

Generally, the arithmetic mean temperature difference Δt_{am} is greater than the logarithmic mean temperature difference Δt_{lm} , so that the overall coefficient of heat transmission K_{am} is less than K_{lm} .

The recovery efficiency η_{rec} of the heat exchanger can be calculated by eq. (26), and the temperature efficiency Φ_g of the heat exchanger is defined as

$$\Phi_g = (t_{g1} - t_{g2}) / (t_{g1} - t_{w1}). \quad (40)$$

The outlet gas temperature t_{g2} and outlet water temperature t_{w2} can be calculated by expressing the inlet gas temperature t_{g1} and the inlet water temperature t_{w1} as follows:

$$t_{g2} = t_{g1} - \Phi_g (t_{g1} - t_{w1}) \quad ^\circ\text{C}, \quad (41)$$

$$t_{w2} = t_{w1} + (c_{pgm} G_g / c_w G_w) \Phi_g (t_{g1} - t_{w1}) \quad ^\circ\text{C}. \quad (42)$$

The mean value of K_{lm} , η_{rec} , and Φ_g for ST-3 are obtained by experiment as shown in Table A-6 in Appendix.

$$K_{lm} = 35.1 \text{ kcal/m}^2\text{h}^\circ\text{C}, \quad \eta_{rec} = 0.715, \quad \text{and} \quad \Phi_g = 0.860.$$

(b) *Heating capacity Q_{R2} obtained from measurement of inlet and outlet water temperatures*

The heat transferred from the hot exhaust gas to the water will also be estimated by

$$Q_{R2} = c_w G_w (t_{w2} - t_{w1}) + \sum \alpha_a A_{wa} (t_{wm} - t_a) \text{ kcal/h}. \quad (43)$$

The last term of eq. (43) shows the heat loss from the hot surface area of the shell contacting water on its inner side and atmospheric air on its outer side by natural convection. The heating capacity obtained by this method occasionally entails some errors due to the difficulty of measuring the small temperature difference $(t_{w2} - t_{w1})$.

(c) *Heating capacity Q_{R3} obtained from measurement of water temperature rise in a bathtub as heating load*

A new method of obtaining the heating capacity of a heat exchanger was devised by AWANO, one of the authors. It depends on the measuring of mean water temperature rise of a bathtub with time. The principle is as follows:

For a small time interval $d\tau$, the following heat equilibrium equation should be valid:

$$\begin{aligned} (Q_{R3} + Q_p) d\tau &= [K_{am} F_i (t_{gm} - t) + Q_p] d\tau \\ &= (c_w G_w + \sum c_M G_M) dt + \sum \alpha_a A (t - t_a) d\tau, \end{aligned} \quad (44)$$

where

- Q_p : pump work supplied, kcal/h,
 G_w : total weight of water contained in the bathtub, heat exchanger, pump, and pipe lines, kg,
 t_0 : initial temperature of water in the bathtub, °C,
 t_{gm} : mean gas temperature, which is kept nearly constant but rises slightly with time, °C,
 t : instantaneous mean temperature of water, °C,
 $\sum c_M G_M$: total sum of heat capacities of materials to be heated by water, such as shell, tubes, piping, pump, and bathtub, kcal/°C,
 A : cooling surface area of shell, piping, pump, and bathtub, which contact water on the inside and atmospheric air on the outside, m²,
 τ : time, h.

From eq. (44), we get

$$\frac{dt}{d\tau} = a - bt, \quad (45)$$

where

$$a = (K_{am} F_i t_{gm} + \sum \alpha_a A t_a + Q_p) / (c_w G_w + \sum c_M G_M) \text{ } ^\circ\text{C/h}, \quad (46)$$

$$b = (K_{am} F_i + \sum \alpha_a A) / (c_w G_w + \sum c_M G_M) \text{ } 1/\text{h}. \quad (47)$$

By integrating eq. (45), the instantaneous mean water temperature t in the bathtub can be represented by

$$t = (a/b) - [(a/b) - t_0] e^{-b\tau} \text{ } ^\circ\text{C}. \quad (48)$$

For determining the constant term (a/b) in eq. (48), the following approximation is very useful. From eqs. (46) and (47), the constant term (a/b) can be estimated by neglecting the heat-loss terms $\sum \alpha_a A t_a$ and $\sum \alpha_a A$ and pump work Q_p as follows:

$$(a/b) \doteq K_{am} F_i t_{gm} / (K_{am} F_i) = t_{gm} \text{ } ^\circ\text{C}, \quad (49)$$

and from eq. (48),

$$e^{-b\tau} = (t_{gm} - t) / (t_{gm} - t_0). \quad (50)$$

The mean value of b can be determined from eq. (50) by experiment, and the overall coefficient of heat transmission K_{am} can be determined from eq. (47) by

$$K_{am} = [b(c_w G_w + \sum c_M G_M) - \sum \alpha_a A] / F_i \doteq b(c_w G_w) / F_i \text{ kcal/m}^2\text{h}^\circ\text{C}. \quad (50)$$

As an example, the instantaneous water temperature in a bathtub in an experiment with ST-3 as shown in Table A-6 in Appendix can be represented as follows:

$$\begin{aligned} t_{gm} &= 207^\circ\text{C}, & t_0 &= 31^\circ\text{C}, \\ t &= t_{gm} - (t_{gm} - t_0)e^{-bt} = 207 - 176e^{-0.406t} \text{ }^\circ\text{C}, \\ b &= 0.406 \text{ 1/h}, & c_w &= 1 \text{ kcal/kg}^\circ\text{C}, & G_w &= 286.5 \text{ kg}, & F_i &= 6.02 \text{ m}^2. \end{aligned} \quad (51)$$

From eq. (50), we get

$$K_{am} = b(c_w G_w)/F_i = 0.406(1)(286.5)/6.02 = 19.3 \text{ kcal/m}^2\text{h}^\circ\text{C}.$$

As a more accurate calculation,

$$\begin{aligned} c_M G_M &= 0.11(140) + 0.11(169) + 0.11(20) + 0.33(8) + 0.23(7.8) \\ &= 15.4 + 18.6 + 2.2 + 2.6 + 1.8 = 40.6 \text{ kcal/}^\circ\text{C}, \\ &\quad \text{(bathtub) (shell and (pump) (rubber (aluminum} \\ &\quad \quad \quad \text{tube) hose) radiator)} \\ \sum \alpha_a A &= 4.8(1.48) + [5(0.443) + 4(1.584) + 2(0.433)] + 5.5(0.834) \\ &= 7.10 + [2.22 + 6.34 + 0.87] + 4.59 = 21.1 \text{ kcal/h}^\circ\text{C}. \\ &\quad \text{(shell side) (upper (sidewall) (lower (rubber} \\ &\quad \quad \quad \text{surface) surface) hoses)} \\ &\quad \quad \quad \text{[bathtub]} \end{aligned}$$

Hence,

$$\begin{aligned} K_{am} &= [b(c_w G_w + \sum c_M G_M) - \sum \alpha_a A]/F_i = [0.406(1)(286.5) + 0.406(40.6) - 21.1]/6.02 \\ &= [116.3 + 16.5 - 21.1]/6.02 = 18.6 \text{ kcal/m}^2\text{h}^\circ\text{C}, \end{aligned}$$

and the transferred heat

$$\begin{aligned} Q_{R3} &= K_{am} F_i (t_{gm} - t) = 18.6(6.02)(207 - t) \\ &= 112(207 - t) = 23180 - 112t \text{ kcal/h}. \end{aligned} \quad (52)$$

(d) *Heating capacity Q_{R4} obtained by measuring the time necessary for completely melting an ice block floating on the water in a bathtub*

A most simple method for measuring the heating capacity of an exhaust-gas heat exchanger is obtained by measuring the time necessary to completely melt an ice block floating in the water in a bathtub and the temperature of water in the bathtub.

- τ_m : time necessary for melting, h,
- r : latent heat of fusion of ice, $r = 80 \text{ kcal/kg}$,
- t_{B0} : initial temperature of water in the bathtub, $^\circ\text{C}$,

- c_I : specific heat of ice, kcal/kg°C,
 t_{Bf} : final temperature of water in the bathtub, °C,
 t_{Bm} : mean temperature of water, $t_{Bm} = (t_{B0} + t_{Bf})/2$, °C,
 t_a : atmospheric temperature, °C,
 G_w : total weight of water contained in the water system, *i. e.*, heat exchanger, water piping, pump, and a bathtub, kg,
 G_I : weight of ice block, kg,
 t_I : initial temperature of ice, °C.

The equation of heat balance can be written as follows:

$$[Q_{R4} + Q_p + \sum \alpha_a A(t_a - t_{Bm})]\tau_m = G_I(-c_I t_I + r + c_w t_{Bf}) + c_w G_w(t_{Bf} - t_{B0}). \quad (53)$$

Hence, the transferred heat will be

$$Q_{R4} = K_{am}(t_{gm} - t_{Bm})F_i = [G_I(-c_I t_I + r + c_w t_{Bf}) + c_w G_w(t_{Bf} - t_{B0})]/\tau_m - [Q_p + \sum \alpha_a A(t_a - t_{Bm})] \text{ kcal/h.} \quad (54)$$

The last term of eq. (54), which is pump work and the transferred heat from atmosphere to water, may be ignored as a first approximation, whereby the overall coefficient K_{am} can be calculated by

$$K_{am} = [G_I(-c_I t_I + r + c_w t_{Bf}) + c_w G_w(t_{Bf} - t_{B0})]/[\tau_m(t_{gm} - t_{Bm})F_i] \text{ kcal/m}^2\text{h}^\circ\text{C.} \quad (55)$$

As an example, in Table A-5 in Appendix 3 for the heat exchanger ST-2,

$$\begin{aligned}
 G_w &= 257 \text{ kg}, & G_I &= 42.8 \text{ kg}, & t_I &= 0^\circ\text{C}, \\
 t_{B0} &= 20^\circ\text{C}, & t_{Bf} &= 20^\circ\text{C}, & t_a &= 18^\circ\text{C}, \\
 \tau_m &= 0.2388 \text{ h}, & t_{g1} &= 300^\circ\text{C}, & t_{g2} &= 39^\circ\text{C}, \\
 t_{gm} &= 170^\circ\text{C}, & t_{Bm} &= 20^\circ\text{C}, & F_i &= 6.47 \text{ m}^2.
 \end{aligned}$$

From eq. (55),

$$\begin{aligned}
 K_{am} &= [42.8(80 + 1 \times 20) + (1)(257)(20 - 20)]/[0.2388(170 - 20)(6.47)] \\
 &= [4280 + 0]/[0.2388(150)(6.47)] = 18.5 \text{ kcal/m}^2\text{h}^\circ\text{C}, \\
 Q_{R4} &= K_{am}F_i(t_{gm} - t) = (18.5)(6.47)(170 - t) \\
 &= 119.7(170 - t) = 20350 - 120t \text{ kcal/h.}
 \end{aligned} \quad (56)$$

In eq. (56), t denotes the hot-water temperature in the bathtub at any time τ .

In Appendix 2, the theoretical calculations on the design of a shell-and-tube type exhaust-gas heat exchanger ST-3 are explained.

In Appendix 3, the experimental data on the exhaust-gas heat exchangers, such as shell-and-coil type, shell-and-tube type, and vertical fin-tube type, are described.

8. Bubble Type Exhaust-Gas Heat Exchanger

8.1. Two-stage bubble type heat exchanger for making hot water and melting ice

As a result of many experiences during the last twenty years on the exhaust-gas heat exchangers for recovering exhaust-gas energy from diesel engines, the following two difficulties were recognized:

- (a) Acid corrosion of heating surfaces
- (b) Decrease of heat transfer with time due to the soot fouling of heating surfaces.

To overcome these difficulties, the authors developed a new bubble type exhaust-gas heat exchanger for JARE-18 (1976/78). In this heat exchanger, the engine exhaust gas is blown into the bottom of a body of water in a heating tank through many small holes. As the bubbles rise through the water, the heat of the bubbles is transferred to the surrounding water through the bubble surfaces. In other words, the bubble surface itself makes the heating surface, and the temperature of the bubble surface will be nearly equal to the water temperature because the heat transfer coefficient of water boundary layer is very great. Accordingly, the heat transfer from each bubble to water depends mainly on thermal conduction in the bubble. The necessary minimum depth of water required for fully transferring the heat in a bubble to the water decreases with decreasing diameter of the bubble.

In Figs. 45 and 46, a diagrammatic sectional view of the new heat exchanger and its external view are shown. In Fig. 45, the engine exhaust flows down through a center tube surrounded by a water jacket and is blown into the water of the lower heating tank through three perforated plates. The delivery gas from the lower heating tank is led to the surrounding space of an upper heating tank and is blown again into the water of the upper heating tank where its remaining heat is transferred to the water.

The gas from the upper heating tank is exhausted into the air through three collecting pipes and a flexible tube connecting them to a chimney. The cold water is fed to a float chamber in order to keep a constant water level H_U in the upper heating tank. To the side of the lower heating tank, a hot-water reservoir tank was installed, in which a pair of electrodes for detecting the water level H_L

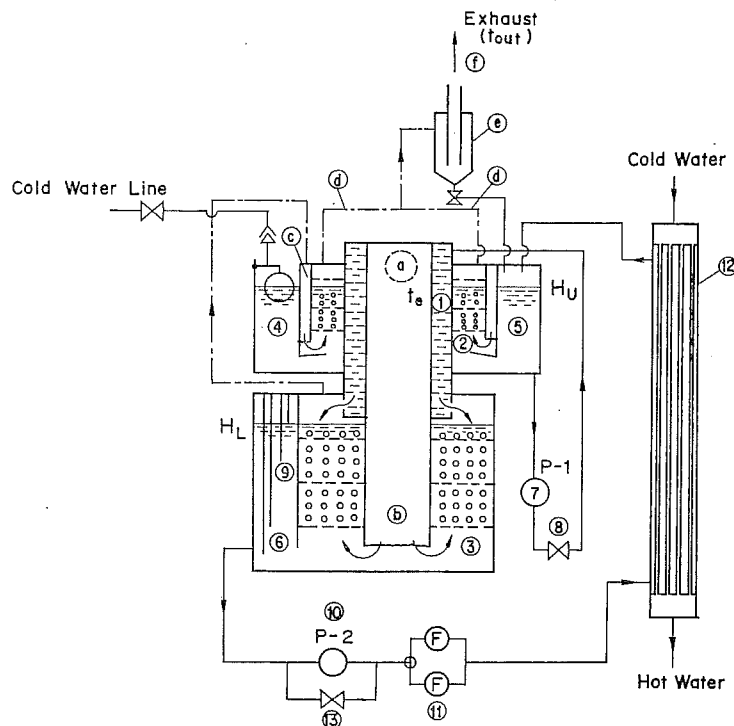


Fig. 45. Bubble type exhaust-gas heat exchanger BE-1 prepared for JARE-18 (1976/78).

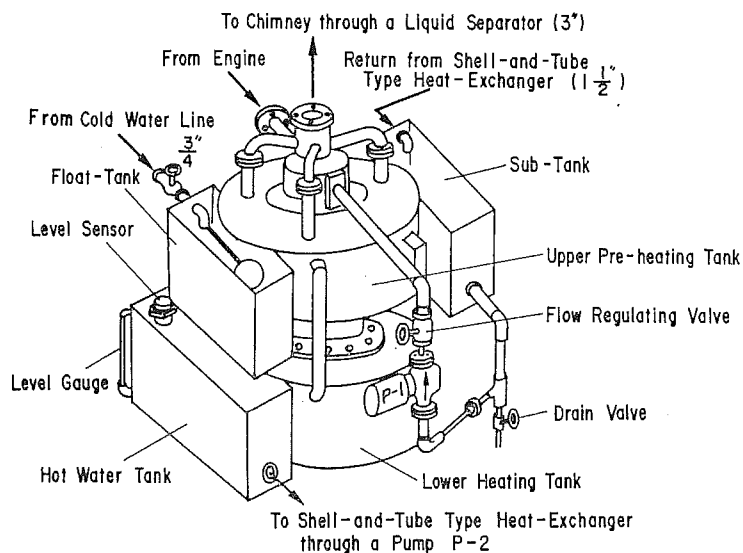


Fig. 46. Bubble type exhaust-gas heat exchanger BE-1 prepared for JARE-18.

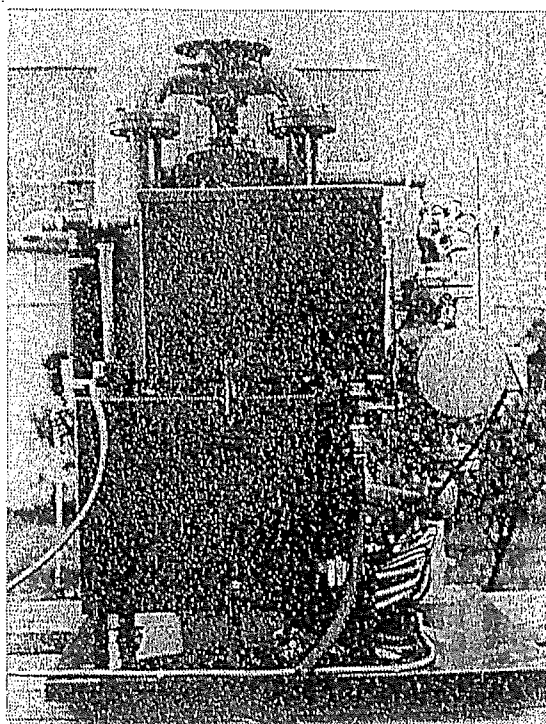


Fig. 47. Bubble type exhaust-gas heat exchanger BE-1 prepared for JARE-18.

and another pair to activate an alarm buzzer indicating an allowable minimum water level of the lower heating tank were installed. The water level of the lower heating tank H_L can be kept constant by on-off control of a motored pump P-1, which transfers the water from the upper heating tank to the lower heating tank.

The hot water in the reservoir tank is recirculated by another water pump P-2 to heating loads, such as fan-coil units for heating rooms, radiators submerged in hot-water tanks, the bathtub filled with water, or the ice-melting tank.

8.2. Experimental analysis of the bubble type heat exchanger

The same notations described in Section 5 will be also used here for analyzing the thermal performance of the new bubble type exhaust-gas-to-water heat exchanger.

The wet intake-air aspirated by a diesel engine G_a is composed of dry air G_{ad} and moisture G_{aw} as shown in eqs. (1) and (2). Fuel gas oil of G_f is injected into the engine cylinder and burnt with dry air G_{ad} with an equivalence ratio ϕ as calculated by eq. (7). The exhaust gas from the diesel engine is composed of dry exhaust gas G_{gd} and steam G_w , which is given as the sum of steam G_{gw} and G_{aw} . The weight flow rate G_{gw} represents the water vapor produced by the combustion of dry air G_{ad} and fuel G_f , which contains H as a component.

When the bubbles of the engine exhaust rise up through the water in the heating tank, the bubble gases will be saturated by steam having a partial pressure p_s corresponding to the water temperature through which the bubbles finally rise. The saturated steam G_{ws} can be calculated by eqs. (18) and (19), and the total gas flow G_{out} can be obtained by eq. (21).

If the outlet gas temperature t_{out} is low, some of the water vapor contained in the engine exhaust G_w will be condensed as the gas flows through the water, and the condensed water vapor G_{wc} can be calculated by eqs. (22) or (23). On the contrary, however, if the temperature t_{out} is higher than about 32–38°C, some part of the water in the heating tank will evaporate until the partial pressure of water vapor reaches p_s corresponding to water temperature of the upper tank. In this case, the evaporated water in the heating tank can be estimated by

$$G_{we} = G_{ws} - G_w = G_{ws} - (G_{aw} + G_{gw}) \text{ kg/h}, \quad (57)$$

and the outlet heat from the chimney can be calculated by

$$Q_{out} = (h_{gd})_{out} G_{gd} + (h'') G_{ws} \text{ kcal/h}. \quad (58)$$

The available maximum energy, Q_A recovered from the energy of the engine exhaust $Q_e = H_{g1}$ calculated by eq. (14) becomes

$$Q_A = Q_e - Q_{out} \text{ kcal/h} \quad (59)$$

$$= Q_B + Q_H + Q_{Loss} \text{ kcal/h}, \quad (60)$$

where

Q_B : heat absorbed by hot-water tank, bathtub, or ice-melting tank,

Q_H : heat required for heating the water contained in the heating tank, pipelines, and radiator,

Q_{Loss} : heat loss from heating tank, pipelines, and bathtub.

The maximum recovery efficiency of the heat exchanger is

$$(\eta_{rec})_{max} = Q_A / Q_e. \quad (61)$$

Two experiments on the new bubble type exhaust-gas heat exchanger were conducted by the authors in the Engine Laboratory of Nihon University, Tokyo.

Figs. 48 and 49 show the flow diagrams of water for No. 1 and No. 2 experiments respectively. In No. 1 experiment, the upper heating tank (HU) and the lower heating tank (HL) of the heat exchanger were connected in series. A recirculating water pump P-2 with a flow rate of 6840 kg/h fed the hot water of the heat exchanger to a radiator submerged in a bathtub containing 200 kg of water. The rise of the water temperature in the bathtub was measured and was

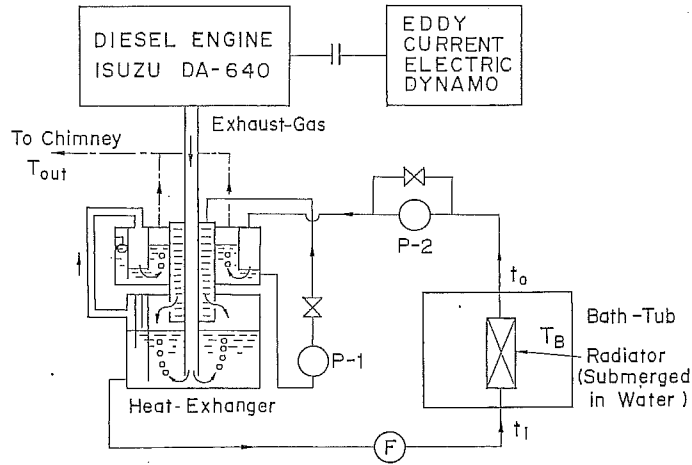


Fig. 48. Flow diagram of water for experiment No. 1 on a BE-1 bubble type exhaust-gas heat exchanger.

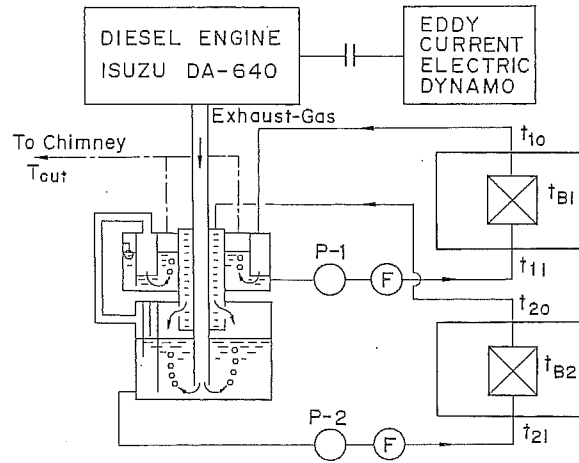


Fig. 49. Flow diagram of water for experiment No. 2 on a BE-1 bubble type exhaust-gas heat exchanger.

$t_{B0}=11.5^{\circ}\text{C}$ at the start of heating and increased gradually with time $\tau(h)$, reaching $t_B=55.4^{\circ}\text{C}$ after one hour. The temperature of the bathtub water at any time τ can be represented as in eq. (48),

$$t_B = (a/b) - [(a/b) - t_{B0}]e^{-b\tau} \text{ }^{\circ}\text{C}, \quad (62)$$

where

$$a = \frac{K_R F_R t_{Rm} + \sum \alpha_a A t_a}{c_w G_B + \sum c_M G_M} \text{ }^{\circ}\text{C/h}, \quad (63)$$

$$b = \frac{K_R F_R + \sum \alpha_a A}{c_w G_B + \sum c_M G_M} \text{ 1/h}, \quad (64)$$

$$a/b = (K_R F_R t_{Rm} + \sum \alpha_a A t_a) / (K_R F_R + \sum \alpha_a A) \div t_{Rm}, \quad (65)$$

K_R : overall coefficient of heat transmission of radiator, kcal/m²h°C,

F_R : heating surface area of radiator, m²,

t_{Rm} : mean water temperature in radiator, °C,

G_B : weight of water in the bathtub, $G_B = 200$ kg,

c_w : specific heat of water, 1 kcal/kg°C.

The other notations are the same as in eq. (44).

For No. 1 experiment,

$$t_B = t_{Rm} - (t_{Rm} - t_{B0})e^{-b\tau} = 79 - (79 - 11.5)e^{-0.98\tau} \text{ } ^\circ\text{C}. \quad (66)$$

From eq. (64),

$$K_R F_R \div b c_w G_B = (0.98)(1)(200) = 196 \text{ kcal/h}^\circ\text{C},$$

so that

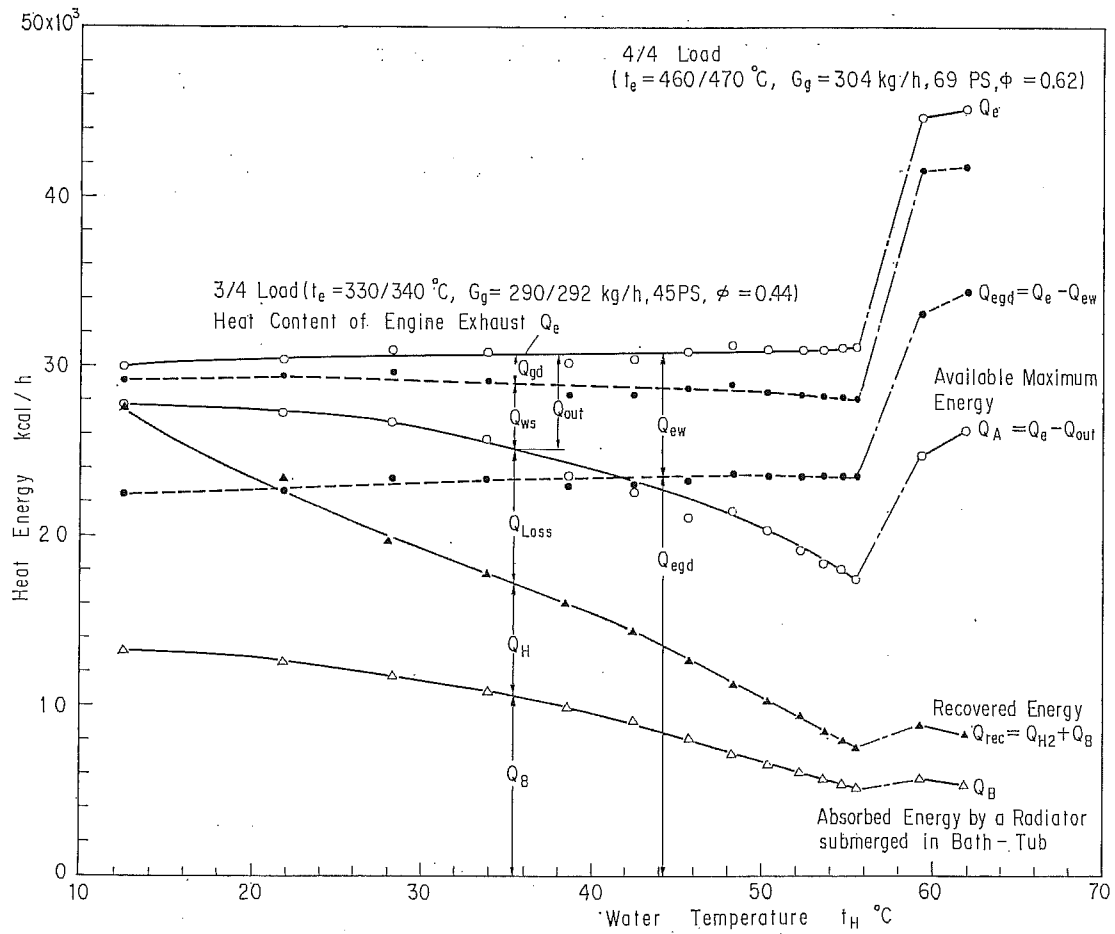


Fig. 50. Results of experiment No. 1 on a BE-1 bubble type exhaust-gas heat exchanger.

$$Q_B = K_R F_R (t_{Rm} - t_B) = 196(79 - t_B) = 15480 - 196t_B \text{ kcal/h.} \quad (67)$$

In the same way, from the measured temperature of water in the heating tank t_H , which is heated by bubble gas with temperature t_{Hb} and the heating surface area F_{Hb}

$$t_H = t_{Hb} - (t_{Hb} - t_{H0})e^{-br} = 59.5 - (59.5 - 12.6)e^{-2.80r} \text{ } ^\circ\text{C}, \quad (68)$$

$$K_{Hb} F_{Hb} \div b c_w G_H = (2.30)(1)(118) = 271 \text{ kcal/h } ^\circ\text{C},$$

$$Q_H = K_{Hb} F_{Hb} (t_{Hb} - t_H) = 271(59.5 - t_H) = 16125 - 271t_H \text{ kcal/h,} \quad (69)$$

where G_H is the weight of the water contained in the heating tank ($G_H = 118 \text{ kg}$) and its initial temperature $t_{H0} = 12.6^\circ\text{C}$. The maximum recovery efficiency $(\eta_{rec})_{max}$ is 0.923–0.562 for this system and 27600–17500 kcal/h can be recovered by this system. The results of No. 1 experiment are shown in Fig. 50.

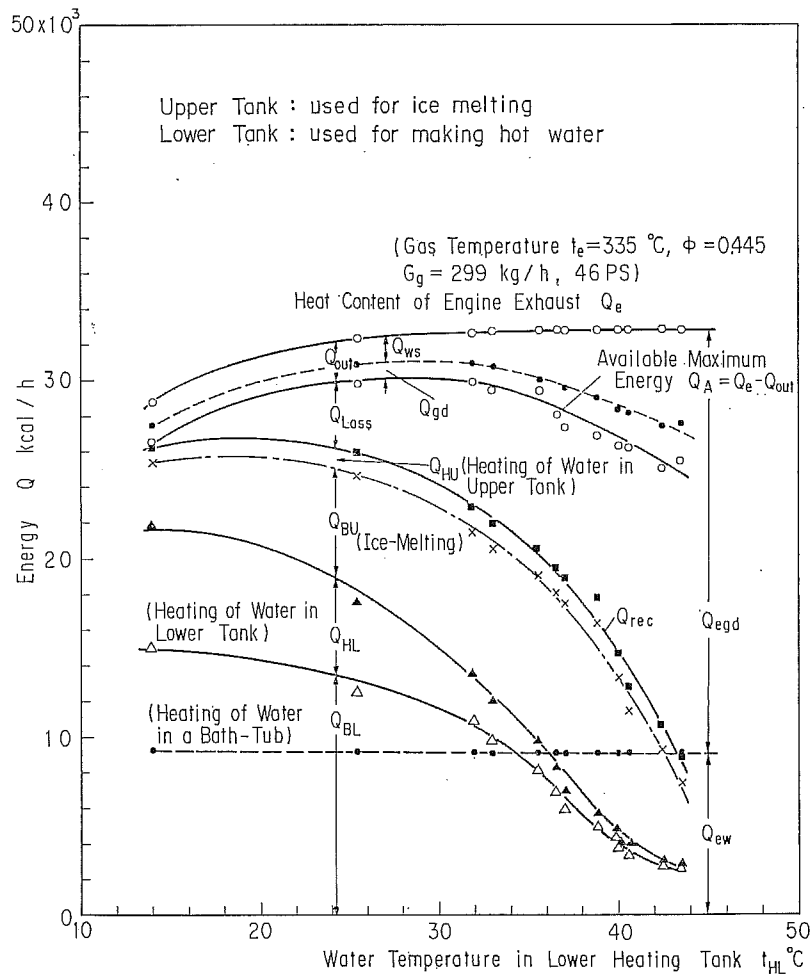


Fig. 51. Results of experiment No. 2 on a BE-1 bubble type exhaust-gas heat exchanger.

In No. 2 experiment, the upper heating tank (HU) system was used for making cold water by melting an ice block of $G_I=51.3$ kg floating on the water of $G_w=148.7$ kg at 8.0°C . On the other hand, the lower heating tank (HL) system was used for making hot water by heating the water $G_B=210.9$ kg in a bathtub.

In the upper heating tank system, a recirculating pump P-1 fed the hot water from the upper heating tank to a radiator submerged in the ice-melting tank at a rate of 6430 kg/h. After one hour, the water temperature increased from 8.0°C to 24.9°C and the ice block was completely melted.

The heat absorbed during one hour was as follows:

$$\begin{aligned} Q_{BU} &= c_w G_w (t_{w2} - t_{w1}) + G_I (80 + t_{w2}) \\ &= (1)(148.7)(24.9 - 8.0) + 51.3(80 + 24.9) = 2513 + 5381 = 7894 \text{ kcal/h.} \end{aligned}$$

During one hour, the water contained in the upper heating tank was also heated from 10.5°C to 30.5°C . The weight was $G_{HU}=56.3$ kg, so that

$$Q_{HU} = c_w G_{HU} (t_{HU2} - t_{HU1}) = (1)(56.3)(30.5 - 10.5) = 1126 \text{ kcal/h.}$$

Hence,

$$Q_{BU} + Q_{HU} = 7894 + 1126 = 9020 \text{ kcal/h.}$$

On the other hand, in the lower heating tank system, the temperature of the hot water in the other bathtub was represented by

$$t_{BL} = t_{Rm} - (t_{Rm} - t_{BL0})e^{-b\tau} = 46.4 - (46.4 - 7.7)e^{-1.83\tau} = 46.4 - 38.7e^{-1.83\tau} \text{ } ^\circ\text{C},$$

$$K_R F_R = b c_w G_B = (1.83)(1)(210.9) = 386 \text{ kcal/h}^\circ\text{C},$$

$$Q_{BL} = K_R F_R (t_{Rm} - t_{BL}) = 386(46.4 - t_{BL}) = 17910 - 386t_{BL} \text{ kcal/h.}$$

At the same time, the water contained in the lower heating tank of $G_{HL}=61.5$ kg was heated from $t_{HL0}=8.5^\circ\text{C}$ to 41.6°C after one hour.

$$t_{HL} = t_{Hb} - (t_{Hb} - t_{HL0})e^{-b\tau} = 41.6 - (41.6 - 8.5)e^{-3.5\tau} = 41.6 - 33.1e^{-3.5\tau} \text{ } ^\circ\text{C},$$

$$K_{Hb} F_{Hb} = b c_w G_{HL} = (3.5)(1)(61.5) = 215 \text{ kcal/h}^\circ\text{C},$$

$$Q_{Hb} = K_{Hb} F_{Hb} (t_{Hb} - t_{HL}) = 215(41.6 - t_{HL}) = 8944 - 215t_{HL} \text{ kcal/h.}$$

In the lower heating tank system, $Q_{BL} + Q_{Hb}$ can be absorbed. The available maximum recovery efficiency $(\eta_{\text{rec}})_{\text{max}} = 0.906 - 0.773$, which is better than in experiment No. 1 because the temperature t_{HU} is lower than that of experiment No. 1. The experimental results are shown in Fig. 51.

9. Heat Exchangers for Recovering Coolant Heat of Diesel Engines

9.1. Horizontal shell-and-tube type heat exchangers

For JARE-7 (1965/67), the authors developed a horizontal shell-and-tube type heat exchanger for recovering coolant waste heat from diesel engines driving 45-kVA electric generators. General views are shown in Fig. 52, and the design data are given in Table 11. Two sets of the coolant-to-water heat exchangers

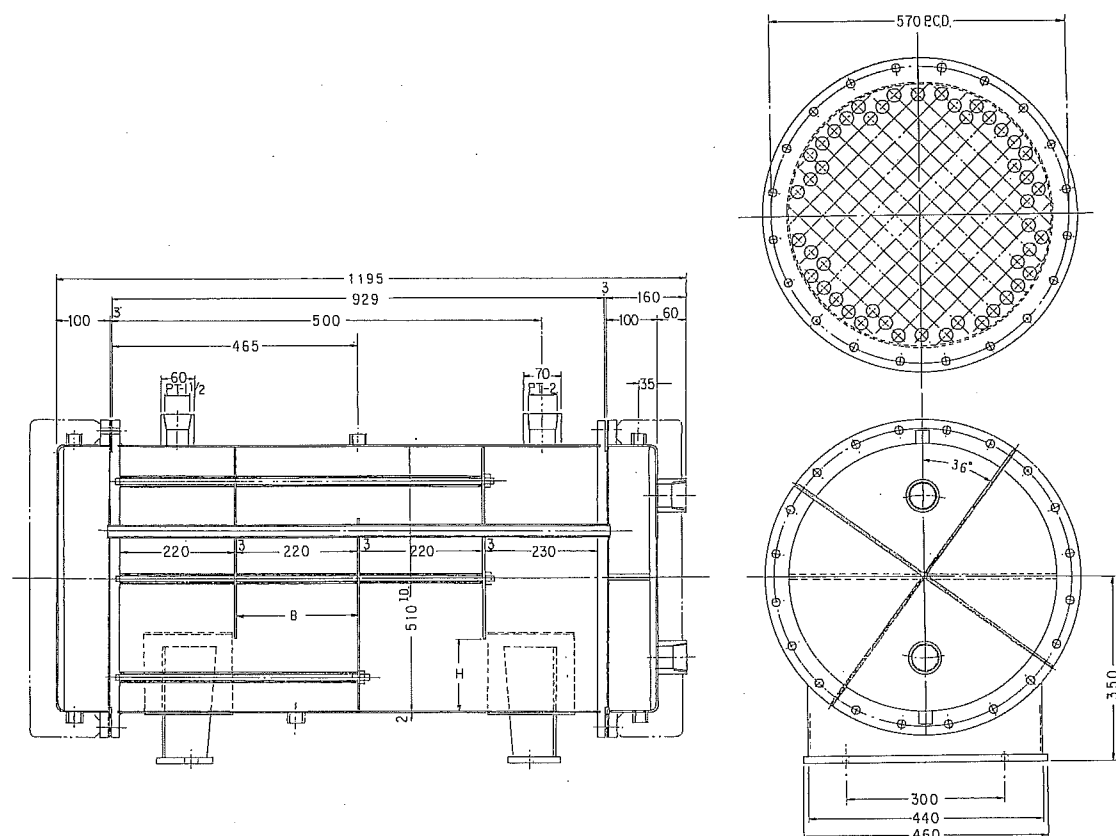


Fig. 52. Horizontal shell-and-tube type heat exchanger prepared for JARE-7 (1965/67) to recover coolant heat of diesel engine.

Table 11. Shell-and-tube type coolant heat recovery heat exchangers and secondary heat exchanger prepared for JARE.

Items	JARE-7 (1965/67)	JARE-10 (1968/70)	JARE-13 (1971/73)- JARE-20 (1978/80)
Capacity Q_c (kcal/h) Type Purpose	50000 horizontal coolant heat recovery (45 kVA)	60000 horizontal coolant heat recovery (65 kVA)	45000 vertical secondary heat exchanger
Shell side			
Shell			
Material	SUS-28	SUS-32	SUS-28TPA
Outer diam. (mm)	515	506	165.2 (6B)
Inner diam. (mm)	510	500	158.4
Total length (mm)	1255	1836	1179
Center height (mm)	350	350	—
Liquid (t_{o1} , t_{o2} °C)	coolant (80, 74)	cold water (65, 70)	cold water (12, 16)
Tube plate			
Material	SUS-28	SUS-32	SUS-28
Outer diam. (mm)	610	655	265
Thickness (mm)	18	22	22
Pipe arrangement	square	triangle	triangle
Pitch P_t (mm)	32	34	23
Number of tubes Z	164	146	26
Baffle plate			
Number N	3	7	5
Thickness	3	3	3
Distance	220×3+230	179×7+176	160×5+173
Equivalent diam. of tube space D_e (mm)	27.2	26	14.4
Effective flow area a_s (m ²)	0.02482	0.02396	0.00539
Flow rate (l/min)	160	200	200
(kg/h)	9600	12000	12000
Flow velocity w_o (m/s)	0.1035	0.1426	0.618
Reynolds number Re	7163	8897	7119
Baffle-cut ratio A/D_s	0.278	0.263	0.242
Nusselt number Nu	58.1	69.5	95.8
Heat transfer coefficient α_o (kcal/m ² h°C)	1227	1526	3380
Temperature efficiency Φ_o	0.404	0.328	0.0556
Pipe diameter	2B	2B	1B
Tube side			
Material	SUS-28TPS	SUS-32TPS	SUS-28TB
Outer diam. d_o (mm)	25	25	18

Items	JARE-7 (1965/67)	JARE-10 (1968/70)	JARE-13 (1971/73)- JAER-20 (1978/80)
Capacity Q_c (kcal/h) Type Purpose	50000 horizontal coolant heat recovery (45 kVA)	60000 horizontal coolant heat recovery (65 kVA)	45000 vertical secondary heat exchanger
Inner diam. d_i (mm)	20	21	14
Length l (mm)	929	1500	988
Number of tubes	164	146	26
Liquid (t_{i1} , t_{i2} °C)	cold water (65, 70)	coolant (80, 74)	hot water (75, 38)
Total surface area, outer F_o (m ²)	11.97	17.20	1.453
inner F_i (m ²)	9.58	14.45	1.130
Flow area A_i (m ²)	0.02576	0.02528	0.00200
Flow rate (l/min)	205	160	20.2
(kg/h)	12300	9600	1212
Flow velocity w_i (m/s)	0.1328	0.1055	0.1685
Reynolds number Re	6380	5637	5657
Nusselt number Nu	37.9	33.0	34.3
Heat transfer coefficient α_i (kcal/m ² h°C)	1082	904	1400
Overall coefficient of heat transmission K_o (kcal/m ² h°C)	464	473	736

were installed in the JARE-7 engine room as shown in Figs. 6, 7 and 8. Hot coolant of about 80/85°C was led to the shell side of the heat exchanger and returned to the inlet of the engine coolant pump after transferring its heat to the water flowing through the heating tubes. The hot water heated from about 60°C to 70°C was held in reserve in a stainless-steel hot-water tank installed in the

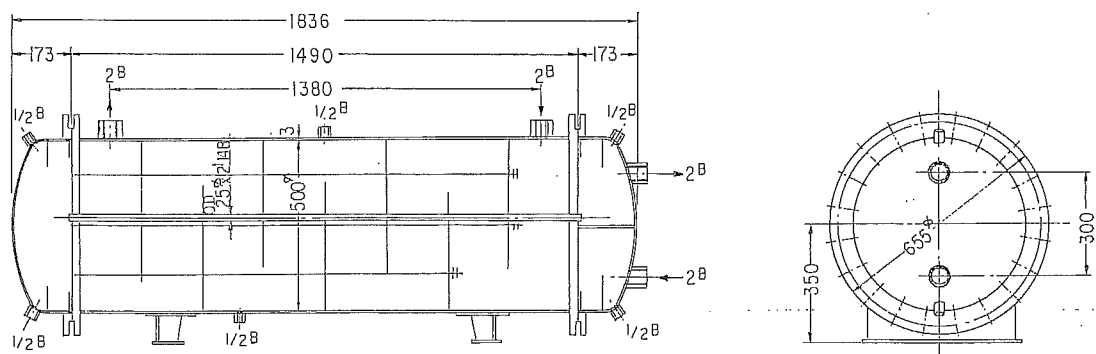


Fig. 53. Horizontal shell-and-tube type heat exchanger prepared for JARE-10 (1968/70) to recover coolant heat of diesel engine.

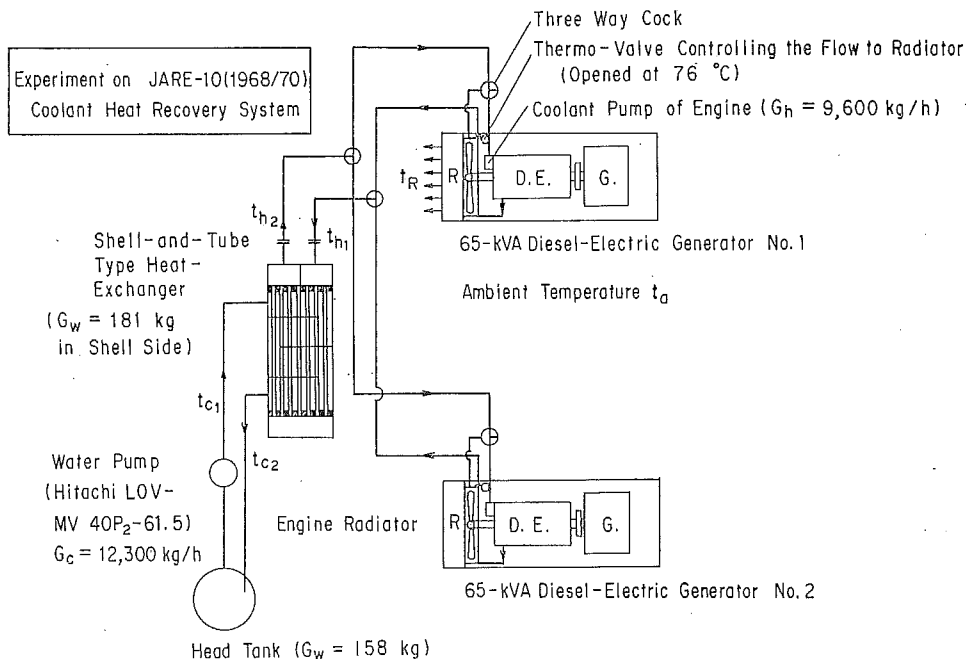


Fig. 54. Experiment on the horizontal shell-and-tube type heat exchanger prepared for JARE-10 (1968/70) to recover coolant heat of diesel engine. Experiment was carried out at Nakajima Denki Co., Ltd. in November 1968. The results are shown in Table 12.

JARE-7 engine room. The hot water was fed to the living quarters and to a bathtub.

For JARE-10 (1968/70), the same type horizontal heat exchanger, but with slightly larger heating capacity, was manufactured and installed in the JARE-9 engine room and was used for absorbing the coolant heat of a diesel engine driving a 65-kVA electric generator (Figs. 53, 54; Table 12). The hot water was stored in another hot-water tank installed in the JARE-9 engine room and used for room-heating in the living quarters of the JARE-9 engine room (Fig. 16).

For JARE-19 (1977/79), engine power was increased for driving the 110-kVA electric generator, and the waste heat recovering system was also rearranged from that shown in Fig. 16 to that in Fig. 18.

The engine coolant of the 110-kVA engine was recirculated through a head-tank installed near the engine, and the coolant in the headtank was fed to a horizontal heat exchanger installed in the JARE-7 engine room, which was separated from the JARE-9 engine room by about 40 m, and also to a radiator submerged in a 10 k/ snow melting tank installed outdoors and returned again to the head-tank. The hot water heated by the horizontal heat exchanger was stored in the hot-water tank of the JARE-7 engine room and supplied to a bathtub and the living quarters.

Another horizontal heat exchanger was used as a secondary heat exchanger in

Table 12. Results of experiments on the performance of horizontal shell-and-tube coolant-to-water heat exchanger for recovering coolant energy of diesel engine prepared for JARE-10 (1968/70).

Run No.	Time h min	Condition	V	I (A)	kVA	kW	t_{g1}	t_{h1}	t_{h2}	Δt_h	t_{c1}	t_{c2}	Δt_c	t_R	t_a	Δt_R	Q_R (kcal/h)
1	10 00	Engine starts Without H. Ex., water pump off	200	0	2	1.6	—	—	—	—	—	—	—	13.8	11.0	2.8	3500
2	10 10		200	75	15	12	170	74.0	—	—	—	—	—	15.5	—	—	—
3	10 15		200	100	20	16	235	78.5	—	—	—	—	—	29.5	13.5	16.0	20000
4	10 20		200	125	25	20	292	78.5	—	—	—	—	—	34.6	13.2	21.4	26800
5	10 25	With H.Ex., W.P. off	200	150	30	24	341	79.0	—	—	—	—	—	37.6	14.1	23.5	29400
6	10 31		200	150	30	24	348	78.0	—	—	—	—	—	18.0	13.8	4.2	5260
7	10 41		200	150	30	24	348	78.0	—	—	—	—	—	18.2	15.1	3.1	3900
8	10 55		200	150	30	24	351	78.8	63.0	15.8	—	—	—	19.8	16.1	3.7	4600
9	11 03	Thermo-valve to radiator opened	200	150	30	24	353	81.8	76.0	5.8	—	—	—	20.0	17.0	3.0	3800
10	11 08		200	150	30	24	353	82.3	78.0	4.3	—	—	—	34.1	16.8	17.3	21700
11	11 12	With H.Ex., W.P. in	200	150	30	24	—	—	—	—	—	—	—	—	—	—	—
12	11 18		200	150	30	24	333	78.0	59.5	18.5	61	63	2	19.7	16.5	3.2	4000
13	11 23		200	150	30	24	349	79.5	66.2	13.3	66	68	2	20.7	17.5	3.2	4000
14	11 28		200	150	30	24	353	80.0	71.0	9.0	71	73	2	20.3	17.1	3.2	4000
15	11 33	Thermo-valve to radiator opened	200	150	30	24	353	81.5	75.8	5.7	75	77	2	21.3	16.5	4.8	6000
16	11 45		200	150	30	24	323	82.3	78.5	3.8	78.5	80	1.5	20.8	16.5	4.3	5400
17	11 49	Engine stopped	200	150	30	24	—	—	—	—	—	—	—	—	—	—	—

1) Isuzu ZX-75 Type 65-kVA, diesel engine Isuzu DA-640T, turbocharged, 6373 cc. $N=1500$ rpm. 2) Engine coolant was led to tube side of heat exchanger $G_h=9600$ kg/h (160 l/min). 3) Cold water to be heated circulated by water pump (Hitachi LOV-MV), measured flow rate $G_c=12300$ kg/h (205 l/min). 4) Cooling air flow rate to engine radiator $G_{Ra}=7.192(3.13)(3600)(0.3878)=5215$ kg/h, $Q_R=c_{pa}G_{Ra}\Delta t_R=0.24(5215)\Delta t_R=1252 \Delta t_R$ kcal/h. 5) Recovered heat $Q_c=c_wG_c\Delta t_c=12300 \Delta t_c=24600$ kcal/h, heating area $F_o=17.20$ m². 6) Run No. 1 to Run No. 10 were preliminary running, and Run No. 11 to No. 17 were normal running state with cooling water pump running. 7) t_{h1} : Coolant inlet temp., °C, t_{h2} : Coolant outlet temp., °C, t_{c1} : Water inlet temp., °C, t_{c2} : Water outlet temp., °C, t_R : Delivery air temp. from radiator, °C, Q_R : Heating load of radiator.

as high as possible. For this purpose, the horizontal heat exchanger was used as a secondary heat exchanger. In the secondary water-to-water heat exchanger, the heat recovered from the exhaust gas was transferred again to the secondary hot water with a lower temperature than the primary water. The secondary hot water was supplied to eight radiators for room-heating in the living quarters of the JARE-9 engine room. A fan-coil unit radiated 4500 kcal/h of heat. During JARE-20 (1978/80), the waste heat recovery system was rearranged again from that shown in Fig. 18 to that in Fig. 19. In this system, the horizontal heat exchanger, which had been used as a secondary heat exchanger in the exhaust-gas energy recovering system was removed because the hot-water tank in the JARE-9 engine room could supply hot water directly to the exhaust-gas heat exchanger, but the horizontal coolant-to-water heat exchanger in the coolant-heat recovery system has been retained until now.

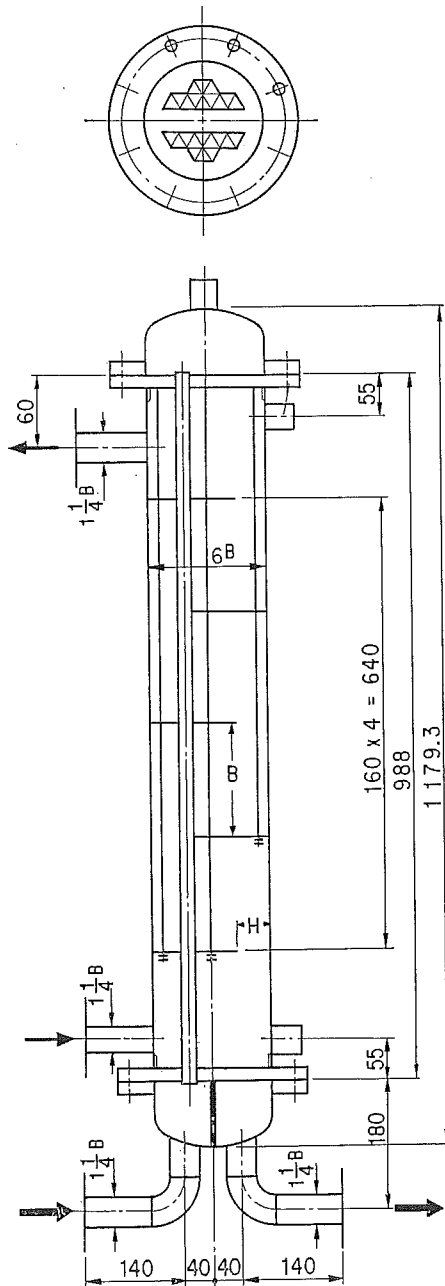


Fig. 55. Vertical shell-and-tube type heat exchanger prepared for JARE-13 (1971/73) as the secondary water-to-water heat exchanger to recover exhaust-gas energy of diesel engine.

9.2. Vertical shell-and-tube type water-to-water secondary heat exchanger

For two reasons, the first to protect the heating surface of exhaust-gas heat exchangers from acid corrosion, and the second to remove the radiators submerged in the 130-k/ outdoor cold-water reservoir tank and in the 10-k/ snow-melting tank, a small vertical shell-and-tube type heat exchanger as shown in Fig. 55 was developed by the authors for JARE-13 (1971/73).

Hot water receives heat during its flow through the coil of an exhaust-gas heat

exchanger and is led to the inside of tubes of a vertical water-to-water heat exchanger to warm the cold water of shell side, which recirculates to outdoor water tanks.

This process has been used effectively since JARE-13 up to the present to keep the water in outdoor tanks from freezing.

9.3. Estimation of thermal performance of a shell-and-tube heat exchanger

Thermal performance can be estimated by the following method:

Symbols;

- Q_c : recovered coolant heat, kcal/h,
- t : temperature of water, °C,
- G : flow rate of water, kg/h,
- F_o : total outer heating area, m²,
- d : diameter of tubes, m,
- z : total number of tubes,
- l : length of a tube, m,
- A_i : total inner sectional area of tubes, m²,
- D_s : inner diameter of shell, m,
- N : number of baffle plates,
- H : height of baffle cut, m,
- B : distance between to baffle plates, m,
- D_e : equivalent diameter for flow at the outside of tubes, m,
- α : heat transfer coefficient, kcal/m²h°C,
- K_o : overall coefficient of heat transmission, kcal/m²h°C,
- Δt_{lm} : logarithmic mean temperature difference, °C,
- V : volume flow rate, m³/s,
- a_s : area of minimum flow among three tubes (triangle arrangement) or four tubes (square arrangement), m²,
- δ_M : thickness of tube wall, m,
- λ_M : thermal conductivity of tube material, kcal/mh°C,
- Φ : temperature efficiency,
- w : velocity, m/s,
- ν : kinematic viscosity, m²/s,
- Re : Reynolds number, $Re = wD_e/\nu$,
- Pr : Prandtl number,
- Nu : Nusselt number, $Nu = \alpha D_e/\lambda$,
- λ : thermal conductivity of water, kcal/mh°C,
- c : specific heat of water, kcal/kg°C

Subscripts;

i : inside of tubes, o : outside of tubes,
 1: inlet, 2: outlet.

In a steady state, the following equations of heat balance should be satisfied:

$$Q_c = c_i G_i (t_{i1} - t_{i2}) = c_o G_o (t_{o2} - t_{o1}) \text{ kcal/h,} \quad (70)$$

$$\Phi_i = (t_{i1} - t_{i2}) / (t_{i1} - t_{o1}), \quad (71)$$

$$\Phi_o = (t_{o2} - t_{o1}) / (t_{i1} - t_{o1}). \quad (72)$$

Substituting eqs. (71) and (72) into (70), we get (JSME, 1966)

$$Q_c = c_i G_i \Phi_i (t_{i1} - t_{o1}) = c_o G_o \Phi_o (t_{i1} - t_{o1}) \text{ kcal/h.} \quad (73)$$

The heat transfer coefficient of water flowing through the tubes is given by

$$(Nu)_i = 0.023 (R_e)_i^{0.8} (Pr)_i^{0.4} = \alpha_i d_i / \lambda_i, \quad (74)$$

and the heat transfer coefficient of flow outside of the tubes can be estimated by LYDERSEN (1979)

$$(Nu)_o = 0.224 (R_e)_o^{0.587} (H/D_s)^{-0.4} (Pr)_o^{1/3} = \alpha_o D_e / \lambda_o. \quad (75)$$

In this equation, the equivalent diameter D_e for the flow across the tube bank can be represented by

(for triangle arrangement with pitch P_t)

$$D_e = [2\sqrt{3} P_t^2 / (\pi d_o)] - d_o \text{ m,} \quad (76)$$

and

(for square arrangement with pitch P_t)

$$D_e = [4 P_t^2 / (\pi d_o)] - d_o \text{ m.} \quad (77)$$

The velocity w_o past the tube bank in, or closest to, the center is

$$w_o = V_o / a_s \text{ m/s,} \quad (78)$$

where

$$a_s = (D_s / P_t) (P_t - d_o) B \text{ m}^2, \quad (79)$$

which is the minimum area between two tubes of a flow passage.

The Reynolds number of flow at the outside of tubes is calculated by

$$(R_e)_o = w_o D_e / \nu_o. \quad (80)$$