The overall coefficient of heat transmission K_o is given as the reciprocal of the total heat resistance of the heat flow through tube wall.

$$K_o = 1/\sum R \ \ \text{kcal/m}^2\text{h}^\circ\text{C}, \tag{81}$$

$$\sum R = (1/\alpha_i)(d_o/d_i) + (1/\alpha_o) + (\delta_M/\lambda_M) \ \ \text{m}^2\text{h}^\circ\text{C/kcal}, \tag{82}$$

and the heat recovered by this heat exchanger is given by

$$Q_c = K_o F_o \Delta t_{lm} \ \ \text{kcal/h}. \tag{83}$$

An example of calculation on the vertical shell-and-tube type water-to-water heat exchanger is shown in Appendix 4. The results of calculations concerning the three kinds of water-to-water heat exchangers are shown in Table 11.

10. Coolant Heat Recovery System at Mizuho Station

Mizuho Station was established on continent ice at 70°41′53″S and 44°19′54″E at an elevation of 2230 m, about 270 km inland from Syowa Station, for meteorology and glaciology investigations including the deep core drilling of continental ice.

The weather at Mizuho Station is more severe than that at Syowa Station. In 1978, the observed yearly mean maximum, mean and minimum air temperatures were −27.7°C, −31.7°C and −36.0°C, respectively, and the yearly mean values of monthly maximum and mean wind velocities were 19.8 m/s and 10.1 m/s, respectively.

JARE-11 built a semi-tubular corrugated steel hut (8.5×3.2 m) in 1970. A living hut (19.4 m^2) made of prefabricated aluminum panels and three trench rooms, *i.e.*, a glaciology laboratory (3.6×3.6 m), an ice-boring room (3.8×4.8 m), and an engine room (4.3×6.8 m) were added in 1971.

In the trench engine room, a caboose containing a 12-kVA diesel-electric generator (ZX-500B type, 3 phase, 200 V, 50 Hz, Isuzu 221 type diesel engine, 2207 cc) was installed as the main electric power plant.

A coolant heat recovery system for the 12-kVA diesel-electric generator was designed and tested by the authors in 1970 in Tokyo. It was transported to Mizuho Station set as shown in Fig. A-4 in Appendix and operated successfully. The engine ran continuously for 1593 hours from October 1971 to January 1972, consuming 5.4 k*l* fuel.

As the engine coolant, 30–50% aqua-solution of ethylene glycol was used to prevent freezing when the engine is stopped. The coolant was bypassed to the headtank installed near the engine and returned directly to the inlet of the engine coolant pump. The hot coolant in the headtank was fed to a tube type heat exchanger submerged in a stainless-steel snow-melting tank (capacity 200 *l*) and also to a fin-tube type aluminum radiator (measuring 520×405×98 mm, fin area 7.5 m^2, tube area 1.9 m^2) submerged in a bathtub for heating water and returned to the headtank by a recirculating pump (AC 200 V, 500 W, flow rate 450 *l*/min). The test results in Tokyo and at Mizuho Station are shown in Tables A-9 and

A-10 in Appendix 5. In the experiment in Tokyo, an exhaust-gas energy recovering system using a shell-and-coil heat exchanger was also examined, but only 4000 kcal/h of the exhaust-gas energy was recovered. This was because of the low gas velocity in the heat exchanger, due to the low flow rate of gas which was only 100 kg/h exhausted from the small engine.

In contrast, the recovered coolant heat reached 15000–20000 kcal/h and was sufficient for supplying heat to three fan-coil units (the capacity of one unit being 4350 kcal/h). For this reason, the coolant heat recovering system at Mizuho Station was utilized only for making washing water by melting snow blocks and also for heating water in the bathtub, but the room-heating system was not put into operation until JARE-17 (1975/77). For heating the living room, a pot-type oil stove was used and drinking water was made by melting snow in a vessel put on the stove. About 10–20 *l* of kerosene was consumed daily in the oil stove.

In JARE-15 (1973/75), a new observation hut (22.5 m^2), a trench ice-boring room (17.4 m^2), and a new trench engine room (15.5 m^2) were added. The 12-kVA diesel-electric generator was operated only from 15 May 1974 to 2 June 1974 for test running and also from 22 November 1974 to 22 January 1975 for deep ice-boring. Accordingly, the heat-recovery system was also used only during these periods. The total fuel consumption for the 12-kVA diesel was 20 drums, corresponding to 50–57.4 *l*/day.

Unfortunately, on 29 January 1975, a fire occurred in the trench engine room housing the 12-kVA set. The engine, the coolant heat recovery system, the electric power generator, and other equipment were destroyed. The cause of the accident has not yet been clarified. The camp was used temporarily by JARE-16 (1974/76).

In May 1976, Mizuho Station was reopened with two engine rooms, one for a 16-kVA diesel-electric generator (3-phase, 200 V, 50 Hz, C240 type Isuzu diesel) as an emergency plant and another for a 12-kVA diesel-electric generator (3-phase, 200 V, 50 Hz, DX-500B, C220 type Isuzu diesel) for continuous operation. Until January 1977, the 12-kVA set was operated 6600 h and consumed fuel at a rate of 60–65 *l*/day, engine oil 1 *l*/day, and antifreezing liquid 0.6 *l*/day.

The same coolant heat recovery system as in JARE-12 was prepared again for each of the two diesel-electric generators, and moreover a hot-water heating system using the coolant heat was utilized in a living room and an observation room. A fan-coil unit (single phase, 100 V, 40 W, Hitachi PF-200 type, cpapcity 4350 kcal/h) was installed for each, and the room temperature was kept between 10°C and 20°C without any other furnaces. The temperature of another observation room had been kept at 20°C by using a 1.5 kW panel heater, but it was replaced by an electric heating source powered by a windmill, as well as warm air transferred from the adjoining electrical instrument observation room by a fan

Table 13. Fuel consumptions in Mizuho Station.

Items		JARE-17 (1975/77)	JARE-18 (1976/78)	JARE-19 (1977/79)
Main engine generators (kVA)		12	12	16
Spare engine generators (kVA)		16	16	12
Yearly mean load (kW)		2.45	2.4	2.6
Maximum peak load (kW)		5.5	4.7	6.3
Number of wintering members		4	4	4
Duration of wintering		29 April 1976 26 Jan. 1977	27 Jan. 1977 1 Feb. 1978	2 Feb. 1978 16 Jan. 1979
Days of wintering (days)		242	371	349
Antarctic gas oil	(*l*)	20300	21600	19000
Antarctic kerosene	(*l*)	5800	600	130
Kerosene	(*l*)	0	0	0
New antarctic engine oil	(*l*)	330	310	420
Antifreezing liquid	(*l*)	270	270	300
Gasoline	(*l*)	0	0	22
Total	(*l*)	26700	22780	19872

at the end of November 1976 (SHIGA, 1977). In the coolant heat recovery system, the engine radiator was removed and all of the hot coolant was conducted to a 60 *l* headtank, but the other parts of the system remained as before. A pot-type oil heater was installed in the living room, but it was used only for melting snow to make drinking water. The total quantities of fuel consumed in Mizuho Station are shown in Table 13.

During 1977/78, a new research room (18.2 m^2), a storage hut for machinery, a 1-kVA engine room (4.7 m^2), and a garage for a snowmobile (7.3 m^2) were newly built. The 12-kVA and 16-kVA diesel-electric generators were operated 15640 h and 103 h respectively, without any trouble. The 16-kVA engine room was kept at 30°C by coolant heat of the 12-kVA engine and thus readied for easy starting in an emergency. Drinking water was also supplied by the snow-melting tank. The coolant heat recovery system not only saved much fuel (assumed saving 10–13 k*l* during 278 days) but ensured the safety of the wintering. By this system, the room temperatures of the living and observation rooms were kept between +15°C and +20°C without the aid of other heating sources, although the air temperature of the passage in the trench was −20°C∼−30°C. During 1978/79, the 16-kVA diesel-electric generator was used as the main source of electricity, and the 12-kVA set was reserved for emergency use. The coolant heat recovery system remained as in the previous year.

Under ordinary conditions, the temperature of the coolant returned to the engine was about 5°C lower than the outlet temperature from the engine. But the returned coolant temperature was 20–30°C lower than the outlet temperature when a snow block was thrown into the snow-melting tank as shown in Fig. A-5 in Appendix 5, as observed by Mr. TAGA. The extremely low coolant temperature occasionally caused incomplete combustion in the diesel cylinder, some lowering of engine speed, and some diesel knock. These defects were avoided by providing an automatic thermo-valve, which opened at 65°C, on the delivery side of the engine coolant system.

To keep the room air clean, the engine exhaust gas should be conducted to the outside atmosphere through a chimney which was occasionally buried in the deep snow. The snow around the chimney and snow falling on the chimney melted and drained into the engine room through a gap formed around the chimney.

To prevent the drain immersion, an empty drum, its two end plates cut off, was installed around the chimney to prevent snow from hitting the hot chimney directly.

On a part of the exhaust port of the diesel engine, a slight crack was observed. This was probably due to thermal stress caused by contact with the reversed flow of cold air through the chimney. It is advisable to use an exhaust fan which is capable of exhausting hot gases even in blizzards.

Monthly mean and peak electric loads during 1978/79 were 2–3 kW and 3–5.8 kW respectively, and fuel and oil consumption by the 16-kVA set was 48–62 *l*/day and 0.8–1 *l*/day, respectively.

11. Conclusions

The authors have been involved in the matter of the Antarctic logistics from 1956 to the present and experienced many interesting problems in the special field of engineering in Antarctica. At the start of JARE-1, the Special Committee on Mechanical Engineering, in which one of the authors was included, proposed unifying the fuel to diesel fuel and the main prime mover to diesel engines to save fuel and avoid fire accidents. According to this proposal, diesel engine of the same type was used for the electric generators and snow vehicles.

At the same time, the authors planned and manufactured some fuel-saving systems by utilizing the coolant heat and exhaust-gas energy of the diesel engines driving electric generators as much as possible.

At Syowa Station, the waste heat of the diesel engines, the capacity of which were increased from 20-kVA during JARE-1 to 110-kVA during JARE-19, was fully utilized. For these purposes, many types of exhaust-gas heat exchangers and coolant-heat recovering devices, a snow-melting tank, a hot-water tank, a bathtub, a cold-and-hot water feeding pipe, and a hot-water room-heating system were developed by the authors. By these systems, about 20–30 k*l*/year of gas oil has been saved at the present.

At Mizuho Station, coolant heat recovery systems for the 12-kVA diesel-electric generator were prepared by the authors. During JARE-15, 1-kVA gasoline engine electric generators were mainly used and a 12-kVA diesel-electric generator was also operated supplementally. Unfortunately, a fire hazard destroyed the 12-kVA engine room, while it was running, and all of the heat recovery systems, so that the camp was unmanned but was used occasionally by JARE-16.

By JARE-17, Mizuho Station was reopened and enlarged, and 12-kVA and 16-kVA diesel electric generators and coolant heat recovery systems were prepared again in the new trench engine rooms. The coolant heat of the diesel engines was supplied not only to a snow-melting tank and a bathtub, but also to fan-coil units in the living and observation rooms. The heat supplied was about 12000 kcal/h, and the room temperature has been held at $+10°C \sim 20°C$ without any other furnace. With this system at Mizuho Station, about 10–13 k*l*/year of gas oil has been saved, and the safety of wintering members in the trench rooms has been assured.

From the experiences during the past 24 years, the following conclusions can be derived:

(a) As the main power source of an antarctic station, a diesel-electric generator is most advisable and gas oil is the safest fuel. The starting of a diesel engine is easier than that of a gasoline engine, especially in cold weather below $-30°C$.

However, for starting a diesel engines exposed for a long period to the cold of $-30°C$, it was necessary to preheat it by a master heater. It is advisable to keep the emergency engine room above $+30°C$ at all times.

(b) The coolant heat from diesel engines can be easily recovered, and the quantity of heat is large and stable at any engine load.

(c) The recovery of exhaust-gas energy from diesel engines is more difficult than that of the coolant heat because of the acid corrosion of heating surfaces in heat exchangers and gradual lowering of the recovery efficiency with the increase of carbon soot adhering to the heating surface.

(d) It is most important to increase the gas velocity flowing through an exhaust-gas heat exchanger to increase its heating capacity.

(e) In an exhaust-gas heat exchanger, thermal stress should be prevented in the shell surface of the entrance chamber for hot gas. Maximum thermal stress may occur on a part of the shell against which the inlet hot gas-jet impinges.

(f) To prevent acid corrosion of the heating surfaces of an exhaust-gas heat exchanger, the water temperature should be as high as possible, at least $50°C$. For this purpose, a secondary water-to-water heat exchanger is very effective, *i.e.*, cold water should not be heated directly.

(g) Fin-tube type exhaust-gas heat exchangers made of aluminum alloys were developed, which can resist acid corrosion, because of the high temperature of their heating surfaces.

(h) A new bubble type heat exchanger for recovering exhaust-gas energy was developed. The bubble surface itself forms the heating surface, so that the problems of acid corrosion and soot fouling can be overcome completely, and the rate of heat transfer from the bubble to water is very great, but the outlet gas contains much steam vapor.

(i) A new method of estimating the heating capacity and overall coefficient of heat transmission of a heat exchanger from the temperature-time curve of water in a heated bathtub has been introduced.

(j) By means of a waste heat recovering system, about 30–40 k*l*/year of fuel can be saved at Syowa and Mizuho Stations.

APPENDIX 1

1.1. Experiments on a piping unit prepared for JARE-7 to feed hot and cold water

The piping unit as shown in Fig. 9a was tested by the authors in the low temperature testing room of the Research Institute of Transportation Technology in August 1965. The equipment for testing is shown in Fig. A-1. The room temperature was kept at about $-20°C \sim -21.0°C$ by refrigerators. The hot water heated by a 1.5 kW electric heater in a hot-water tank was recirculated by a pump through two aluminum pipes, and cold water was also recirculated by another pump through the remaining two pipes and a cold-water tank. The hot-

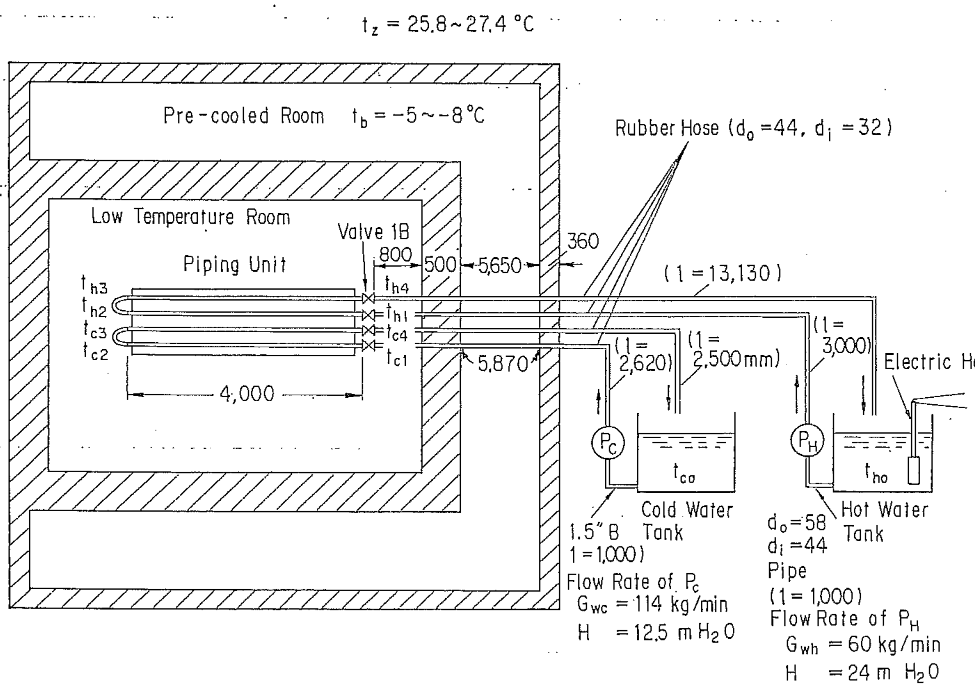

Fig. A-1. Equipment for test pipe unit prepared for JARE-7 for feeding hot and cold water (Low Temperature Testing Room of Research Institute of Transportation Technology, August 1965).

water temperatures t_{h1}, t_{h2}, t_{h3}, t_{h4} (the mean of which is t_h), cold-water temperatures t_{c1}, t_{c2}, t_{c3}, t_{c4} (the mean of which is t_c), and room temperature t_a were measured for 1.3 h in Exp. No. 1 and 1.8 h in Exp. No. 2.

The experimental results are shown in Table A-1 and Fig. A-2 and indicate that the cold water temperautre t_c can be always kept above the freezing point by using the heat loss from the two hot-water pipes. It was verified experimentally that this heating system was very effective for transporting cold water from the ice-melting tank to living or cooking quarters at Syowa Station.

Table A-1. Results of low-temperature test of piping unit as shown in Fig. 9(a). (In low temperature testing room of Res. Inst. of Transportation and Technology, Tokyo)

Exp. No. 1 (30 August 1965) $l=4$ m, $r_2=0.070$ m, $r_1=0.014$ m, $Z_h=Z_c=2$

No.	Time (h)	t_c	t_h	t_a (°C)	t_c-t_a	t_h-t_a	m	ξ_h	Heater input (kW)	Remarks
1	0	—	55.7	−22.2	—	77.9	—	—	1.487	
2	0.300	2.8	54.9	−21.1	23.9	76.0	0.314	0.686	1.530	
3	0.633	2.6	52.7	−20.4	23.0	73.1	0.315	0.685	1.522	
4	0.967	2.5	50.2	−20.1	22.6	70.3	0.321	0.679	1.496	
5	1.300	3.2	48.0	−20.0	23.2	68.0	0.341	0.659	1.496	

Exp. No. 2 (31 August 1965)

No.	Time (h)	t_c	t_h	t_a (°C)	t_c-t_a	t_h-t_a	m	ξ_h	Heater input (kW)	Remarks
1	0	0.6	61.6	−21.0	21.6	82.6	0.262	0.738	1.478	Heater in and pumps start.
2	0.167	0.7	61.7	−21.0	21.7	82.7	0.262	0.738	1.504	
3	0.333	0.9	61.4	−21.0	21.9	82.4	0.266	0.734	1.504	
4	0.500	1.9	61.3	−20.4	22.3	81.7	0.273	0.727	1.470	
5	0.667	4.7	60.7	−20.3	25.0	81.0	0.309	0.691	1.484	
6	1.83	25.1	50.1	−19.3	44.4	69.4	0.640	0.360	0	Heater is cut off, but pumps continue to run.

Flow rate: hot water pump 60 l/min, head $H=24$ m H_2O,
cold water pump 114 l/min, head $H=12.5$ m H_2O.

1.2. Theoretical analysis of the piping unit

r_1: radius of aluminum pipes, $r_1=0.014$ m,
r_2: radius of piping unit, $r_2=0.070$ m,
l: length of aluminum tubes, $l=4$ m,
Z_h: number of hot-water tubes, $Z_h=2$,
Z_c: number of cold-water tubes, $Z_c=2$,
t_h: mean temperature of hot water, °C,
t_c: mean temperature of cold water, °C,
t_a: room temperature, °C,
λ: thermal conductivity of insulating material, kcal/mh°C,
α_a: heat transfer coefficient due to natural convection from the outside of unit to air, $\alpha_a=2.35$ kcal/m²h°C (estimated),
ξ: ratio of heat loss from the external surface of unit to total heat loss of tubes, including the effect of eccentricity between an aluminum tube and an outer tube.

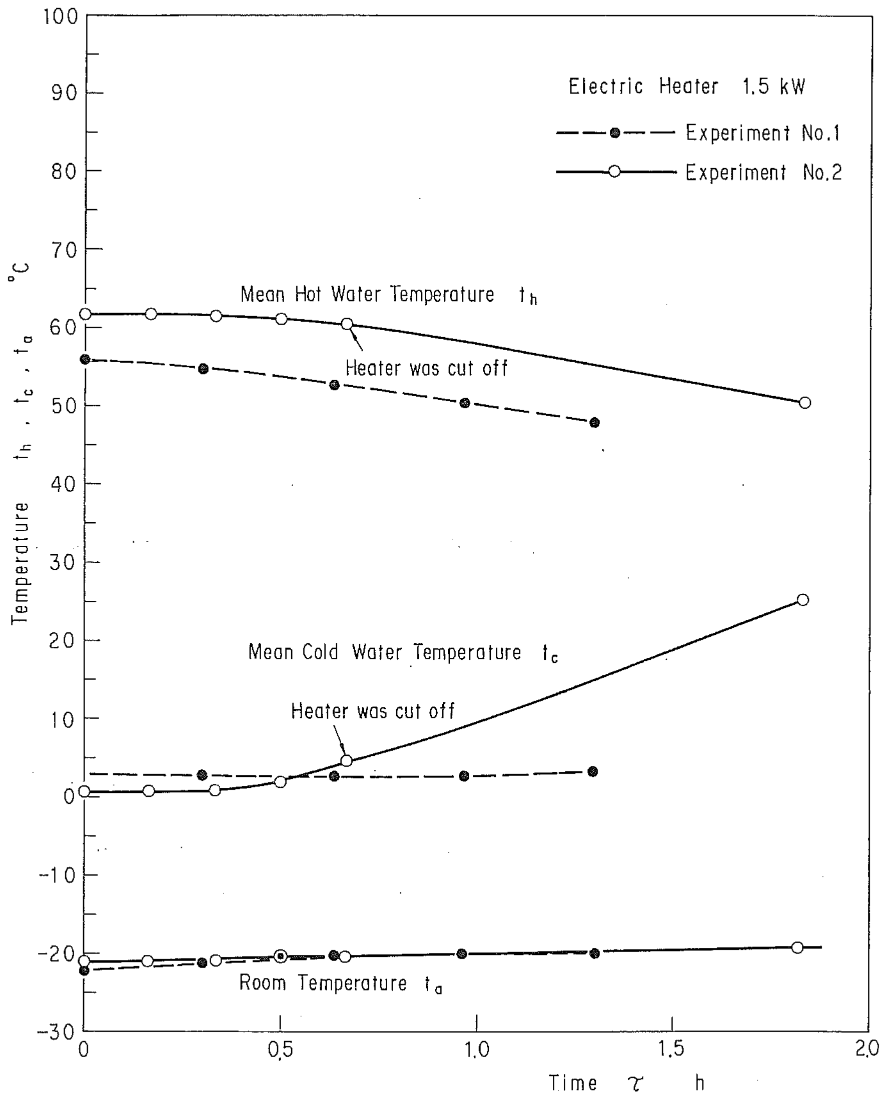

Fig. A-2. Experimental results on pipe unit to feed hot and cold water in low temperature room.

Heat which originated from the hot water transferred to the air by natural convection from the external lower surface of the pipe unit, can be represented approximately as follows:

$$Q_{ha} = 2\pi K r_2 l (t_h - t_a) Z_h \xi_h \text{ kcal/h,} \tag{A-1}$$

where

$$K = 1/[(1/\alpha_a) + (r_2/\lambda)\ln(r_2/r_1)] \text{ kcal/m}^2\text{h}^\circ\text{C.} \tag{A-2}$$

In the same way, heat which originated from the cold water transferred to air by natural convection from the external upper surface of the pipe unit, can be

Table A-2. Estimated minimum hot-water temperature t_h at atmospheric air temperature t_a for avoiding freezing of cold water in piping unit.

($Z_h=2$, $Z_c=2$, $m=0.274$, $\xi_h=0.726$)

Air temp. t_a (°C)	Hot-water temp. t_h (°C)
−40	(106)
−35	93
−30	80
−25	66
−20	53
−15	40
−10	27
− 5	14

$t_h=61.0°C$, $t_c=1.9°C$, $t_a=-20.4°C$,
$Z_h=Z_c=2$, $m=0.274$, $\xi_h=\xi_c=0.726$,
$r_2=0.070$ m, $r_1=0.014$ m, $r_2/r_1=5$,
$\ln(r_2/r_1)=1.609$, $l=4$ m,
$\alpha_a=2.35$ kcal/m²h°C, $\lambda=0.25$ kcal/mh°C,
$K=1.14$ kcal/m²h°C from eq. (A-2),
$Q_{ha}=237$ kcal/h calculated from eq. (A-1),
$Q_{ca}=65$ kcal/h from eq. (A-3),
$Q_{hc}=65$ kcal/h from eq. (A-4),

total heat loss from a piping unit

$$Q_{ha}+Q_{ca}=237+65=302 \text{ kcal/h},$$

total heat loss from hot-water tube

$$Q_{ha}+Q_{hc}=237+65=302 \text{ kcal/h}.$$

This example shows that the heat transferred from hot water to cold water in a piping unit is balanced by the heat loss from the cold water to atmospheric air by natural convection. The result of calculation of the heat loss from hot water coincides with the observed results at Syowa Station.

APPENDIX 2

2.1. Theoretical analysis of a shell-and-tube type exhaust-gas heat exchanger

By heat-transfer technology, we can estimate the approximate performance of an exhaust-gas heat exchanger as follows: As shown in eqs. (38), the total heat resistance $\sum R$ is composed of the resistance of gas-side boundary layer R_g, the resistance of adhering soot layer R_c, the resistance of tube wall R_m, and the resistance of water-side boundary layer R_w. The total resistance $\sum R$ mainly depends on R_g and R_c because R_m and R_w are smaller than the others. The theoretical calculation on ST-3 will now be set forth as an example (Fig. 36).

2.1.1. Estimation of α_g

w: velocity of gas, m/s,
d_o: outer diameter of tube, $d_o=0.0254$ m,
d_i: inner diameter of tube, $d_i=0.0214$ m,
l: length of a tube, $l=0.800\times 2=1.60$ m,
z: number of tubes in a half shell area, $z=112/2=56$,
D_s: inner diameter of the shell, $D_s=0.540$ m,
δ_c: thickness of soot layer adhering on the internal surface of tubes, m,
d_e: equivalent diameter, m,
R_e: Reynolds number of gas $R_e=w_g d_e/\nu_g$,
ν_g: kinematic viscosity of combustion gas, m²/s,
γ_{go}: specific weight of combustion gas at $t_g=0°C$ and 1 atm, kg/m³,
γ_g: specific weight of combustion gas at t_{gm} and 1 atm, kg/m³,
λ_g: thermal conductivity of combustion gas at t_{gm}, $\lambda_g=0.0325$ kcal/mh°C,
α_g: coefficient of heat transfer of gas-side boundary layer, kcal/m²h°C,
Nu: Nusselt number, $Nu=\alpha_g d_e/\lambda_g$,
Pr: Prandtl number, $Pr=0.7$ for combustion gas,
G_g: weight flow rate of combustion gas, $G_g=0.121$ kg/s (453.6 kg/h).

For the combustion gas flowing through the inside of tubes

$$\gamma_{go}=1.296\ \text{kg/m}^3 \text{ at } t_g=0°\text{C},\quad p=1\ \text{atm},\quad \phi=0.45,$$
$$\gamma_g=1.293(273/480)=0.7371\ \text{kg/m}^3 \text{ at } t_{gm}=207°\text{C},$$
$$\eta_g=\text{viscosity of combustion gas, kg s/m}^2,$$
$$\eta_g=2.50\times 10^{-6}\ \text{kg s/m}^2 \text{ for } t_{gm}=207°\text{C and } \phi=0.45.$$

g: gravity acceleration, $g=9.8$ m/s².

Kinematic viscosity of combustion gas

$$\nu_g=\eta_g g/\gamma_g=(2.50\times 10^{-6})\times 9.80/0.7371=33.2\times 10^{-6}\ \text{m}^2/\text{s}.$$

Total internal sectional area of tubes covered with soot of thickness δ_c is

$$A_c=z\pi(d_i-2\delta_c)^2/4,\ \text{m}^2.$$

The gas velocity

$$w_g=G_g/(A_c\gamma_g)=0.121/(A_c\times 0.7371),\ \text{m/s}. \qquad \text{(A-9)}$$

Reynolds number

$$R_e = w_g d_e/\nu_g = w_g(d_i - 2\delta_c)/\nu_g. \tag{A-10}$$

For turbulent flow, the following relation should hold:

$$Nu = \alpha_g d_e/\lambda_g = 0.023\ R_e^{0.8} Pr^{0.4}, \tag{A-11}$$

where the equivalent diameter is $d_e = d_i - 2\delta_c$.

The coefficient of heat transfer on the gas side can be calculated by

$$\alpha_g = \lambda_g Nu/d_e \ \ \mathrm{kcal/m^2h°C}. \tag{A-12}$$

2.1.2. Estimation of α_w

The coefficient of heat transfer on the water side can be estimated as follows: The sectional passage area for water A_w outside of tubes is

$$A_w = (\pi D_s^2/8) - (\pi d_o^2 z/4) = (\pi\ 0.540^2/8) - (\pi\ 0.0254^2 \times 56/4)$$
$$= 0.1145 - 0.0284 = 0.0861\ \mathrm{m^2},$$

and the wetted perimeter length is

$$P = (\pi D_s/2) + D_s + \pi d_o z = 5.857\ \mathrm{m}.$$

The equivalent diameter on the water side is

$$d_e = 4\ A_w/P = 0.05882\ \mathrm{m}.$$

The kinematic viscosity of water at a mean temperature $t_{wm} = 54°\mathrm{C}$ is given by

$$\nu_w = 0.514 \times 10^{-6}\ \ \mathrm{m^2/s}.$$

When the volume flow rate of water supplied by a pump is represented by Q_w $\mathrm{m^3/s}$, the water velocity w_w is

$$w_w = Q_w/[\pi d_e^2/4] = 0.001/0.002717 = 0.368\ \mathrm{m/s}\ \ \text{for}\ \ Q_w = 0.001\ \mathrm{m^3/s}.$$

The Reynolds number of the water flow is

$$R_e = w_w d_e/\nu_w = 42100,$$

and for water of $t_{wm} = 54°\mathrm{C}$, the Prandtl number $Pr = 3.4$, and thermal conductivity $\lambda_w = 0.556\ \mathrm{kcal/mh°C}$, so that the Nusselt number can be calculated by

$$Nu = 0.023\ R_e^{0.8} Pr^{0.4} = 0.023(42100)^{0.8}(3.4)^{0.4} = 188.$$

The coefficient of heat transfer on the water side can be obtained as follows:

$$\alpha_w = \lambda_w Nu/d_e = 0.556 \times 188/0.05882 = 1780\ \mathrm{kcal/m^2h°C}.$$

2.1.3. Heat resistances of tube wall and soot layer

Thermal conductivity of stainless steel $\lambda_m = 14$ kcal/mh°C.
Thickness of tubes $\delta_m = 0.002$ m.
Thermal conductivity of carbon soot layer $\lambda_c = 0.10$ kcal/mh°C.
Thickness of soot layer δ_c m.
Thermal resistance of stainless-steel tube $R_m = \delta_m/\lambda_m = 0.002/14 = 0.000143$.
Thermal resistance of soot layer $R_c = \delta_c/\lambda_c = 10\,\delta_c$ m²h°C/kcal.

2.1.4. Overall coefficient of heat transmission K_{lm}

The total heat resistance

$$\sum R = (1/\alpha_w)(d_i/d_o) + (1/\alpha_g)(d_i/d_c) + R_m + R_c. \tag{A-13}$$

Overall coefficient of heat transmission

$$K_{lm} = 1/\sum R \ \text{kcal/m}^2\text{h°C}. \tag{A-14}$$

The total inner surface area of the stainless-steel tubes in a clean state contacting exhaust gas directly, is adopted as a standard heating area

$$F_i = z(\pi d_i l) = 56(\pi \times 0.0214 \times 1.60) = 6.02 \text{ m}^2,$$

sectional area for gas flow

$$A_i = z(\pi d_i^2/4) = 56(\pi\ 0.0214^2/4) = 0.02014 \text{ m}^2,$$

gas velocity

$$w_g = G_g/(A_i \gamma_g) = 0.121/(0.02014 \times 0.7371) = 8.15 \text{ m/s},$$

Reynolds number

$$R_e = w_g d_i/\nu_g = (8.15)(0.0214)/(33.2 \times 10^{-6}) = 5253,$$

Nusselt number

$$Nu = 0.023\ R_e^{0.8} Pr^{0.4} = 0.023(5253)^{0.8}(0.7)^{0.4} = 0.023(946.9)(0.867) = 18.9,$$

coefficient of heat transfer on gas side

$$\alpha_g = \lambda_g(Nu)/d_i = (0.0325)(18.9)/(0.0214) = 28.7 \text{ kcal/m}^2\text{h°C},$$

total heat resistance

$$\sum R = (1/\alpha_w)(d_i/d_o) + R_m + (1/\alpha_g) = (1/1780)(0.0214/0.0254) + (0.000143) + (1/28.7)$$
$$= 0.000473 + 0.000143 + 0.03484 = 0.03546 \text{ m}^2\text{h°C/kcal},$$

overall coefficient of heat transmission

Table A-3. Theoretical calculation of the thermal resistance of the shell-and-tube type exhaust-gas heat exchanger ST-3 with soot adhesion.

δ_c (m)	d_e (m)	A_c (m^2)	w_g (m/s)	R_e	Nu	α_g	F_c (m^2)	R_g	R_c	$\sum R$	K_{lm} (kcal/m^{2}h°C)
0	0.0214	0.02014	8.15	5253	18.9	28.7	6.02	0.0348	0	0.0355	28.2
0.001	0.0194	0.01655	9.92	5797	20.4	34.2	5.46	0.0322	0.010	0.0429	23.2
0.002	0.0174	0.01332	12.3	6446	22.2	41.5	4.90	0.0296	0.020	0.0502	19.9
0.003	0.0154	0.01043	15.7	7283	24.5	51.7	4.34	0.0268	0.030	0.0575	17.4
0.004	0.0134	0.00790	20.8	8395	27.5	66.7	3.77	0.0239	0.040	0.0645	15.5
0.005	0.0114	0.00572	28.7	9855	31.2	88.9	3.21	0.0211	0.050	0.0717	13.9

$$K_{lm}=1/\sum R=1/0.03546=28.2\ \text{kcal/m}^2\text{h}^\circ\text{C}.$$

The heat transferred can be calculated by

$$Q_r=K_{lm}F_i\Delta t_{lm}=(28.2)(6.02)\Delta t_{lm}=170\ \Delta t_{lm}\ \ \text{kcal/h}.$$

In Table A-3 the effect of soot accumulation on the inner surface of tubes on the overall coefficient of heat transmission K_{lm} is shown, and the decrease of K_{lm} with soot accumulation coincides with the results shown in Figs. 33 and 34.

APPENDIX 3

3.1. Experimental data on the performance of shell-and-coil type exhaust-gas heat exchanger

The thermal performance of a shell-and-coil type exhaust-gas heat exchanger, which was used at Syowa Station during JARE-1 and JARE-18 as described in Subsection 6.1, was tested in the Engine Laboratory of Nihon University in Tokyo, by using an Isuzu DA-640 diesel engine. The test results are shown in Table A-4.

3.2. Experimental data on the performance of vertical shell-and-tube type exhaust-gas heat exchanger

As shown in Subsection 6.2, the shell-and-tube type heat exchanger was replaced by the shell-and-coil type from JARE-19 (1977/79) to the present in order to facilitate the scraping off of the soot adhering on the heating surface while the system was in operation. The prototype was ST-1, which was later developed into ST-2 and ST-3. In Tables A-5 and A-6, the test data on ST-2 and ST-3 are shown. Reference is made to Table 9 for the basic design data.

Table A-5a. Performance of a vertical shell-and-tube type exhaust-gas heat exchanger (ST-2) for JARE-20.

Run No.	Time (min)	Engine speed (rpm)	Load W (kg)	Output power L_e (PS)	G_a (kg/h)	G_f (kg/h)	G_g (kg/h)	b_e (gr/PSh)	t_{g1} (°C)	t_{g2} (°C)	Δt_g	Q_{R1}* (kcal/h)	ϕ	t_{w1} (°C)	t_{w2} (°C)	t_B (°C)	Remarks
1	0	1500	62	46.5	293	9.65	303	207	—	30	—	—	0.470	25.6	22.4	22.0	Water only.
2	5	1500	62	46.5	293	9.65	303	207	240	39	201	15400	0.470	27.5	22.7	22.0	Ice is thrown in.
3	10	1500	62	46.5	291	9.61	301	207	280	44.5	235.5	17900	0.469	27.5	23.1	20.0	Ice melting is
4	15	1500	62	46.5	291	9.68	301	208	300	49	251	19100	0.473	27.5	25.2	23.5	finished
5	20	1500	62	46.5	291	9.61	301	207	300	53	247	18800	0.469	27.8	28.9	27.0	τ_m=14.33 min.
6	25	1500	62	46.5	291	9.40	300	202	300	58	242	18400	0.459	30.6	33.6	32.0	
7	30	1500	62	46.5	291	9.47	301	204	300	61.5	238.5	18100	0.462	35.1	37.9	36.0	
8	35	1500	62	46.5	291	9.47	301	204	300	64.5	235.5	17900	0.462	39.1	41.8	40.0	
9	40	1500	62	46.5	291	9.47	301	204	300	69	231	17500	0.462	42.9	44.5	44.0	
10	45	1500	62	46.5	291	9.50	301	204	300	72.5	227.5	17300	0.466	47.4	48.2	47.0	
11	50	1500	62	46.5	291	9.47	301	204	300	75.5	224.5	17000	0.462	50.1	51.9	50.8	P_a=759.4 mm
12	55	1500	62	46.5	291	9.50	301	204	300	79	221	16800	0.466	54.2	55.8	54.5	Hg. abs.
13	60	1500	62	46.5	291	9.58	301	206	300	81	219	16700	0.467	57.9	58.6	57.0	t_{ad}=18°C
14	65	1500	62	46.5	291	9.50	301	204	300	83.5	216.5	16400	0.466	60.0	62.7	60.5	t_{aw}=17°C at start
15	70	1500	62	46.5	291	9.50	301	204	300	86	214	16200	0.466	64.2	65.8	63.8	t_{ad}=22.5°C
																	t_{aw}=21.0°C at final.
16	75	1500	62	46.5	293	9.50	303	204	300	89	211	16100	0.461	66.3	69.0	66.8	
17	80	1500	62	46.5	293	9.65	303	207	300	91	209	16000	0.468	69.2	71.8	69.5	
18	85	1500	62	46.5	293	9.83	303	211	310	94	216	16600	0.476	72.0	74.2	72.5	
19	90	1500	62	46.5	293	9.47	303	204	300	91	209	16000	0.459	74.8	76.2	75.5	
20	95	1500	62	46.5	293	9.58	303	204	310	94	216	16600	0.464	77.9	78.4	77.5	
21	100	1500	92	69.0	297	13.5	311	196	410	108	302	24400	0.645	79.7	83.0	80.7	Power is increased
22	105	1500	92	69.0	297	13.7	311	198	420	115	305	24700	0.654	83.9	86.6	85.0	

Note: 1) Specific weight of air γ_a=1.20 kg/m³.
2) Specific weight of fuel γ_f=840 kg/m³.
3) Water flow rate G_w=110.9 l/min=6654 kg/h at P=0.3 atg.
4) Specific of combustion gas c_{pg}=0.253 kcal/kg°C for Run No. 1–20, c_{pg}=0.260 kcal/kg°C for Run No. 21–22.
5) Quantity of water

in bathtub	150 (l)	
in heat exchanger	100 (l)	
in piping	7 (l)	
Sum		G_B=257 kg

6) Weight of ice block G_I=42.8 kg.