. Fundamental Principle and Selection of Working Fluid

The principle of this system is shown in Fig.1(a), and for comparison that of the air-lift pump in Fig.1(b). The system is composed of a downcomer, an evaporating tube, a power generating equipment, and a condenser. The boiling two-phase flow occurs by heating the evaporating tube, and the specific weight in the tube decreases. The fluid circulates owing to the difference of the specific weight between the two-phase flow in the evaporating tube and the liquid in the downcomer. The thermal energy given in the evaporating tube is transformed into the potential energy and the kinetic energy at the outlet of the evaporating tube, and converted to the power by using a two-phase flow turbine. As the vapor is condensed in the condenser placed in the upper drum, the liquid level of the drum is kept constant. Also this system is operated at the low temperature in the order of the boiling temperature of the working fluid ac-

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** Professor, The Faculty of Mechanical Engineering, Kobe University, Rokkodai-cho, Nada-ku, Kobe, 657.

*** Assistant, The Faculty of Mechanical Engineering, Kobe University.

**** Graduate Student, The Faculty of Mechanical Engineering.

***** Engineer, Mitsubishi Motors Co., Kurashiki City, Okayama.

cording to the system pressure. On the other hand, in the air-lift pump the liquid is lifted by the air injected by a compressor which is driven by an external kinetic energy.

Under an assumption of neglecting the relative velocity between the vapor and the liquid, the following equation is given from the pressure balance.

$$(H_A + H_B)\rho_m = H_B\rho_l$$

where $\rho_m = V_l\rho_l/(V_l + V_g)$

Accordingly,

$$H_A/H_B = V_g/V_l \quad (1)$$

where ρ : density, V : volumetric flowrate, and subscripts l , g , and m refer to liquid, gas, and average, respectively. The maximum height of the lift $H_{\max}$ is given from Eq.(1), and power output L_{out} for both cases is expressed, neglecting the friction loss and the kinetic energy at the tube exit, as follows:

$$L_{\text{out}} = V_l\rho_l H_A g = V_g\rho_l H_B g \quad (2)$$

The vapor flowrate G_g is expressed by Eq.(3).

$$G_g = Q_0/r \quad (3)$$

where Q_0 : the total heat absorbed per unit hour, γ : heat of vaporization. From Eqs. (2) and (3), L_{out} is given.

$$L_{\text{out}}(a) = \{Q_0\rho_l g/(\rho_g r)\} H_B \quad (4)$$

The thermal efficiency of the system for the heat flux Q_1 is expressed by Eq.(5).

$$\eta_{t(a)} = \{\rho_l/(\rho_g r)\} H_B \gamma_B \quad (5)$$

where $\gamma_B = Q_0/Q_1$

On the other hand, the input power needed for air-injection in air-lift pump is $L_{\text{in}}(b) = \rho_l g H_B V_g/\eta_c$ and the system efficiency is expressed as $\eta_s(b) = L_{\text{out}}/L_{\text{in}}(b) = \eta_c$ (η_c : internal efficiency of compressor). Therefore, it is evident that the system (b) is not suitable for the power generation, because the output power is obtained from the power which drives the compressor. Now back to the system (a), it is inevitable that the thermal efficiency $\eta_s(a)$ expressed by Eq.(5) is low*1 because of the low temperature in the input energy sources, which have been exhausted without utilization up to this time. The important

(a) Principle of this system

(b) Principle of air-lift pump

Fig. 1 Power generation by means of natural circulation

Fig. 2 Comparison of working fluids

*1 Exergy efficiency $\eta_{E(a)}$ is expressed as $\eta_{E(a)} = \eta_{t(a)}/(1 - T_0/T_1) = (\text{thermal efficiency})/(\text{Carnot efficiency})$ under the condition of the constant exhaust temperature T_1 .

point is how to convert it to a higher quality energy with an economical apparatus.

The effect of the physical properties of working fluids on the power output can be estimated from the value of $\rho_l/(\rho_g\gamma)$ as seen from Eq.(5). The values of $\rho_l/(\rho_g\gamma)$ are shown in Fig.2. If the system is assumed to be operated under the conditions of the temperature of about 50 °c and the atmospheric pressure, R-113 might be more suitable for the working fluid as can be seen from Fig.2.

3. General Performance based on Approximate Analysis

The fundamental characteristics of the circulation flowrate and the maximum output will be discussed below. The following assumptions are made in the analysis.

- (i) The friction and local losses in the downcomer are negligible comparing with those in the evaporating tube.
- (ii) The self-evaporation and suppression-evaporation do not occur.
- (iii) The specific weight of liquid is constant, independently of the head H .

The momentum equations for each section between 1 - 2, or 2 - 3', ... in Fig.1(a) are expressed as follows⁽¹⁰⁾:

$$\text{Downcomer : } P_2 - P_1 = \rho_l g H_B$$

$$\text{Evaporating tube : }$$

$$P_2 - P_3' = (1 + \zeta_{in}) \rho_l W_0^2/2,$$

$$P_3' - P_3 = \lambda_0 (L/D) (W_0^2/2) \rho_l + \rho_l g L$$

$$P_3 - P_4 = \int_{3'}^4 d(W_m^2/2) \rho_m + \int_{3'}^4 \lambda_m (dl/D) (W_m^2/2) \rho_m + \int_{3'}^4 \rho_m g dl \quad (6)$$

$$P_4 - P_5 = \zeta_b (W_m^2/2) \rho_m$$

$$+ \int_4^5 \lambda_m (dl/D) (W_m^2/2) \rho_m$$

$$P_5 - P_6 = (\zeta_{out} - 1) (W_m^2/2) \rho_m$$

where ζ_{in} , ζ_b , and ζ_{out} are the pressure loss coefficients at inlet, bend, and outlet, respectively, w : velocity, λ : friction coefficient, D : diameter of the evaporating tube, L : length, and subscripts 0 and m refer to single-phase and two-phase, respectively. Summing up the above equations and assuming $P_1 = P_6$, Eq.(7) is given.

$$\begin{aligned} & \rho_l g H_B - \left(\int_3^4 \rho_m dl + \rho_l L \right) g \\ &= (1 + \zeta_{in}) (W_0^2/2) \rho_l + \int_{3'}^4 d(W_m^2/2) \rho_m \\ &+ \left\{ \lambda_0 (L/D) (W_0^2/2) \rho_l + \int_{3'}^5 \lambda_m (dl/D) (W_m^2/2) \rho_m \right\} \\ &+ \zeta_b (W_m^2/2) \rho_m + (\zeta_{out} - 1) (W_m^2/2) \rho_m \end{aligned} \quad (7)$$

Therefore,

$$\Delta P_{driv} = \Delta P_{in} + \Delta P_a + \Delta P_f + \Delta P_b + \Delta P_{out} \quad (7)'$$

where the left-hand side in the above equation

corresponds to the driving force of circulation and the right-hand side to the flow resistance due to inlet loss ΔP_{in} , acceleration loss ΔP_a , friction loss ΔP_f , bend loss ΔP_b , and outlet ΔP_{out} . Eq.(7) is a fundamental equation of the circulation, and has a complicated integral form on the length l for density, velocity, and friction losses. Therefore, the performance of the system was analyzed by the following simple method.

3.1 General expression of the performance by a simple solution

In this section, additional assumptions are made: (i) there is no slip between the vapor and the liquid and (ii) the friction coefficient of vapor-liquid two-phase flow λ_m is constant over the tube length, and the generalized performance of the circulation velocity w_0 and the output power L_{max} is expressed in dimensionless forms. In a steady operation, the driving force ΔP_{driv} , which corresponds to the left-hand side in Eq.(7), is expressed using the average density in the evaporating tube ρ_{mH} .

$$\Delta P_{driv} = H_B \rho_l g - (H_A + H_B) \rho_{mH} g \quad (8)$$

where

$$\rho_{mH} = \int_0^{H_A + H_B} \rho dl / (H_A + H_B) = \rho_e \ln(\rho_l/\rho_e) / (1 - \rho_e/\rho_l)$$

Also, each term on the right-hand side of Eq.(7) is given.

$$\begin{aligned} \Delta P_{in} &= (1 + \zeta_{in}) \rho_l (W_0^2/2), \\ \Delta P_a &= \int_{3'}^4 d(W_m^2/2) = (W_e^2 \rho_e - W_0^2 \rho_l) \\ \Delta P_f &= \lambda_0 (L/D) (W_0^2/2) \rho_l + \lambda_m \{ (H_A - L \\ &\quad + H_B + L_B)/D \} (W_m^2/2) \rho_m \\ &= \lambda_0 (L/D) (W_0^2/2) \rho_l + \lambda_m \{ (H_A - L \\ &\quad + H_B + L_B)/D \} (W_0^2/2) \rho_l (\rho_l/\rho_{mf}) \end{aligned} \quad (9)$$

where

$$\rho_{mf} = 2\rho_e \rho_l / (\rho_e + \rho_l)$$

$$\Delta P_b = \zeta_b (W_e^2/2) \rho_e = \zeta_b (W_0^2/2) (\rho_l^2/\rho_e),$$

$$\Delta P_{out} = (\zeta_{out} - 1) (W_0^2/2) (\rho_l^2/\rho_e)$$

Also, L_B is the bend length, and the subscript e refers to the outlet state of the evaporating tube. Substitution of Eqs.(8) and (9) into Eq.(7) gives

$$\begin{aligned} & H_B g (\rho_l - (H_A + H_B) \rho_{mH} / H_B) \\ &= [(1 + \zeta_{in}) + 2(\rho_l/\rho_e - 1) + \lambda_0 (L/D) \\ &\quad + \lambda_m \{ (H_A + H_B + L_B - L)/2D \} (\rho_l/\rho_e + 1) \\ &\quad + \zeta_b \rho_l/\rho_e + (\zeta_{out} - 1) \rho_l/\rho_e] (W_0^2/2) \rho_l \quad (10) \end{aligned}$$

$$\begin{aligned} \bar{W}_0 &= \frac{W_0}{\sqrt{2gH_B}} \\ &= \sqrt{\frac{(1 - h \frac{\ln K_1}{K_1 - 1}) / [\lambda_m \frac{H_B}{D} (h + \frac{L_B - L}{H_B}) \frac{K_1 + 1}{2} + 2(K_1 - 1)]}{(1 + \zeta_{in}) + \zeta_b K_1 + (\zeta_{out} - 1) K_1 + \lambda_0 \frac{L}{D}}} \end{aligned} \quad (11)$$

where $h = (H_A + H_B)/H_B$, $K_1 = \rho_l/\rho_e$, and L is preheated

length. The maximum output $L_{\max}$ is the sum of the kinetic energy at the outlet of the evaporating tube and the potential energy at the liquid level of the downcomer, and is expressed by

$$L_{\max} = L_{\max,H} + L_{\max,V} = G(H_{AG} + W_e^2/2) \\ = (\pi/4) D^2 \rho_i W_0 (H_{AG} + W_0^2 K_1^2/2) \dots (12)$$

This equation is expressed in the following dimensionless form.

$$L_{\max} = L_{\max,H} + L_{\max,V} \\ = L_{\max} / \{ \sqrt{2gH_B} H_B g (\pi/4) D^2 \rho_i \} \\ = \bar{W}_0 \{ (h-1) + \bar{W}_0^2 K_1^2 \} \dots (12)'$$

The relationship between the void fraction at the outlet of the evaporating tube f_{ge} and the outlet density ρ_e becomes

$$f_{ge} = (\rho_i - \rho_e) / (\rho_i - \rho_g) \doteq 1 - \rho_e / \rho_i \dots (13)$$

As seen from Eqs.(11) and (12)', the factors which affect w_0 and $L_{\max}$ consist of (i) ρ_l , ρ_g and L , which are determined by the working fluid and the external operation condition of heating and cooling, (ii) D , H_B , H_A , and L_B in the system design parameters, and (iii) pressure loss coefficients λ_m , ζ_{in} , ζ_b and ζ_{out} which are affected by the former factors (i) and (ii).

The effect of these factors on w_0 and $L_{\max}$ can be obtained by substituting each value into Eqs.(11) and (12)', and the generalized characteristic diagrams for the factor (i) and (ii) are shown below for the cases of $L = L_B \approx 0$.

3.2 The effect of void fraction f_{ge} at the outlet of the evaporating tube

The relationship between the dimensionless circulation velocity $\bar{w}_0$ and f_{ge} (ρ_l/ρ_e) is shown with parameters of $\lambda_m H_B/D$ and the height ratio h in Fig.3. The values of each pressure loss coefficient used in this calculation are $\zeta_{in}=0.03$, $\zeta_b=0.8$, $\zeta_{out}=1.0$ for a general case, and the thin solid line, which connects the $\circ$ mark in the figure, expresses the locus of f_{ge} which maximizes $\bar{w}_0$ at the constant values of h and $\lambda_m H_B/D$. As can be

Fig. 3 Relationship between w_0 and f_{ge}

seen from the figure, the circulation velocity $\bar{w}_0$ has a maximum value, because the driving force which is shown in the numerator of Eq.(11) increases with f_{ge} but the loss which corresponds to the denominator increases, too. The value f_{ge} which maximizes $\bar{w}_0$ is determined approximately by assuming $\partial \bar{w}_0 / \partial (\rho_l/\rho_e) = 0$ in Eq.(8) and by putting $\partial \bar{w}_0 / \partial (\rho_l/\rho_e) = 0$.

$$\frac{\rho_l}{\rho_e} \bigg|_{\bar{w}_{0\max}} = 2h - 1 \\ + \sqrt{2h \{ 2h + (\zeta_{in} - 3 - \zeta_b) / (\lambda_m \frac{H_B}{D} \frac{h}{2} + 2 + \zeta_b) \}} \dots (14)$$

Thus, it is shown that the maximum value of $\bar{w}_0$ decreases and the optimum value of f_{ge} , which maximizes $\bar{w}_0$, increases with $\lambda_m H_B/D$ and h . Also, in Fig.3 the point of intersection of the horizontal axis and $\bar{w}_0$ curve for each h (as shown in $\bullet$ mark for $h=1.0, 1.1, \dots$) is obtained from the equation of $1 - h \ln K_1 / (K_1 - 1) = 0$ by putting the driving force $= 0$ in Eq.(14), and this shows that there is no natural circulation of the working fluid at f_{ge} lower than this intersection point.

Next, the relationship between $L_{\max}$ and f_{ge} is shown in Fig.4. The broken line in the figure represents the value of $L_{\max,H}$ and the difference between the solid and the broken lines represents $L_{\max,V}$. When the heat input increases under a constant condition of h and $\lambda_m H_B/D$, and the value of f_{ge} increases, the work by the potential energy $L_{\max,H}$ increases, takes a maximum value (mark $\circ$ in the figure) and finally decreases. On the other hand, the work by the kinetic energy $L_{\max,V}$ increases monotonically owing to the increase of the outlet velocity of the fluid. Also, it is apparent that $L_{\max,H}$ increases and $L_{\max,V}$ decreases with an increase of h .

3.3 The influence of the height ratio h between the evaporating tube and the downcomer

The influence of the height ratio h on $\bar{w}_0$ and $L_{\max}$ is shown for a case of $\lambda_m H_B/D=5$ in Fig.5. As seen from the figure, when h increases at the constant f_{ge} , $\bar{w}_0$ decreases

Fig. 4 Relationship between w_0 and $L_{\max}$

owing to a decrease of the driving force and an increase of the friction loss. The lower f_{ge} is, the larger the decreasing rate of w_0 is. On the other hand, the value of L_{max} has a maximum value against the value of h owing to the relative relationship between the increase of potential energy (the fine line in the figure) and the decrease of the kinetic energy.

4. The Application to Apparatus with Practical Scale

In the previous section the general characteristics of the system, which are independent of the working fluid, system pressures and temperatures, are described. In this section the analysis is conducted, taking into account the characteristics of the friction loss and slip ratio of two-phase flow, for an apparatus which is postulated to have the dimension of a practical plant, and the performance of the apparatus using R-113 as a working fluid is described.

First, the equations for the values of the preheating length L , the local density of the two-phase flow ρ_m , the friction coefficient of the two-phase flow and the slip ratio in the fundamental equation (7) of the flow are derived. In the bottom of the downcomer the liquid is subcooled owing to the potential head and the heat losses, and the preheating zone L exists in the evaporating tube. The length L in ΔP_F term in Eq. (7) can be obtained as follows: The total heat absorbed over the preheating length is expressed by

$$\pi D q dl = (\pi/4) D^2 \rho_l W_0 c_p (T_l - T_2)$$

Accordingly,

$$T_l = 4ql / (D \rho_l W_0 c_p) + T_2 \quad \dots\dots\dots (15)$$

where q : the heat absorbed per unit surface area and unit hour and c_p : the specific heat of the liquid.

The pressure of the preheating section is given by

$$P_l = P_2 - (1 + \zeta_{in}) W_0^2 \rho_l / 2 - \lambda_0 (W_0^2 / 2) (l/D) \rho_l - \rho_l g l \quad \dots\dots\dots (16)$$

Fig. 5 Effect of h on w_0 and L_{max}

Assuming that the relationship between saturated pressures and temperatures of the working fluid is expressed by a linear equation of $T_{sat} = mP_{sat} + n$ over the pressure ranges in the system, Eq. (17) is given from Eqs. (15) and (16).

$$L = \frac{[m(P_2 - (1 + \zeta_{in}) \rho_l W_0^2 / 2) + n - T_2]}{[4q / (W_0 D \rho_l c_p) + m \rho_l \{ (\lambda_0 / D) (W_0^2 / 2) + g \}]} \quad \dots\dots\dots (17)$$

Next, the distributions of the density ρ_m and the quality x are obtained from the mass conservation equation as follows:

$$G_0 / A = W_0 \rho_l = (G_l + G_g) / A = W_g f_{gt} \rho_g + W_l (1 - f_{gt}) \rho_l$$

also, $W_{g0} = f_{gt} W_g$, $S = W_g / W_l$

$$f_{gt} = W_{g0} \rho_l / (S W_0 \rho_l + W_{g0} \rho_l - S W_{g0} \rho_g) \\ = 1 / (S W_0 / W_{g0} - S \rho_g / \rho_l + 1) \quad \dots\dots\dots (18)$$

where A : the cross-sectional area of tube, w_{g0} : the superficial velocity of vapor in the tube and s : slip ratio.

Substitution of the above equation into the correlation of $\rho_m = \rho_l - f_{gt} (\rho_l - \rho_g)$ gives

$$\rho_m = \rho_l \{ 1 - (1 - \rho_g / \rho_l) / (S W_0 / W_{g0} - S \rho_g / \rho_l + 1) \} \quad \dots\dots\dots (19)$$

where the relationship between w_{g0} and the heat input q is expressed by heat of vaporization γ .

$$W_{g0} = \{ \pi D / (A \rho_g) \} \int_0^l q dl / r = 4ql (D \rho_g r) = l / \beta \quad \dots\dots\dots (20)$$

where $\beta = D \rho_g r / 4q$,

Substitution of Eq. (20) into Eq. (19) gives

$$\rho_m = \rho_l \{ 1 - (1 - \rho_g / \rho_l) / (S W_0 \beta / l - S \rho_g / \rho_l + 1) \} \quad \dots\dots\dots (21)$$

And the quality x is obtained in a similar way.

$$x = W_{g0} \rho_g / (W_{g0} \rho_g + W_0 \rho_l) \\ = 1 / \{ 1 + (W_0 / W_{g0}) (\rho_l / \rho_g) \} \\ = 1 / \{ 1 + (W_0 \beta / l) (\rho_l / \rho_g) \} \quad \dots\dots\dots (22)$$

Next, the distribution of slip ratios is assumed to be expressed by the following Smith's correlation^{(12)*2}.

$$S = e + (1 - e) \sqrt{\frac{\rho_l}{\rho_g} \left[\frac{1 + e(1/x - 1)(\rho_g / \rho_l)}{1 + e(1/x - 1)} \right]} \quad \dots\dots\dots (23)$$

where $e = 0.4$.

*2 The values of f_{ge} for $e = 0.4$ in Eq. (23) agree well with the range of experimental data ($P = 0.1 \sim 14.8$ MPa, $G = 650 \sim 2500$ kg/m²s, $D = 6 \sim 38$ mm, steam-water two-phase flow)⁽¹⁰⁾. For the working fluid R-113 the system pressure $0.1 \sim 0.3$ MPa corresponds to the $1 \sim 2.5$ MPa of steam in ρ_g / ρ_l .

and the driving force decreases. Also, it can be seen from the figure that the higher the system pressure is, the higher the circulation velocity is. This is explained from the fact that the length of the preheat zone decreases with an increase of the system pressure as shown by dotted lines in Fig.6*3.

Next, the effect of the downcomer water level H_B on w_0 is shown in Fig.7. The driving force of the circulation ΔP_{driv} decreases and also w_0 decreases linearly with a decrease of the downcomer water level H_B . This tendency agrees well with the results of the approximate calculation in the previous section. The relationship between L_{max} and q is shown using a parameter of the inlet subcooled temperature in Fig.8. There is an optimum heat flux which maximizes L_{max} . The lower the inlet subcooled temperature is, the larger the optimum heat flux is. Also,

Fig. 8 Effect of the inlet temperature on L_{max}

Fig. 9 Effect of H_B on L_{max}

the void fraction at the outlet of the evaporating tube increases at very low heat flux and afterwards decreases gradually as shown by a dotted line in the figure. The influence of the downcomer water level H_B on L_{max} is shown in Fig.9. It can be seen from the figure that there is an optimum value of heat flux q and that the work by kinetic energy $L_{\text{max},V}$ decreases and the work by potential energy $L_{\text{max},H}$ increases with an increase of h . It is apparent that for a given length of the evaporating tube there are an optimum height ratio h and an optimum heat flux q which maximize the output.

Then, the relationship between D and the length of the tube and L_{max} , which is obtained for the optimum heat flux, is shown in Fig.10. The maximum output increases in proportion to

Fig. 10 Effects of diameter and total length of the evaporating tube on L_{max}

Fig. 11 Effect of local heating on L_{max}

*3 The value of w_0 has a maximum value in the range of higher values of q , which is not shown in Fig.6 for the case of "with preheat zone".

the length of the tube, and $2.5 \sim 3$ power of the tube diameter.

4.2 The effect of local heating of the evaporating tube

The possibility of the increase of the output by a local heating of the evaporating tube is investigated. When only the lower part of the evaporating tube is heated in the whole tube length, the driving force ΔP_{driv} increases owing to a decrease of the value of $\int_0^L \rho_m dl$ in Fig.7 but also the friction loss increases. Therefore, the output may be increased by the local heating in the relationship between the magnitude of both factors. Here the ratio of the local heating length y is defined as follows:

$$y = \frac{\text{heating length}}{\text{total length of evaporating tube}} = \frac{L_H}{H_A + H_B} \quad \dots\dots\dots(28)$$

The output L_{max} for the local heating can be calculated from Eq.(27) and a modified form of Eq.(12) which is obtained for the evaporating tube composed of a heating zone and a non-heating zone. The relationship between L_{max} and the total heat absorbed Q_0 (solid line in the figure) or the heat absorbed per unit surface area q (dotted line in the figure) is shown in Fig.11. It can be seen from the figure that the maximum value of L_{max} is affected little by the heating length y and the higher heat flux q is necessary to get the maximum output for the lower value of y as shown by dotted line, but the maximum output can be obtained by the lower value of y with the lower total heat as shown by solid lines.

5. Conclusions

As a method for the effective utilization of the low temperature heat resources, a power generating system based on the natural circulation force was proposed. The generalized dimensionless performance was shown by a simple approximated theoretical analysis, and the effects of each parameter on the performance were shown for a system of actual size, using R-113 as a working fluid.

- (1) The output increases with the tube diameter and the tube length. There is an optimum value of the heat flux for a constant tube length and also an optimum value of height ratio h for a constant heat flux to maximize the output.
- (2) The maximum output of about 35 W can be obtained for a tube of 5 m length and 37.1 mm ID at a pressure of 0.15 MPa.
- (3) The output can be increased by local heating of the lower part of the tube at a constant heat input.

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References

- (1) JSME, Report on the effective utilization of low temperature energy (in Japanese), I (1979-3), p.1.
- (2) Honma, T., et al., Natural energy (in Japanese), (1980-5), Kyoritsu Pub. Co. Ltd.
- (3) JSME, Report on the effective utilization of low-temperature energy (in Japanese), I (1979-3), p.146.
- (4) Suzuki, T., Journal of the JSME, Vol.75, No.648 (1972), p.1863.
- (5) Nicklin, D. J., Trans. Inst. Chem. Eng., Vol.41, No.1 (1963), p.29.
- (6) Stenning, A., H. and Martin, C. B., Trans ASME, Ser.A, Vol.90 (1968-4), p.106.
- (7) Todoroki, K., Trans. JSME, Vol.38, No.312 (1972-8), p.2085.
- (8) Mori, Y., et al., Trans. JSME, Vol.39, No.325 (1973-9), p.2797.
- (9) Ono, T., et al., Preprint of the JSME, No.18 (1979-3), p.133.
- (10) Ishigai, S. and Akagawa, K., Water Circulation of Boiler (in Japanese), (1953-12), Corona Pub. Co. Ltd.
- (11) Akagawa, K., Gas-Liquid Two-Phase Flow (in Japanese), (1974-5), Corona Pub. Co. Ltd.
- (12) Smith, S. L., Proc. Inst. Mech. Eng., Vol.87, No.36 (1970-1), p.647.
- (13) Martinelli, R. C. and Nelson, D. B., Trans. ASME, Vol.70, No.6 (1948), p.695.