Table A-5b. Overall coefficint of heat transmission K_{lm}, recovery efficiency η_{rec}, and temperature efficiency Φ_g of ST-2.

Run No.	Time (min)	Q_{R1}^* (kcal/h)	t_{gm} (°C)	ΔQ_{R1} (kcal/h)	Q_{R1} (kcal/h)	Δt_{lm} (°C)	K_{lm} (kcal/m²h°C)	G_g (kg/h)	t_{g1} (°C)	$[c_{pgm}]_0^{t_{g1}}$ (kcal/kg°C)	Q_{g1} (kcal/h)	η_{rec}	$t_{g1}-t_{g2}$ (°C)	$t_{g1}-t_{w1}$ (°C)	Φ_g	$t_{gm}-t_{wm}$ (°C)	K_{am} (kcal/m²h°C)
1	0	—	—	—	—	—	—	303	—	—	—	—	—	—	—	—	—
2	5	15400	140	370	15030	70.0	33.2	303	240	0.250	18200	0.826	201	212.5	0.946	115	20.2
3	10	17900	163	441	17460	88.4	30.5	301	280	0.251	21200	0.824	235.5	252.5	0.933	138	19.6
4	15	19100	175	478	18620	99.4	29.0	301	300	0.252	22750	0.836	251	272.5	0.921	151	19.1
5	20	18800	177	485	18320	103.3	27.4	301	300	0.252	22750	0.805	247	272.2	0.907	149	19.0
6	25	18400	179	491	17910	105.1	26.3	300	300	0.252	22680	0.790	242	269.4	0.898	146	19.0
7	30	18100	181	497	17600	102.7	26.5	301	300	0.252	22750	0.774	238.5	264.9	0.900	144	18.9
8	35	17900	182	500	17400	100.3	26.9	301	300	0.252	22750	0.765	235.5	260.9	0.903	141	19.1
9	40	17500	185	509	16990	100.6	26.1	301	300	0.252	22750	0.747	231	257.1	0.898	141	18.6
10	45	17300	186	512	16790	98.3	26.4	301	300	0.252	22750	0.738	227.5	252.6	0.901	138	18.8
11	50	17000	188	518	16480	97.7	26.1	301	300	0.252	22750	0.724	224.5	249.9	0.898	137	18.6
12	55	16800	190	525	16280	95.9	26.2	301	300	0.252	22750	0.716	221	245.8	0.899	135	18.6
13	60	16700	191	528	16170	93.0	26.9	301	300	0.252	22750	0.711	219	242.1	0.905	133	18.8
14	65	16400	192	531	15870	92.4	26.4	301	300	0.252	22750	0.698	216.5	240.0	0.902	130	18.9
15	70	16200	193	534	15670	89.5	27.1	301	300	0.252	22750	0.689	214	235.8	0.908	128	18.9
16	75	16100	195	540	15560	89.8	26.7	303	300	0.252	22900	0.679	211	233.7	0.903	127	18.9
17	80	16000	196	543	15460	87.9	27.2	303	300	0.252	22900	0.675	209	230.8	0.906	125	19.1
18	85	16600	197	546	16050	90.1	27.5	303	310	0.252	23670	0.678	216	238.0	0.908	125	19.8
19	90	16000	196	543	15460	79.1	30.2	303	300	0.252	22900	0.675	209	225.2	0.928	123	19.4
20	95	16600	202	562	16040	78.4	35.2	303	310	0.252	23670	0.678	216	232.1	0.931	124	20.0
21	100	24400	259	738	24330	122.0	30.8	311	410	0.259	33030	0.737	302	330.3	0.914	177	21.2
22	105	24700	268	765	23940	127.4	29.1	311	420	0.259	33830	0.708	305	336.1	0.907	182	20.3
							28.2**					0.737**			0.910**		19.2**

Note:
1) $Q_{R1}^*=G_g c_{pgm}(t_{g1}-t_{g2})$ kcal/h.
2) $\Delta Q_{R1}=\sum \alpha_a A(t_{gm}-t_a)$ kcal/h (heat loss from gas-side surfaces) $=3.09(t_{gm}-t_a)=3.09(t_{gm}-20)$ kcal/h.
3) Corrected transferred heat $Q_{R1}=Q_{R1}^*-\Delta Q_{R1}$.
4) $\eta_{rec}=Q_{R1}/Q_{g1}$.
5) $Q_{g1}=G_g c_{pgm} t_{g1}$ kcal/h.
6) $Q_{R1}=K_{lm}F_i\Delta t_{lm}$.
7) $F_i=6.47$ m².
8) $K_{lm}=28.2$ kcal/m²h°C.
9) $\eta_{rec}=0.737$.
10) $\Phi_g=0.910$.
11) $K_{am}=19.2$ kcal/m²h°C.

**Mean value.

Table A-6. *Performance of a vertical shell-and-tube type exhaust-gas heat exchanger (ST-3) for JARE-21 (1979/81).*

Run No.	Time (min)	Engine speed N (rpm)	Load W (kg)	Output power L_e (PS)	G_a (kg/h)	G_f (kg/h)	b_e (gr/PSh)	t_{g1} (°C)	t_{g2} (°C)	Δt_g (°C)	G_g (kg/h)	$[c_{pgm}]_{t_{g2}}^{t_{g1}}$ (kcal/kg°C)	ϕ	Q_{R1}* (kcal/h)	ΔQ_{R1} (kcal/h)	Q_{R1} (kcal/h)	t_{w1} (°C)	t_{w2} (°C)	t_B (°C)
1	0	2034	59.5	60.5	428	13.7	227	310	63	247	442	0.254	0.455	27800	510	27300	31.2	35.8	30.8
2	5	2034	58.5	59.5	425	13.2	223	320	72	248	438	0.254	0.442	27700	530	27200	—	—	37.0
3	10	2030	58.0	58.9	421	13.1	223	323	79	244	434	0.254	0.442	27000	550	26450	42.8	47.0	42.7
4	15	2032	57.0	57.9	421	13.0	225	323	85	238	434	0.254	0.441	26300	560	25740	47.8	51.9	47.9
5	20	2034	56.5	57.5	418	12.9	224	323	92	231	431	0.254	0.437	25300	570	24730	52.8	56.9	53.2
6	25	2030	60.0	60.9	418	12.8	214	330	98	232	431	0.255	0.435	25600	590	25000	57.9	62.0	58.5
7	30	2030	59.5	60.4	418	13.2	219	330	102	228	431	0.255	0.455	25100	595	24500	62.6	66.7	63.4
8	35	2030	59.0	59.9	418	13.2	221	330	106	224	431	0.255	0.455	24700	600	24100	67.0	71.0	68.0
9	40	2024	59.0	59.7	418	13.0	218	330	111	219	431	0.255	0.444	24100	610	23500	71.8	75.6	72.5

Run No.	Time (min)	Δt_{lm} (°C)	K_{lm} (kcal/m²h°C)	t_{g1} (°C)	$[c_{gpm}]_0^{t_{g1}}$ (kcal/kg°C)	Q_{g1} (kcal/h)	η_{rec}	$t_{g1}-t_{w1}$ (°C)	Φ_g	t_{gm} (°C)	t_{wm} (°C)	$t_{gm}-t_{wm}$ (°C)	K_{am} (kcal/m²h°C)
1	0	112.5	40.3	310	0.252	34500	0.791	279	0.885	187	34	153	29.6
2	5	—	—	320	0.252	35300	0.771	—	—	196	—	—	—
3	10	118.1	37.2	323	0.252	35300	0.749	280	0.871	201	45	156	28.2
4	15	117.7	36.3	323	0.252	35300	0.729	275	0.865	204	50	154	27.8
5	20	118.5	34.7	323	0.252	35100	0.705	270	0.855	208	55	153	26.8
6	25	120.0	34.6	330	0.252	35800	0.698	272	0.853	214	60	154	27.0
7	30	117.9	34.5	330	0.252	35800	0.684	267	0.854	216	65	151	27.0
8	35	116.2	34.5	330	0.252	35800	0.673	263	0.852	218	69	149	26.9
9	40	115.1	33.9	330	0.252	35800	0.656	258	0.849	221	74	147	26.6
			35.1**				0.715**		0.860**				26.8**

Note:
1) Tested at Nihon University in Tokyo (6 October 1979).
2) Diesel engine: Isuzu DA-640.
3) Flow rate of water G_w=3600 kg/h.
4) Water contained in the bathtub, heat exchanger and pipes G_B=286.5 kg.
5) Heating surface area F_i=6.02 m² (inner surface area of tubes).
6) Atmospheric pressure and temperatures
start P_z=759.4 mmHg t_{ad}=19.5°C t_{aw}=17.1°C,
final P_z=758.9 mmHg t_{ad}=19.4°C t_{aw}=17.2°C.
7) $\Delta Q_{R1}=3.0\,(t_{gm}-t_a)$ kcal/h. Correction for heat loss.
8) In calculating mean values, No. 1 and No. 2 are omitted.
** Mean value.

Table A-7. Performance of a vertical fin-tube type exhaust-gas heat exchanger (AL-F-2) for JARE-21 (1979/81).

Run No.	Time (min)	Engine speed N (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)	Equivalence ratio ϕ	t_{g1} (°C)	t_{g2} (°C)	Δt_g (°C)	t_{gm} (°C)	$[c_{pgm}]_{t_{g1}}^{t_{g2}}$ (kcal/kg°C)	Q_{R1}* (kcal/h)	Q_{R1} (kcal/h)	t_{w1} (°C)	t_{w2} (°C)
1	0	2028	60.5	61.3	410	13.9	424	226	0.478	305	80	225	193	0.254	24300	23240	31.7	33.8
2	5	2030	60.5	61.4	414	13.7	428	223	0.469	335	123	212	229	0.256	23250	22190	36.8	39.3
3	10	2028	60.0	60.8	410	13.4	423	221	0.465	338	138	200	238	0.257	21800	20740	41.4	44.6
4	15	2028	59.5	60.3	410	13.2	423	220	0.459	338	143	195	241	0.257	21300	20240	45.9	49.2
5	20	2030	59.5	60.4	407	13.2	420	219	0.463	335	148	187	242	0.257	20240	19180	50.9	54.2
6	25	2028	59.0	59.8	407	13.2	420	222	0.463	335	151	184	243	0.257	19900	18840	55.6	58.4
7	30	2028	58.5	50.3	407	13.0	420	219	0.455	335	155	180	245	0.257	19500	18440	59.8	62.5
8	35	2026	58.0	58.8	407	13.0	420	222	0.457	330	157	173	244	0.257	18700	17640	64.3	67.7
9	40	2028	57.5	58.3	407	12.9	420	222	0.450	330	160	170	245	0.257	18400	17340	68.4	71.3

Run No.	Time (min)	Δt_w (°C)	t_{wm} (°C)	Δt_{lm} (°C)	K_{lm} (kcal/m²h°C)	$[c_{pgm}]_0^{t_{g1}}$ (kcal/kg°C)	Q_{g1} (kcal/h)	η_{rec}	$t_{g1}-t_{w1}$ (°C)	Φ_g	$t_{gm}-t_{wm}$ (°C)	K_{am} (kcal/m²h°C)	Water temp. in bathtub t_B (°C)
1	0	2.1	33	129	30.2	0.252	32600	0.713	273	0.824	160	24.3	31.6
2	5	1.5	38	170	21.9	0.253	36280	0.612	298	0.711	191	19.5	36.7
3	10	3.2	43	177	19.6	0.253	36170	0.573	297	0.673	195	17.8	41.0
4	15	3.3	48	176	19.3	0.253	36170	0.560	292	0.668	193	17.6	45.8
5	20	3.3	53	173	18.6	0.253	35600	0.539	284	0.658	189	17.0	50.4
6	25	2.9	57	170	18.6	0.253	35600	0.529	279	0.659	186	17.0	55.0
7	30	2.7	61	169	18.3	0.252	35460	0.520	275	0.655	184	16.8	60.0
8	35	3.4	66	163	18.1	0.252	34930	0.505	266	0.650	178	16.6	64.2
9	40	2.9	70	161	18.0	0.252	34930	0.496	262	0.649	175	16.6	68.2
					18.6**			0.531**				16.9**	

Note: 1) Tested at Nihon University in Tokyo (6 October 1979).
2) Diesel engine: Isuzu DA-640.
3) Flow rate of water G_w=3600 kg/h.
4) Water contained

in a bathtub	200	(l)
in the heat exchanger	50	(l)
in pipe	8.5	(l)
Total	258.5 kg	

5) Heating surface area F_o=5.971 m² (gas side).
6) ΔQ_{R1}=1060 kcal/h (correction for heat loss).
7) P_a=758.9 mmHg, t_{ad}=19.4°C, t_{aw}=17.2°C.
** Mean value. In calculating mean values, No. 1 and No. 2 are omitted.

3.3. Experimental data on the performance of vertical fin-tube type exhaust-gas heat exchanger

Three kinds of fin-tube type exhaust-gas heat exchangers were developed by the authors as described in Subsection 6.3 and in Table 10. In Table A-7, the test results of AL-F-2 are shown. This type of heat exchanger is now undergoing development. AL-F-3 has a larger heat capacity than AL-F-1 and AL-F-2 because of its large coefficient of heat transfer on the gas side obtained by increasing gas velocity to 2.26 times that of AL-F-1.

APPENDIX 4

4.1. An example of theoretical analysis of a vertical shell-and-tube type water-to-water heat exchanger

As described in Subsection 9.3, a vertical shell-and-tube type water-to-water heat exchanger was developed by the authors for JARE-13 and JARE-20. Its features are shown in Fig. 55, and the design data are as follows:

Inner diameter of shell	$D_s=0.155$ m,	
outer diameter of tubes	$d_o=0.018$ m,	
inner diameter of tubes	$d_i=0.014$ m,	
number of tubes	$z=26$,	
length of a tube	$l=0.988$ m,	
number of baffle plates	$N=5$,	
height of baffle cut	$H=0.0375$ m	$H/D_s=0.242$,
triangular pitch	$P_t=0.023$ m,	
distance between two baffle plates	$B=0.160$ m.	

4.1.1. Hot-water flow inside of the tube

The hot-water is recirculated by a centrifugal pump.

Hot-water pump manufactured by Kawamoto Seisakusho CS-32-CO4T
flow rate 46 l/min with a head of 8 m,
17 l/min with a head of 25 m,
inlet pipe diameter $1\cdot 1/2''$ was reduced to $1''$.

Estimated flow rate of water	$V_i=1.212\ \mathrm{m^3/h}=0.000337\ \mathrm{m^3/s}$,
heating surface area	$F_i=\pi d_i l z=\pi(0.014)(0.988)(26)=1.130\ \mathrm{m^2}$,
sectional area	$A_i=(\pi/4)d_i^2\,(z/2)=(\pi/4)(0.014)^2(13)=0.00200\ \mathrm{m^2}$,
velocity	$w_i=V_i/A_i=0.000337/0.00200=0.1685\ \mathrm{m/s}$,
kinematic viscosity	$\nu_i=0.417\times 10^{-6}\ \mathrm{m^2/s}\ (70°\mathrm{C})$,
Prandtl number	$(Pr)_i=2.69$,
thermal conductivity	$\lambda_i=0.571$ kcal/mh°C,
Reynolds number	$(R_e)_i=w_i d_i/\nu_i=(0.1685\times 0.014)/(0.417\times 10^{-6})=5657$,

Nusselt number

$$(Nu)_i = 0.023(R_e)_i^{0.8}(Pr)_i^{0.4} = 0.023(5657)^{0.8}(2.69)^{0.4} = 34.3,$$

heat transfer coefficient

$$\alpha_i = \lambda_i(Nu)_i/d_i = 0.571 \times 34.3/0.014 = 1400 \text{ kcal/m}^2\text{h}^\circ\text{C}.$$

4.1.2. Cold-water flow across tube bank

The cold water of the outdoor 130-k*l* water tank was recirculated by a water pump submerged in the tank.

Manufactured by Sakuragawa Seisakusho V-40F,
inlet diameter 50 mm, motor 400 W,
flow rate 200 *l*/min, 12000 kg/h,
$V_o = 0.00333$ m^3/s with a head of 6 m,
total outer heating surface area

$$F_o = \pi d_o l\, z = \pi(0.018)(0.988)(26) = 1.453 \text{ m}^2,$$

triangular pitch $P_t = 0.023$ m,
equivalent diameter

$$D_e = [2\sqrt{3}\, P_t^2/(\pi d_o)] - d_o$$
$$= [2\sqrt{3}\,(0.023)^2/(\pi \times 0.018)] - 0.018 = 0.0144 \text{ m},$$

minimum flow area

$$a_s = (D_s/P_t)(P_t - d_o)B = (0.155/0.023)(0.023 - 0.018)0.160 = 0.00539 \text{ m}^2,$$

velocity $w_o = V_o/a_s = 0.00333/0.00539 = 0.618$ m/s,
kinematic viscosity $\nu_o = 1.25 \times 10^{-6}$ m^2/s (12°C),
thermal conductivity $\lambda_o = 0.508$ kcal/mh°C,
Prandtl number $(Pr)_o = 8.91$,
Reynolds number

$$(R_e)_o = w_o D_e/\nu_o = (0.618 \times 0.0144)/(1.25 \times 10^{-6}) = 7120,$$

Nusselt number

$$(Nu)_o = 0.224(R_e)_o^{0.537}(H/D_s)^{-0.4}(Pr)_o^{1/3}$$
$$= 0.244(7120)^{0.537}(0.242)^{-0.4}(8.91)^{1/3} = 95.8,$$

heat transfer coefficient

$$\alpha_o = \lambda_o(Nu)_o/D_e = 0.508 \times 95.8/0.0144 = 3380 \text{ kcal/m}^2\text{h}^\circ\text{C},$$

thickness of tube wall $\delta_m = 0.002$ m,
thermal conductivity of SUS-28 $\lambda_m = 14$ kcal/mh°C,
total heat resistance

$$\sum R = (1/\alpha_i)(d_o/d_i) + (1/\alpha_o) + (\delta_m/\lambda_m)$$
$$= (1/1400)(0.018/0.014) + (1/3380) + (0.002/14)$$
$$= 0.000919 + 0.000296 + 0.000143 = 0.001358 \text{ m}^2\text{h}^\circ\text{C/kcal},$$

overall coefficient of heat transmission

$$K_o = 1/\sum R = 1/0.001358 = 736 \text{ kcal/m}^2\text{h}^\circ\text{C},$$
$$K_i = K_o(F_o/F_i) = 736(0.018/0.014) = 947 \text{ kcal/m}^2\text{h}^\circ\text{C},$$

$$W_i = c_i G_i = (1)(1212) = 1212 \text{ kcal/h}^\circ\text{C},$$
$$W_o = c_o G_o = (1)(12000) = 12000 \text{ kcal/h}^\circ\text{C},$$
$$W_i/W_o = 1212/12000 = 0.101,$$
$$K_i F_i / W_i = (947 \times 1.130)/(1212) = 0.883.$$

The temperature efficiency Φ_i is given by BOŠNJAKOVIĆ *et al.* (1951) in a graph as the function of $K_i F_i / W_i$ and W_i/W_o. By the aid of this graph, we can estimate as $\Phi_i = 0.55$, and from eqs. (71), we get $t_{i2} = 40^\circ\text{C}$ by putting $t_{i1} = 75^\circ\text{C}$. From eq. (73),

$$W_i \Phi_i = W_o \Phi_o, \qquad \Phi_o = \Phi_i (W_i/W_o) = 0.55(0.101) = 0.0556$$

and from eq. (72), we can estimate the outlet temperature of the cold water as follows:

$$t_{o2} = 15.5^\circ\text{C} \text{ for } t_{o1} = 12^\circ\text{C}.$$

The logarithmic mean temperature difference is calculated as $\Delta t_{lm} = 42^\circ\text{C}$, and the heat recovered is given by

$$Q_c = K_o F_o \, \Delta t_{lm} = (736)(1.453)(42) = 45000 \text{ kcal/h}.$$

APPENDIX 5

5.1. Ice-melting experiment in Tokyo utilizing coolant energy of 12-kVA diesel-electric generator to be installed at Mizuho Station

The authors designed and manufactured a waste heat recovery system for utilizing the exhaust gas and coolant energy of a 12-kVA diesel-electric generator for JARE-12 (1970/72), and it was tested at Nakajima Denki Co., Ltd. in Tokyo on 11 November 1970. The test results are given in Table A-8. The first plan for Mizuho Station was as follows:

12-kVA diesel-electric generator set in a caboose: ZX 500B type of Meidensha Co., Ltd.

Diesel engine: Isuzu C221 (total piston displacement volume 2207 cc, 21 PS/1500 rpm).

Electric generator: E-AF type, 12-kVA, 3-phase, AC 200 V, 34.6 A, 50 Hz, power factor 0.8, 9.6 kW/1500 rpm.

Waste heat recovery systems

(1) Exhaust-gas energy recovery system for ice-melting

A shell-and-coil type heat exchanger, ice-melting tank and tube type water-to-water heat exchanger and a recirculating pump (flow rate $G_w = 41.5\ l/\text{min}$).

Table A-8. Results of experiments on the ice-melting system by utilizing the exhaust-gas heat exchanger (shell-and-coil type) and on the hot-water room-heating system by the coolant energy of a 12-kVA diesel-electric generator prepared for Mizuho Station in JARE-12 (1970/72).

Run No.	Time (h min)		V	A	kW	t_{g1} (°C)	t_{g2} (°C)	Δt_g (°C)	Q_{R1} (kcal/h)	t_{w1} (°C)	t_{w2} (°C)	Δt_w (°C)	Q_{R2} (kcal/h)	t_B (°C)	t_H (°C)	t_{H1} (°C)	t_{H2} (°C)	Δt_H (°C)	Q_H (kcal/h)
/	10 42	start	200							14.2	14.0			16.7					
1	10 47	1/4 load	200	6.9	2.21	—	24.3	—	—	19.7	20.7	1.0	2500	17.8					
2	10 50		200	6.9	2.21	154.5	26.6	127.9	2600	22.2	23.2	1.0	2500	19.1	41.2	38.5	33.8	5.7	15400
3	11 05		200	6.9	2.21	154.5	29.9	124.6	2500	25.1	26.0	0.9	2300	20.6	47.0	45.0	39.8	5.2	14000
4	11 15		200	6.9	2.21	154.5	31.4	123.1	2460	27.6	28.5	0.9	2300	23.2	51.0	49.5	43.8	4.3	11600
/	11 17	1/2 load	200	13.6	4.35	—	—	—	—	—	—	—	—	—	—	—	—	—	—
5	11 22		200	13.6	4.35	187.5	34.3	153.2	3130	29.8	30.9	1.1	2800	25.6	54.2	52.9	46.5	6.4	17300
6	11 32		200	13.6	4.35	187.5	37.3	150.2	3050	32.8	33.9	1.1	2800	28.7	56.8	55.2	48.8	6.4	17300
/	11 34	3/4 load	200	20.4	6.53	—	—	—	—	—	—	—	—	—	—	—	—	—	—
7	11 44		200	20.4	6.53	208.0	43.2	164.8	3320	38.2	39.4	1.2	3000	33.9	61.0	60.5	52.8	7.7	20800
8	11 50		200	20.4	6.53	208.0	44.4	163.6	3280	39.4	40.7	1.3	3200	34.5	62.5	60.6	53.5	7.1	19200
/	11 51	4/4 load	200	30.4	9.73	—	—	—	—	—	—	—	—	—	—	—	—	—	—
9	11 56		200	30.4	9.73	245.5	47.4	198.1	4040	42.5	43.9	1.4	3500	37.5	65.0	63.1	55.4	7.7	20800
10	12 01		200	30.4	9.73	245.5	49.3	196.2	3980	44.0	45.5	1.5	3700	38.5	64.2	64.5	56.4	8.1	21900
/	12 14	stop																	

Note: 1) Tested at Nakajima Denki Co., Ltd. on 12 November 1970 by the authors. Room temperature t_a=14.6°C (initial), 16.9°C (final).

2) 12-kVA diesel-electric generator to be installed in Mizuho Station by JARE-12.

3) The flow rate of a water pump recirculating the water between the exhaust-gas heat exchanger and a headtank and an ice-melting tank was 41.5 *l*/min (2490 kg/h). The total weight of water to be heated was
G_B=54 kg (in the coil and headtank)+90 kg (in the ice-melting tank)=144 kg.

4) The coolant was bypassed to a headtank and the hot water in the headtank was fed to three fan-coil units connected in parallel by another water pump having a flow rate of 45 *l*/min (2700 kg/h). The total weight of the water contained in the fan-coil and recirculating system was 47.8 *l*.

5) The air flow of the diesel engine was G_a=0.0281 kg/s=101 kg/h. Recovered exhaust-gas energy by the shell-and-coil type heat exchanger can be represented by Q_{R1} (calculated from gas side) or Q_{R2} (calculated from water side).

6) The temperatures t_{w1} and t_{w2} show the inlet and outlet water temperatures of the exhaust-gas heat exchanger.

7) t_B: Water temperature in the ice-melting tank. t_H: Coolant temperature in a headtank for coolant system.
t_{H1}: Feed coolant temperature to fan-coil units. t_{H2}: Return coolant temperature from fan-coil units.
Q_H: Total heat radiated from three fan-coil units as warm air.

(2) Coolant-heat recovery system for making hot water for room-heating.

A headtank for engine coolant.

A bathtub and heating radiator.

A recirculating pump for coolant (flow rate $G_{wH}=41.5\ l/\mathrm{min}$).

Three fan-coil units (1 phase, 100 V, 40 W, 4350 kcal/h).

The experimental results shown in Table A-8 are summarized as follows:

(1) Exhaust-gas energy recovery system

The recovered exhaust-gas energy was

$Q_{R1}=2500$ kcal/h at 2.2 kW and 4000 kcal/h at 9.73 kW.

(2) Coolant heat recovery system

Three fan-coil units were heated by using the coolant heat of the diesel engine. The heating capacity of one unit was 4350 kcal/h, so that the total available heat was about 13000 kcal/h. The coolant heat was abundant compared with the exhaust-gas energy available and was more easily utilized. At Mizuho Station, only the coolant system was rebuilt during JARE-12 for making cold and hot water, and the room-heating system was put into operation during JARE-17 (1975/77) and has been used successfully thereafter.

5.2. Experimental data on a system to melt ice and produce hot water at Mizuho Station in JARE-12

During JARE-12 (1970/72), only the coolant heat recovery system was rebuilt and actually used. The data observed by Mr. TAGA is shown in Table A-9. The experiments were continued from 3:00 of July 12 to 10:30 of July 13, 1971 for 31.5 h.

The system is shown in Fig. A-4. The coolant was bypassed to a headtank and returned to the coolant pump of the engine. The hot water stored in the headtank was recirculated to an ice-melting tank and to a bathtub by a recirculating pump and heated these vessels indirectly through a heating tube and a radiator submerged in each of them.

Six kinds of experiments were continued as shown in Table A-10. In series A, B, C and D, 100 kg of ice, the temperature of which was $-22°$C, was thrown into the ice-melting tank containing 100 kg of hot water. The water temperature dropped steeply until the ice block was melted completely, and then it was raised again gradually with time.

The following relations should be held for the heating of water in the ice-melting tank:

Q_c: transferred heat from coolant to water, kcal/h,

τ: time, h,

t_{cm}^*: an ultimate coolant temperature at $\tau=\infty$, °C,

Table A-9. Results of tests on the coolant heat recovery system of 12-kVA in Mizuho Station (observed by TAGA *in 12–13 July 1971, JARE-12).*

Run No.	Time	Load (kW)	G_{wR} (*l*/min)	G_{wH} (*l*/min)	Ice-melting tank	B_o (*l*/h)	kW	Atm. temp. t_a(°C)	Trench temp. (°C)	Caboose temp. (°C)	Coolant outlet temp. (°C)	Temp. return to head-tank (°C)	Temp. in ice-melting tank (°C)	
A 1	12 3 00	1.5	6	6	100 kg of ice (−22°C)	1.85	1.4	−31	−4	38	91	89	87	
2	4 00	(15.6%)	(50%)	(50%)	was put into the		1.5	−31	−3	36	93	91	89	
3	4 15				ice-melting tank		1.5	−30	−3	35	72	55	28	4.10 snow-in
4	4 20				containing 100 *l*		1.5	−31	−3	34	60	48	30	
5	5 15				of hot water		1.5	−32	−6	24	62	56	48	
6	5 20						1.5	−33	−6	24	63	57	50	
7	9 15						1.5	−33	−4	34	94	89	87	
B 8	10 50	1.6	0	12	Same as No.1	1.88	1.5	−35	−0	34	98	94	92	10.50 snow-in
9	12 01	(16.7%)	(0%)	(100%)			1.6	−36	−9	14	72	67	60	
10	13 00						1.6	−37	−14	12	87	83	77	
11	13 50						1.7	−38	−8	14	96	92	87	
C 12	14 05	3.2	0	12	Same as No. 1	2.25	3.2	−40	−14	12	64	54	23	13.56 snow-in
13	15 05	(33.4%)	(0%)	(100%)			3.2	−41	−13	9	66	60	50	
14	16 05						3.2	−43	−14	12	86	82	75	
15	16 30						3.4	−43	−13	14	94	89	84	
D 16	16 40	5.3	0	12	Same as No. 1	2.45	5.2	−43	−15	12	95	90	86	
17	16 55	(55%)	(0%)	(100%)			5.3	−43	−17	11	74	62	28	16.50 snow-in
18	18 25						5.3	−43	−12	13	90	85	78	
E 19	18 30	5.2	12	0	Without ice melting	2.34	5.2	−43	−14	13	92	88	80	
20	19 25	(54%)	(100%)	(0%)			5.2	−43	−7	43	79	78	80	
21	19 30						5.2	−43	−7	46	78	77	80	
22	20 33						5.2	−43	−7	39	78	72	78	
F 23	21 30	1.9	9.6	2.4	Without ice melting	1.94	2.4	−43	−8	39	72	70	73	
24	22 30	(20%)	(80%)	(20%)			2.0	−44	−7	40	72	69	70	
25	23 30						2.0	−43	−6	41	72	69	70	
26	13 1 20						2.0	−43	−6	41	73	68	69	9.00 recording was stopped
27	9 15						2.0							
28	10 30						0.2							

Note: 1) Coolant flow rate 18 *l*/min=1080 kg/h.
2) G_{wR}: Coolant flow rate to the engine radiator *l*/min. G_{wH}: Coolant flow rate to the headtank *l*/min.
3) B_o: Fuel flow rate *l*/h.

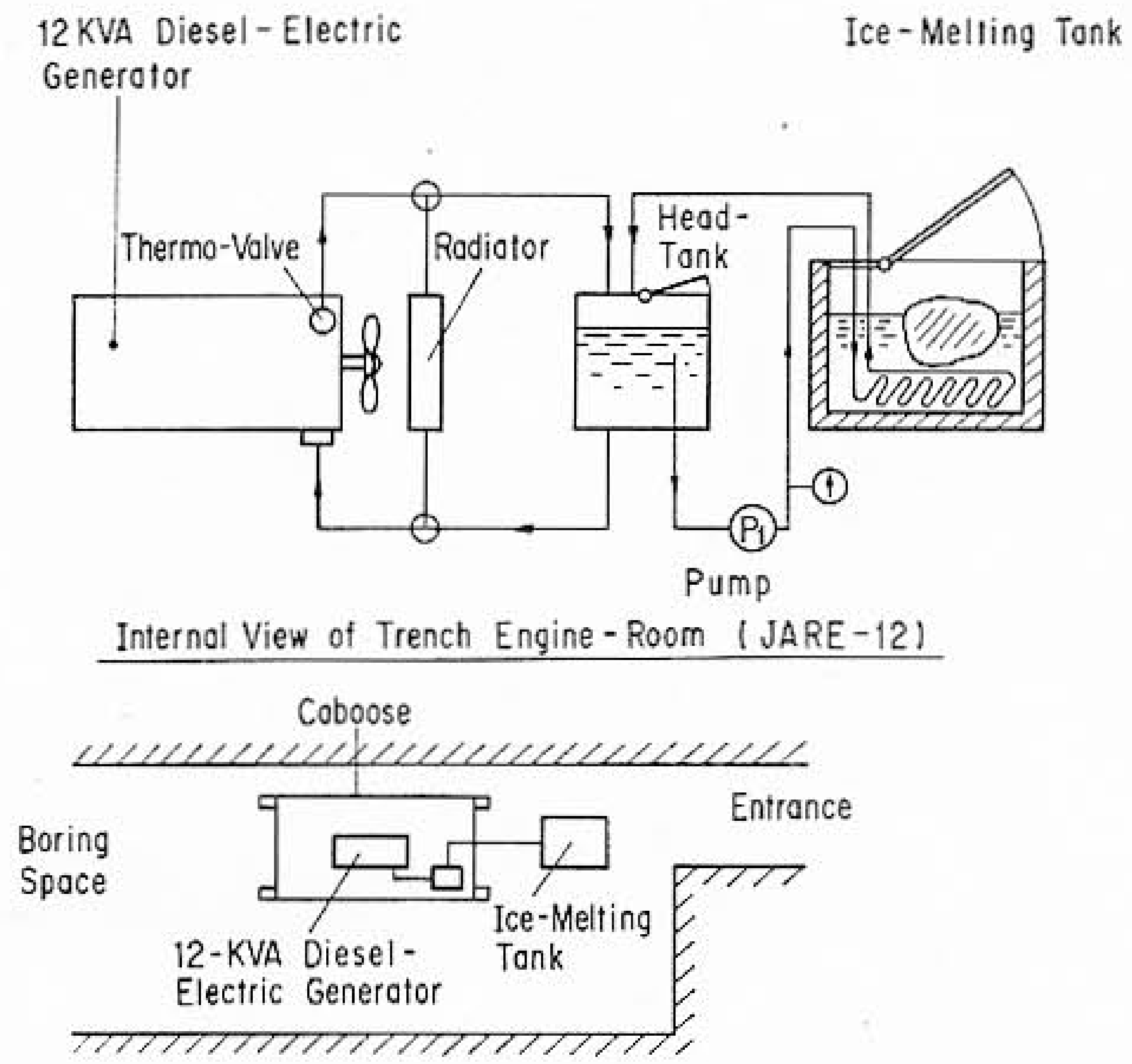

Fig. A-4. System to heat water and melt ice or snow with coolant heat in a trench engine room of Mizuho Station in JARE-12 (1970/72).

t: an instantaneous water temperature in the ice-melting tank, °C,
c_w: specific heat of water, kcal/kg°C,
G_w: total weight of water to be heated, kg,
C_M: specific heat of tank materials, etc., kcal/kg°C,
G_M: weight of tank materials, etc., kg,
α_a: coefficient of heat transfer from hot surfaces to atmosphere, kcal/m²h°C,
A: surface area relating to heat loss of tank, etc., m^2,
t_a: atmospheric air temperature, °C,
K_c: overall coefficient of heat transmission of heating coil, referring to F_c and $(t_{cm}{}^* - t)$, kcal/m²h°C,
F_c: heating area, m².

The equation of heat balance can be written as follows, similarly as for Section 7(c):

$$Q_c \mathrm{d}\tau = K_c F_c (t_{cm}{}^* - t)\mathrm{d}\tau = (c_w G_w + \sum C_M G_M)\mathrm{d}t + \sum \alpha_a A(t - t_a)\mathrm{d}\tau \qquad \text{(A-15)}$$

which can be reduced to

$$\frac{\mathrm{d}t}{\mathrm{d}\tau} = a - bt \ \ °\mathrm{C/h}, \qquad \text{(A-16)}$$

Table A-10. Test results of ice-melting system using waste coolant heat of 12-kVA diesel-electric generator in Mizuho Station in JARE-12 (observed by TAGA *in 12–13 July 1971).*

Experiment series	A	B	C	D	E	F	Remarks
Engine coolant to radiater G_{wR} (l/min)	6 (50%)	0 (0%)	0 (0%)	0 (0%)	12 (100%)	9.6 (80%)	Refer Fig. A-4 and Table A-9
Engine coolant to headtank G_{wH} (l/min)	6 (50%)	12 (100%)	12 (100%)	12%) (100%)	0%) (0%)	2.4 (20%)	
Load (kW)	1.5 (15.6%)	1.6 (16.7%)	3.2 (33.3%)	5.3 (55.2%)	5.2 (54.2%)	1.9 (19.8%)	Max. load 9.6 kW (100%)
Fuel consumption							
B_o (1/h)	1.85	1.88	2.25	2.45	2.34	1.94	Specific density
B (kg/h)	1.52	1.54	1.85	2.01	1.92	1.59	$\rho_f=0.82$
Total heat supplied by fuel Q_f (kcal/h)	16500	16700	20000	21800	20800	17200	$H_o=10832$ kcal/kg
Output as electric power (kcal/h)	1290 (7.8%)	1380 (8.3%)	2750 (13.8%)	4560 (20.9%)	4470 (21.5%)	1630 (9.5%)	At $t=70$°C
Ice-melting power (kcal/h)	1920 (11.6%)	3320 (19.9%)	4580 (22.9%)	4510 (20.7%)			
Efficiency (total)	(19.4%)	(28.2%)	(36.7%)	(41.6%)	(21.5%)	(9.5%)	
Coefficient b (1/h)	0.283	0.408	0.385	0.458			
Total weight of water in ice-melting tank (kg)	200	200	200	200			Hot water 100 Ice 100
K_cF_c (kcal/h°C)	56.6	81.6	77	91.6			
$t_{cm}{}^*=b/a$ (°C)	104	110.7	129.5	119.2			

where

$$a=(K_cF_ct_{cm}{}^*+\sum\alpha_aAt_a)/(c_wG_w+\sum c_MG_M) \quad {}^\circ\text{C/h}, \tag{A-17}$$

$$b=(K_cF_c+\sum\alpha_aA)/(c_wG_w+\sum c_MG_M) \quad 1/\text{h}, \tag{A-18}$$

$$a/b=(K_cF_ct_{cm}{}^*+\sum\alpha_aAt_a)/(K_cF_c+\sum\alpha_aA)\doteqdot t_{cm}{}^*. \tag{A-19}$$

From eq. (A-18),

$$K_cF_c=b(c_wG_w+\sum c_MG_M)-\sum\alpha_aA\doteqdot b(c_wG_w) \quad \text{kcal/h}^\circ\text{C}. \tag{A-20}$$

The solution of eq. (A-16) is given as follows:

$$t=(a/b)-[(a/b)-t_o]e^{-b\tau}\doteqdot t_{cm}{}^*-[t_{cm}{}^*-t_o]e^{-b\tau} \quad {}^\circ\text{C}, \tag{A-21}$$

and

$$Q_c=K_cF_c(t_{cm}{}^*-t) \quad \text{kcal/h}. \tag{A-22}$$

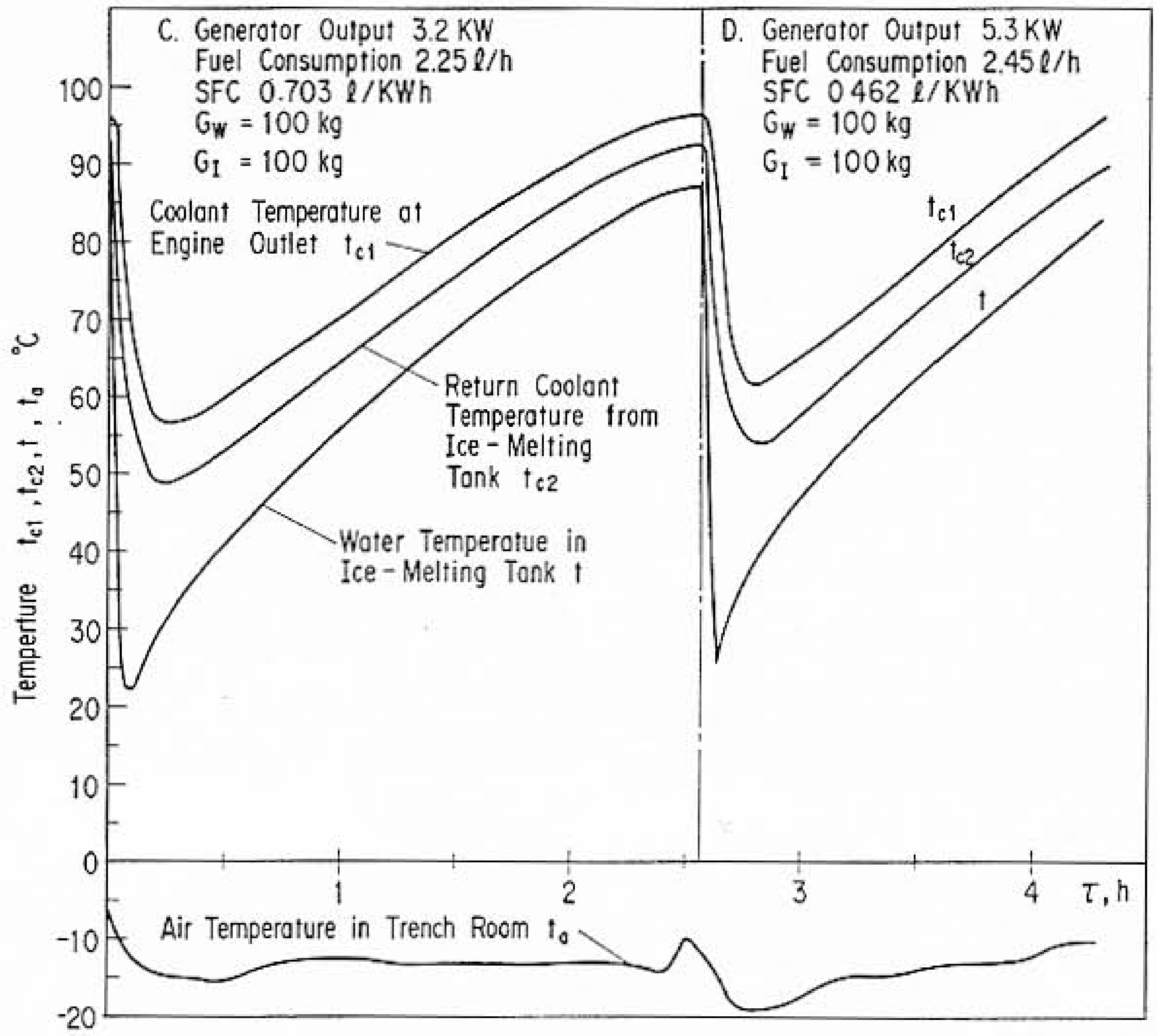

Fig. A-5. Cooling and heating curves of water temperature of ice-melting system using coolant heat of 12-kVA diesel-electric generator (measured by TAGA, 12–13 July 1971).

In eq. (A-21), time τ should be calculated from a point on a temperature increasing curve, where the water temperature t is equal to t_o.

For series A

$$t=104-57.4\,e^{-0.283\tau},$$

$$b=0.283\ 1/\text{h},\quad a/b=104^\circ\text{C},\qquad a=29.4^\circ\text{C/h},$$

$$G_w=200\ \text{kg},\quad c_w=1.0\ \text{kcal/kg}^\circ\text{C},\quad t_{cm}{}^*=104^\circ\text{C}.$$

From eq. (A-20), we get

$$K_cF_c=b(c_wG_w)=0.283(200)=56.6\ \text{kcal/h}^\circ\text{C},$$

$$Q_c=K_cF_c(t_{cm}{}^*-t)=56.6(104-t)=5886-56.6t\ \text{kcal/h},$$

$$Q_c=1924\ \text{kcal/h}(t=70^\circ\text{C}).$$

For series B

$$t=110.7-54.7\,e^{-0.408\tau},\quad b=0.408\ 1/\text{h},\quad a=45.2^\circ\text{C/h},$$

$$Q_c=81.6(110.7-t)=9033-81.6t\ \text{kcal/h},$$

$$Q_c=3321\ \text{kcal/h}\ \ (t=70^\circ\text{C}).$$

For series C

$$t = 129.5 - 82.5\, e^{-0.385\tau}, \quad b = 0.385\ 1/\mathrm{h}, \quad a = 49.9^\circ\mathrm{C/h},$$
$$Q_c = 77(129.5 - t) = 9972 - 77\, t\ \ \mathrm{kcal/h},$$
$$Q_c = 4582\ \mathrm{kcal/h}\ \ (t = 70^\circ\mathrm{C}).$$

For series D

$$t = 119.2 - 75.7\, e^{-0.458\tau}, \quad b = 0.458\ 1/\mathrm{h}, \quad a = 54.6^\circ\mathrm{C/h},$$
$$Q_c = 91.6(119.2 - t) = 10920 - 91.6t\ \ \mathrm{kcal/h},$$
$$Q_c = 4510\ \mathrm{kcal/h}\ \ (t = 70^\circ\mathrm{C}).$$

In Fig. A-5, the cooling and heating curves for series C and D are shown. The summary of the experimental results are shown in Table A-10.

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No. 3. AWANO, S. *et al.*: Axial flow wind air-turbine NU-102 with electric eddy-current brake. October 1979