Condensing Heat Transfer Enhancement on Vertical Spiral Double Fin Tubes With Drainage Gutters

H. Takazawa

T. Kajikawa

Ocean Energy Section,
Electrotechnical Laboratory,
AIST,
1-1-4 Umezono,
Sakura-mura, Niihari-gun,
Ibaraki, Japan 305

Spiral double-fin tubes with drainage gutters are proposed for a vertical condenser to achieve high-condensing heat transfer performance for ocean thermal energy conversion application. There aluminum tubes have 5 or 10 spiral primary fins per pitch of spiral drainage fin. The condensation occurs mainly on the 0.8-mm-high primary fins; the 2-mm-high drainage fin collects the condensate from the primary fins, and a vertical drainage gutter removes the condensate from the drainage fin. Thus performance degradation due to accumulation of condensate in the vertical direction is avoided. Experiments were carried out using R-22 (chlorodifluromethane) as the working fluid in a shell using seven aluminum tubes (900 mm in effective length and 20 mm in nominal diameter). The drainage fin pitch, the primary fin pitch, shape of primary fin, and number of drainage gutters per tube were selected as parameters. One of the tubes had a 0.2-mm-thick titanium cladding on the inside (water side). The measured working-fluid-side condensing heat transfer coefficients for these tubes were four to six times those for a smooth tube based on the outer surface area.

Introduction

An important requirement for the heat exchangers for an Ocean Thermal Energy Conversion (OTEC) system is that their design be such as to contribute to minimum overall OTEC system cost, which may be approached by increasing heat transfer performance or finding lower cost materials or fabrication techniques. Performance can be increased by decreasing the thermal resistance of the working fluid side, the tube wall, and/or the seawater side. The present work addresses performance improvement on the working-fluid side of the condenser through surface modification. In general, the mechanism of condensation is film condensation, for which the thickness of the condensate on the heat transfer surface dominates the heat transfer performance. Many configurations have been proposed for enhancement of condensing heat transfer performance. Typical configurations are a vertical fluted tube [1], a vertical fluted tube with drainage skirts [2] or collars [3], wire wrapped horizontal tube, corrugated tube [4], and horizontal tube covered by narrow grooves with many fins such as "Thermoexel C" [5].

It has been demonstrated that an enhanced horizontal tube shows good performance in the case of a single tube [6]. As OTEC heat exchangers consist of huge tube bundles, the performance degradation due to the condensate inundation greatly affects the overall heat transfer performance. Improved performance for condensing heat transfer can be obtained in the following ways:

1. Thinning the condensate film by the action of surface tension, gravity, and/or the effect of vapor flow to induce additional forced convective heat transfer.

2. Removing condensate from the tube wall as soon as possible to prevent the accumulation and inundation of condensate.

This paper presents experimental data for a new vertical condenser tube surface, which is designed to serve both of these functions. It employs spiral double fins as described later. The material is aluminum, and, in one case, an aluminum tube clad with a thin titanium layer on the inside (seawater side).

Test Facility and Tubes

As shown in Fig. 1 the test facility consists of a cold water loop, a test condenser, R-22 working fluid pump, and a vapor supply loop. To keep input cold water temperature constant with a high level of accuracy, it was necessary to equip a low-temperature chilling unit. The operating temperature in the cold water reservoir was less than 0°C. Therefore, the coolant used a 40 percent solution of ethylene glycol in water in the cold water loop.

Contributed by the Solar Energy Division for publication in the JOURNAL OF SOLAR ENERGY ENGINEERING. Manuscript received by the Solar Energy Division, May, 1984.

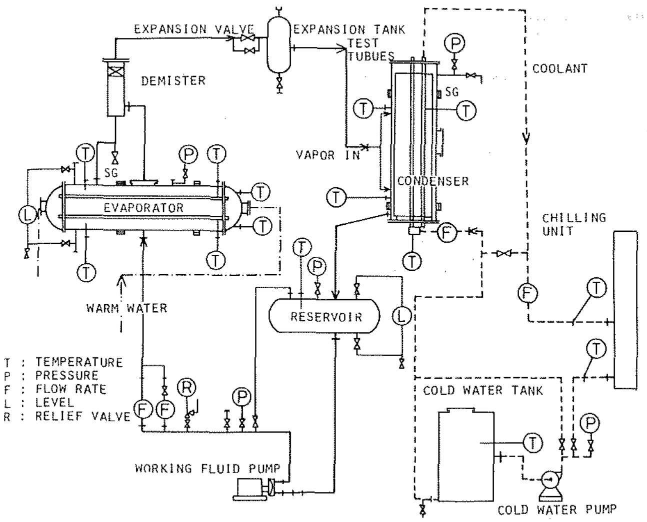

Fig. 1 Schematic design of test facility; see Fig. 2 for condenser detail.

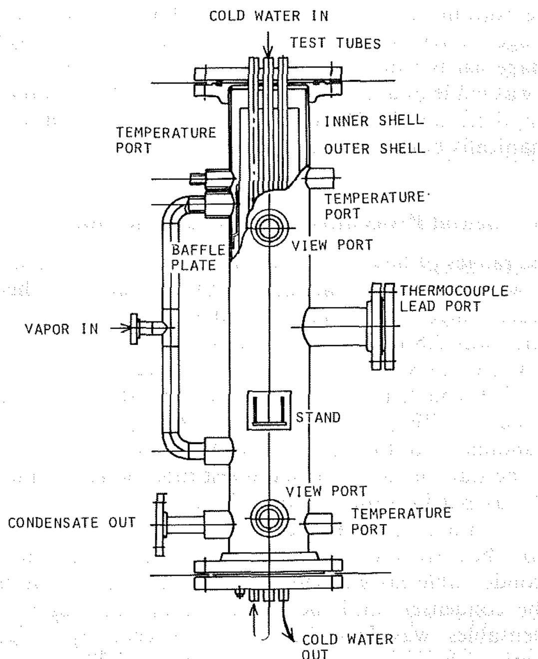

Fig. 2 Schematic diagram of condenser

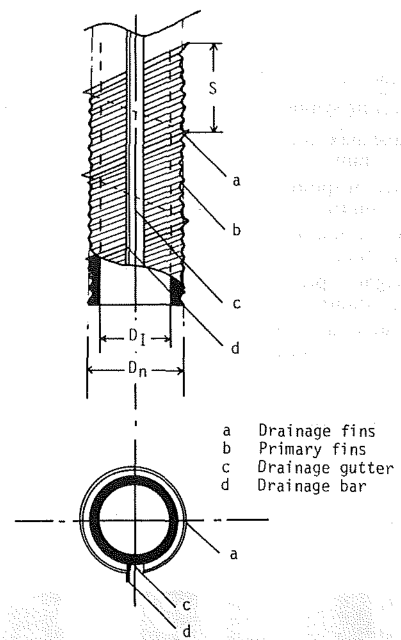

Fig. 3 Schematic cross-sectional view of test tube

The test condenser consists of an outer shell, an inner shell, and test tubes as shown in Fig. 2. The outer shell, made of stainless steel, is covered with thermal insulation material. The inner shell was made of pyrex glass to observe the surface of the tubes through the sight windows of the outer shell. Seven heat transfer tubes are installed in the shell. Each tube is individually fitted with a rubber o-ring at the top and bottom flanges of the shell. The tubes are arranged in a 60 deg triangle with a pitch-diameter of 1.5.

Vapor from a shell-and-tube evaporator heated by warm water flows into the test condenser through an expansion valve. At the inlet of the condenser vapor impinges on the buffles made of stainless steel. The vapor flows from top to bottom along each tube through the clearance between the top of the inner shell and the upper flange of the outer shell to prevent vapor impingement on the tubes, and splashing of the condensate.

The basic generic configuration of the test tubes is shown in Fig. 3. The tube has spiral primary fins, spiral drainage fins, and a vertical drainage gutter. On the primary fins, the condensate film is subjected to the effect of surface tension to produce a thin film. The drainage fin collects the condensate which falls down from a group of primary fins located in the space between drainage fin turns. A vertical drainage gutter removes the condensate accumulated by each turn of the drainage facilitates the drainage. The design parameters are:

Primary fin: number of fins per turn of drainage fin, height of fin, and shape of fin

Drainage fin: pitch of spiral and height of fin

Nomenclature

A = surface area, m^2
c_p = heat capacity, kJ/kg K
D = diameter of the tube, m
g = gravity acceleration, m/s^2
h = heat transfer coefficient, kW/m^{2}K
k = thermal conductivity, kW/mK
L = latent heat, kJ/kg
l = effective tube length, m
$\dot{m}$ = mass flow rate, kg/s
n = constant, equation (13)
Nu = Nusselt number
Pr = Prandtl number
Q = heat duty, kW
q = heat flux = Q/A, kW/m^2
Re = Reynolds number
R = heat resistance, m^{2}K/kW
s = pitch of spiral
T = temperature, K
t = flow-down time, s
U = overall heat transfer coefficient, kW/m^{2}K
V = cold water velocity, m/s

Greek Symbols

α = slope angle, rad
θ = log-mean temperature difference, K
μ = viscosity, kg/ms
ρ = density, kg/m^3
ν = dynamic viscosity, m^2/s

Subscripts

ac = actual
C = condensation
G = gas phase
H = horizontal
I,O = inlet, outlet
L = cold water side
n = nominal
r = reference
W = working fluid
$wall$ = wall

Table 1 Specifications of tubes

Tube name	Tube *A*	Tube *B*	Tube *C*	Tube *D*	Tube *E*	Tube *F* (smooth)
Plotting symbol	○	■	●	△	□	*
Tube max. o.d. (mm)	24.0	24.0	21.5	24.0	24.0	20.0
Pitch of spiral (mm)	12.7	12.7	12.7	6.35	12.7	0
Height of drainage fin (mm)	2.0	2.0	0	2.0	2.0	0
Height of primary fin (mm)	0	0.8	0.8	0.8	0.8	0
Number of primary fins per one pitch of drainage fins	0	10	10	5	10	0
Ratio of actual surface area[a] to smooth tube area	1.28	2.21	2.28	2.11	2.21	1.0
Material	Al	Al	Al	Al	Al/Ti clad	Al

[a] includes to area of both primary and drainage fins except the area of drainage bar.

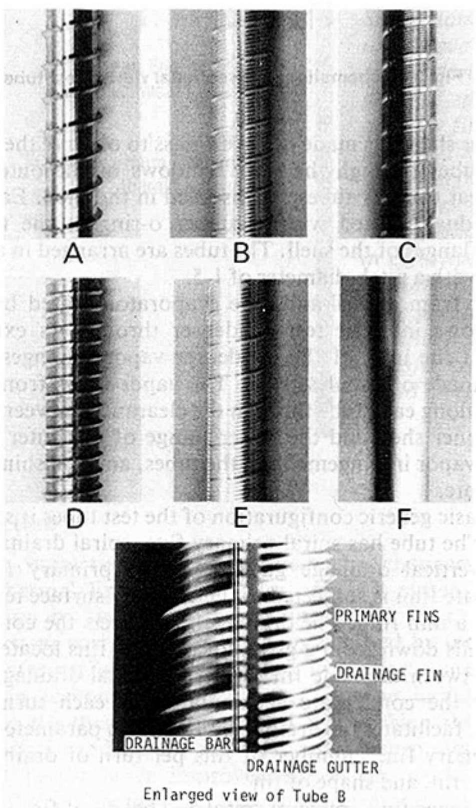

Fig. 4 Photographs of test tubes

Drainage gutter: number of longitudinal gutters per tube and width of gutter

Drainage bar: height of bar, shape of bar, and material.

As the present study is a proof-of-concept test, several kinds of tubes were tested as shown in Table 1 and photographed in Fig. 4. The numbers of drainage and primary fins were varied in this study, although shapes, sizes, and pitches of each fin and gutter are apt to be fixed more by practical constraints of current technology and cost of fabrication than by scientific design. These tubes were made of aluminum alloy A5052. The inner surface of each tube was smooth with an inner diameter of 16.0 mm. The maximum outer diameter was 24.0 mm. The outer diameter of the smooth tube was 20.0 mm. The total tube length was 990 mm with an effective length of 900 mm since the condensate stays at the bottom of the condenser in a 90-mm-thick layer. The drainage gutter was 3.0 mm in width and the height of drainage bar is 5 mm. In view of the OTEC application, one tube was made of aluminum clad with a 0.2-mm titanium film inside (i.e., on the seawater side). The titanium layer is mechanically bound by high hydraulic pressure.

Experimental Procedure and Instrumentation

The ranges of heat duty and temperature of the cold water loop were chosen to simulate OTEC conditions. The temperature range was from 4 to 10°C. The range of water velocity was 0.6 to 2.3 m/s. The water velocity was usually kept at 2.07 m/s. The heat flux was varied from 3.5 to 17.5 kW/m^2. Freon 22 (R-22) was used as the test fluid (molecular formula: $CHClF_2$; molecular weight: 86.47), since it is one of the candidates for the closed-cycle OTEC system.

At the start of each run, seven test tubes were installed and R-22 was put into the working fluid loop after most of the noncondensable gas was sucked from the loop by a vacuum pump. The system was operated for a few hours to remove the noncondensable gas through the purge valve located at the top of the condenser until the concentration ratio of the noncondensables was less than 0.007 percent by weight as determined by gas chromatograph analysis. The cold water temperature and cold water flow rate were kept constant, while heat flux was varied by the expansion valve and by controlling warm water temperature. Since the effect of super heat on the heat transfer performance was discerned in the low-heat flux range, the heat flux was varied by controlling warm water temperature. If necessary, the number of active tubes could be changed to change heat flux over a wider range.

Table 2 lists the instruments used, the measurement accuracy obtained in the experiments, and measurement quantities for the system. The measurement locations are shown in Fig. 1.

The calibration curve for each thermistor calibrated by a quartz crystal thermometer was stored in the computer. After the experimental data for each run were obtained, atmospheric pressure was measured with a Fortin type Mercury Barometer to an accuracy of ±0.01 percent, and the condensation pressure was corrected based on this measurement. The variation of the inlet cold water temperature for the cold water supply loop was maintained within ±0.1°C. ($T_{LO} - T_{LI}$) ranged from 4° C to 6°C, so that the accuracy of ($T_{LO} - T_{LI}$) was estimated within ±1 percent. The variation of the

Table 2 Instrumentation

Quantity measured	Instrument	Accuracy of measurement	Location of measurement
Temperature	Thermistor (2 mm in sheath outer diameter)	± 0.02 K	Inlet and outlet temperature
		Vapor temperature at the inlet of the condenser	
		Liquid temperature at the outlet of the condenser	
		Liquid temperature in the reservoir	
	Thermocouple (0.5 mm ϕ in sheath diameter)	± 0.1 K	Wall temperature of test tube clad with Ti
Pressure	Quartz crystal transducer	± 0.01 kg/cm^2	Condenser
Flow rate	Electromagnetic flow meter	± 0.1 m^3/h	Cold water loop

water flow rate was controlled within ± 0.5 percent of full scale. Therefore, the heat flux was measured to within ± 2.0 percent. The accuracy for the overall heat transfer coefficient was within ± 4 percent based on a consideration of variation of liquid level in the condenser and the error of the long-mean temperature difference. Experimental data were recorded by the computer for 13 seconds per series. Data for a certain condition in the run were obtained as the mean value of 5 separate series of data. All of the data were stored on computer disks.

Analytical Procedure

Ideally the heat duty of the condenser could be calculated in two independent ways as follows:

1. Based on the cold water loop:

$$Q_L = c_P m_L (T_{LO} - T_{LI}) \tag{1}$$

2. Based on the condensation of the working fluid:

$$Q_w = m_w [L + c_{pw} (T_{GI} - T_C)] \tag{2}$$

Unfortunately, measurement of the working fluid flow rate was not accurate because of a flow rate fluctuation caused by the working fluid diaphragm pump. Therefore, the heat duty was determined by equation (1), and the working fluid flow rate was calculated from equation (2). The heat flux q was calculated based on the actual outside tube surface area A_{ac} except for the surface area of the drainage bar.

$$q = Q_L / A_{ac} \tag{3}$$

Once a value of the condensate heat flux is obtained, the overall heat transfer coefficient U is calculated from the conventional definition:

$$U = q/\theta \tag{4}$$

where θ is the log-mean temperature difference:

$$\theta = (T_{LO} - T_{LI}) / ln \frac{(T_C - T_{LI})}{(T_C - T_{LO})} \tag{5}$$

Since accurate measurements of the tube wall temperature could not be at the extended surface, the cold-water side, heat transfer coefficient h_L was calculated from Colburn's empirical correlation including the effect of intake region [7].

$$\mathrm{Nu}_L = 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^{1/3} [1 + (D_I/1)^{0.7}]. \tag{6}$$

Since the overall heat transfer coefficient is defined by

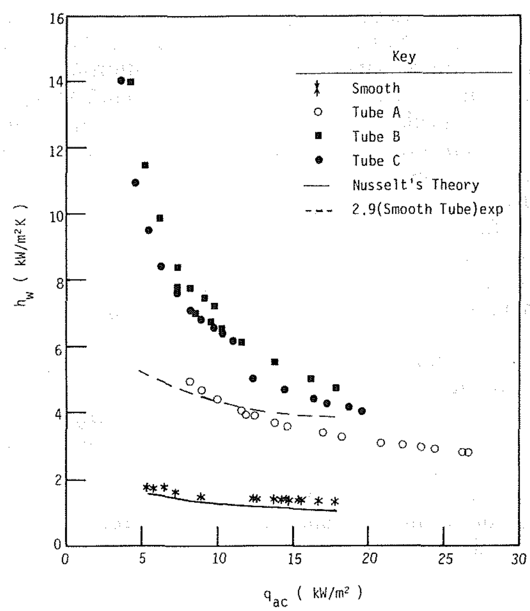

Fig. 5 Condensing heat transfer coefficient for smooth tube, Tube *A*, Tube *B*, and Tube *C*

$$\frac{1}{U} = \frac{1}{h_w} + \frac{1}{h_L}\frac{A_{ac}}{A_L} + R_{wall}' \tag{7}$$

where

$$R_{wall} = \frac{D_n}{2k_{wall}} ln (D_n / D_I), \tag{8}$$

the working-fluid heat transfer coefficient h_w can be calculated by

$$h_w = \left(\frac{1}{U} - \frac{1}{h_L}\frac{A_{ac}}{A_L} - R_{wall} \right)^{-1} \tag{9}$$

In the case of the titanium-clad tube, the wall temperature was measured, so that h_w was calculated

$$h_w = q / (T_c - T_{wall}) \tag{10}$$

Hence, the experimental values of h_w were cross-checked by comparing h_w from equation (9) with h_w from equation (10).

Nusselt's equation for filmwise condensation of vapor on a vertical smooth tube is given by:

$$h_w = 0.943\left(\frac{k_w{}^3 \rho_w{}^2 g L}{l\mu_w (T_C - T_{wall})}\right)^{1/4} \quad (11)$$

The h_w for a smooth tube can also be expressed as a function of a Reynolds number based on the condensate flow rate at the base of the tube:

$$\mathrm{Re}_w = 4m_w / \pi D_n \mu_w \quad (12)$$

The relationship between h_w and Re_w for laminar flow ($\mathrm{Re}_w < 1400$) is given by [8]:

$$h_w (\nu_w{}^2/g)^{1/3}/k_w = n\mathrm{Re}_w{}^{-1/3} \quad (13)$$

where n is 1.47 according to Nusselt's theoretical derivation. However, $n = 1.88$ is recommended by McAdams [7]. In the region $\mathrm{Re}_w > 1800$, the empirical equation recommended by Kirkbride [8] is used:

$$h_w (\nu_w{}^2/g)^{1/3}/k_w = 0.0077\mathrm{Re}_w{}^{0.4} \quad (14)$$

Equation (13) and (14) were applied to the space between one drain fin and the next for a spiral double fin tube with a drainage gutter.

Experimental Results and Discussion

Experimental values of h_w for a smooth vertical tube were compared with Nusselt's theory to confirm the aforementioned method of data reduction. The relationships between h_w and heat flux are shown in Fig. 5. The solid line represents the values calculated from Nusselt's theory, equation (11). The experimental results lie some 20 percent above the calculated values. It is generally accepted that this difference is caused by a ripple effect in the condensation film on the tube surface [8].

First, the effect of the drainage fins and gutter on performance was investigated. The experiment using Tube A, which has no primary fins, but a lot of drainage fins as shown in Fig. 4, was carried out. Tube A is theoretically regarded as a lot of very short smooth tubes, each of a length equal to the drain fin pitch as shown in Fig. 4. As h_w for a smooth tube is proportional to the -0.25 power of the tube length as shown in equation (11), the enhancement ratio can be obtained by $(s/l)^{-0.25}$. The calculated value in the case of Tube A is 2.9. The dashed line in Fig. 5 indicates 2.9 times the experimental values for a smooth tube and is compared with the experimental results for Tube A.

Second, the effect of the primary fins on performance was investigated. Results for spiral double fin tubes B and C in Fig. 5 indicate significant enhancement of h_w, especially in the small q region, which includes the OTEC operating region. Tube C is the same as tube B without drainage fins; comparison of results for tubes B and C indicates that the role of the primary fins is dominant, and that the condensate mostly flows along the ridge of the primary fins to the drainage gutter. Essentially the condensate should be removed from the heat transfer surface as soon as possible. The shape of the primary fins was not ideal in the experiment, because a part of the condensate did not drain by the shortest path. The h_w derived from equation (9) is cross-checked against the results based on wall temperature in Fig. 6; the wall temperature of one of tube E, which was located in the center of the shell, was measured by a thermocouple embedded in the wall at one third from the top of the tube. Based on this measurement the water side heat transfer coefficient and also the working fluid side heat transfer coefficient were derived. Adequate agreement is indicated.

Third, the effect of fin pitch was examined using tubes B and D. The fin pitch affects the speed of drainage of the condensate from the primary fins, as one can see by considering the case of liquid flowing down a tilted plate, of slope angle α for which the flow-down time t is

$$t = \sqrt{\frac{2s}{g}}\,\frac{1}{\sin\alpha} \quad (15)$$

where it is assumed that friction and surface tension effects

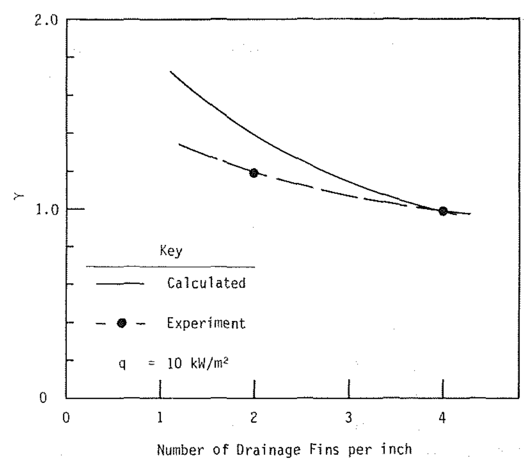

Fig. 7 Effect of the pitch of spiral on condensing heat transfer performance

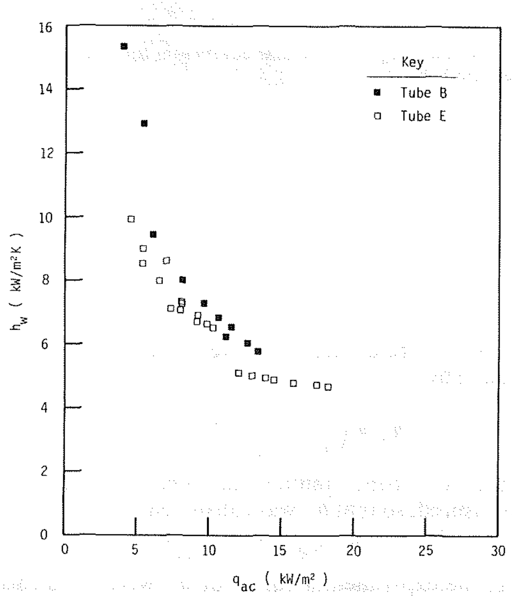

Fig. 6 Cross-check test for condensing heat transfer coefficient

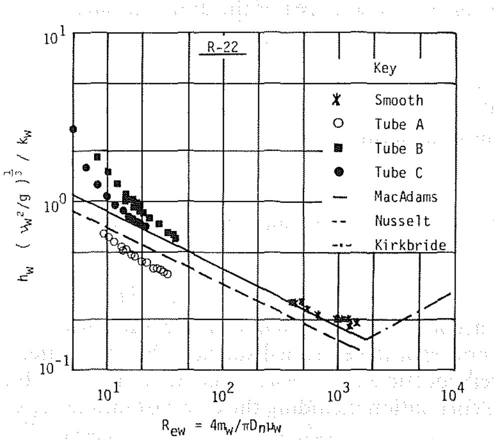

Fig. 8 Variation of nondimensional condensing heat transfer coefficient with film Reynolds number

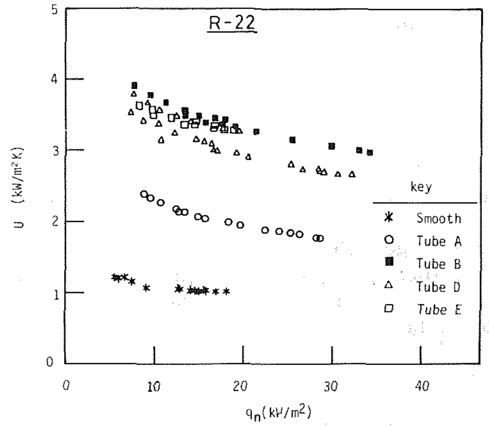

Fig. 9 Overall heat transfer coefficient versus heat duty

are negligible. The performance is inversely proportional to t. If the performance for tube B is regarded as the reference performance, the ratio of the performance for a certain tube to the reference is presented as follows:

$$\gamma = h_w / h_{wr} = t_r / t \tag{16}$$

Figure 7 shows that this model for the effect of spiral pitch can explain the experimental results quantitatively: the smaller the number of drainage fins per inch, the higher is the performance. Practically, the path of drainage of the condensate is long when the number of drainage fins per inch is small, so that the effect of friction and surface tension cannot be neglected.

Finally, the relationships between the condensing heat transfer Nusselt number and film Reynolds number are shown in Fig. 8. The performance for a smooth tube is in good agreement with the calculated value given by equation (13), with n equal to 1.88. For a spiral double-fin tube with a drainage gutter, the film Reynolds number is based on the following condensing flow rate $\dot{m}_w$:

$\dot{m}_w$ = (total condensing flow rate)/(number of drainage fins) (17)

The condensing heat transfer coefficient is based on the actual tube surface area A_{ac}. Most of the enhancement can be explained on the basis of the drainage fins and gutter. An improvement in performance also results from the primary fins as can be seen by comparing the results for tubes A and B.

The relationships between the overall heat transfer coefficient and the heat duty are shown in Fig. 9. Here U is based on the nominal tube surface area. The comparison of the test tubes with a smooth tube was carried out at a constant water velocity of 2.07 m/s. The enhancement ratios for U are 3 to 3.5.

Comparison of the results for tubes E and B shows that the thermal resistance of the wall which was clad with titanium (tube E) degrades U by about 10 pecent.

Performance and Cost Comparison With Horizontal Smooth Tubes

Since no one has proposed a vertical smooth tube condenser for OTEC, it is more appropriate to compare performance and relative heat exchanger cost using the proposed tubes to those of a condenser using horizontal smooth tubes.

The heat transfer coefficient hw_H of the horizontal smooth tubes is presented due to Nusselt's theory [7].

$$hw_H = 0.725\left[\frac{k_w{}^3 \rho_w{}^2 g L}{d\mu_w (T_c - T_{wall})}\right]^{1/4} \tag{18}$$

This value is equal to the heat transfer coefficient for the vertical smooth tubes of 0.91 πd long. In other words, the performance of the horizontal smooth tube is equivalent to that of the vertical smooth tube with drainage fin pitch of $0.91\pi d$ long. For the case d = 0.02 m, the aforementioned drainage fin pitch is 0.0572 m. The performance of a vertical smooth tube with drainage fins with a pitch less than 0.91 πd is superior to that of a horizontal smooth tube. For example, the performance of a vertical smooth tube with drainage fin for which the pitch is one-eighth of $0.91\pi d$ is 1.682 times of that of a horizontal smooth tube. The performance of the proposed tubes having both primary fins and drainage fins with drainage gutters can be much better than that of horizontal smooth tubes, because the surface tension and gravitation effects make the condensate film much thinner on the primary fins.

For a large scale horizontal tube bundle the cumulative inundation effect has to be taken into account. The mean heat transfer coefficient of an n-stage tube bundle is $n^{-1/4}$ times of the heat transfer coefficient of the single tube. On the other hand, the proposed tubes have no bundle effect but a length limitation due to the flooding of the drainage gutter. Hence, appropriate, rather complicated baffle plates collecting the condensate from the drainage gutter should be installed at proper intervals in the shell. The tube-support plates can be modified for this purpose. The relation between the width of the drainage gutter and the interval of the baffle plates should be optimized for a practical use.

To compare the cost of a whole condenser using the proposed tubes with that of a horizontal smooth tube condenser, the added manufacturing cost per tube of 6000 mm in length and 25. 4 mm in diameter is estimated first. It is assumed that 60 tubes per day are produced in one line of the manufacturing system. The manufacturing line consists of six major subsystems, in which the manufacturing technology requires no breakthroughs. Each cost of the manufacturing subsystem was roughly estimated as follows:

(1)	Tube conveyer	10 million yen
(2)	Tube end treatment machine	5
(3)	Transverse roller stage	3
(4)	Manufacturing apparatus for spiral double fins	32
(5)	Cutting machine for drainage gutter	10
(6)	Attachment of drainage bar	5
	Total	65 million yen

If the depreciation rate is 10 percent, manpower cost per hour is 5000 yen and operating factor is 0.83, the added cost of the proposed tube is calculated about 2000 yen per tube. This means that the cost ratio Φ of the proposed tube to a smooth tube is about 1.3. The mass production system will be able to reduce the added manufacturing cost.

Based on the aforementioned estimate and the test results on OTEC-1 condenser [9], the heat transfer area of the proposed condenser is roughly calculated to be about 0.54 ~ 0.59 times of that of the horizontal smooth tube condenser to handle a given heat duty. As the shell diameter is proportional to the square root of the number of tubes, the shell size is reduced to about 0.735 ~ 0.77 times of that of the horizontal smooth tube condenser. Therefore, the overall cost of a condenser using the proposed tubes is $(0.54 \sim 0.59)C_T\Phi + (0.735 \sim 0.77)C_s$, where C_T and C_s are the cost of all smooth tubes and the cost of the shell for a horizontal smooth tube condenser, respectively. Assuming that the cost ratio of C_T to C_s is 1.0, the overall cost of a condenser using the proposed tube is reduced to about 1/1.4 ~ 1/1.3 times of that of a condenser using horizontal smooth tubes. The reduction of the overall size can induce several ripple effects; these are the reduction of the platform size, the ease of piping around the heat exchanger and so on.

Concluding Remarks

These tests of vertical, spiral double fin tubes with drainage gutters using R-22 as the working fluid under OTEC conditions showed that:

1. Such tubes yielded working-fluid-side, condensing heat transfer coefficients (h_w's) four to six and one-half times those from a smooth tube based on the actual surface area. In other words, based on the nominal surface area, their performances increased eight to twelve times that for a smooth tube. In comparison with the performance of a horizontal smooth tube, the enhancement rate for the proposed tube is estimated to be about four to five for a heat flux of 10 kW/m^2.

2. The dependencies of h_w on the design parameters of the spiral double fin tubes with drainage gutters can be explained by simple models qualitatively.

3. The overall heat transfer coefficient, based on the nominal surface tube area, increased three to three and one-half times in comparison with that for a vertical smooth tube.

4. The relative heat exchanger cost using the proposed tubes to those of a condenser using horizontal smooth tubes was roughly estimated under several presumptions. In consideration of the reduction of the heat transfer area and the shell size due to the performance superiority and the cost increment of the manufacturing process, the relative cost using the proposed tubes is reduced to about $1/1.4 \sim 1/1.3$.

Acknowledgment

The authors wish to express their appreciation to Mr. K. Nishiyama, Senior Researcher, Ocean Energy Section, for his contribution to data reduction and discussion and to Dr. M. Sugiura, Head of the Energy Division, Electrotechnical Laboratory, for his encouragement and support. The authors would like to thank Mr. M. Hamaoka, Mitsubishi Heavy Industry, Inc., for his help in sending the cost information on the heat exchanger. The financial support was provided by the Sunshine Project Promotion Headquarters in the Agency of Industrial Science and Technology, Ministry of International Trade and Industry.

References

1 Webb, R. L., "A Generalized Procedure for the Design and Optimization of Fluted Gregorig Condensing Surface," *Proceedings of the Fifth OTEC Conference*, Conf-780236, VI-123-VI-145, Miami Beach, Fla., 1978.

2 Combs, S. K., "Experimental Data for Ammonia Condensation on Vertical and Inclined Fluted Tubes," ORNL-5488, 1979.

3 Kajikawa, T., Agawa, T., Takazawa, H., Amano, M., Nishiyama, K., and Honma, T., "Studies on OTEC Power System Characteristics and Enhanced Heat Transfer Performance," *Proceedings of the 6th OTEC Conference*, Conf-790631, 11.5.1-11.5.8, Washington, D.C., 1979.

4 Marto, P. J., Railly, D. J., and Fenner, J. H., "An Experimental Comparison of Enhanced Heat Transfer Condenser Tubing," *Advances in Enhanced Heat Transfer*, 79-53411, 1-9, ASME, New York, 1979.

5 Torii, T., Hirasawa, S., Kuwahara, H., Yanagida, T., and Fujie, K., "The Use of Heat Exchangers With Thermoexcell's Tubing in Ocean Thermal Energy Power Plants," ASME Paper No. 78-WA/HT-65, 1978.

6 Bergles, A. E., "Survey of Heat Transfer Characteristics of Deep Spirally Fluted Tubing," *Advances in Enhanced Heat Transfer-1981*, HTD-Vol. 18, 21-33, ASME, New York, 1981.

7 McAdams, W. H., *Heat Transmission*, McGraw-Hill, New York, 1954.

8 Katto, Y., *Heat Transfer Technology*, Yokendo Inc., Japan, 1976.

9 Lorenz, J. J., Yung, D., Howard, P. A., Panchal, C. B., and Poucher, F. W., "OTEC-1 Heat Exchanger Test Results," *Proceedings of the 8th OTEC Conference*, VIIA3-1-VIIA3-14, Washington, D.C., 1981.