and recirculated to the headtank.

The hot water reserved in the hot-water tank was circulated to eight fan-coil units through a stainless-steel pipeline and warmed six rooms of the living quarters. Another 65-kVA diesel-electric generator set was installed in the same engine room by JARE-10 (1968/70) as shown in Fig. 14 (ISHIWATA *et al.*, 1970).

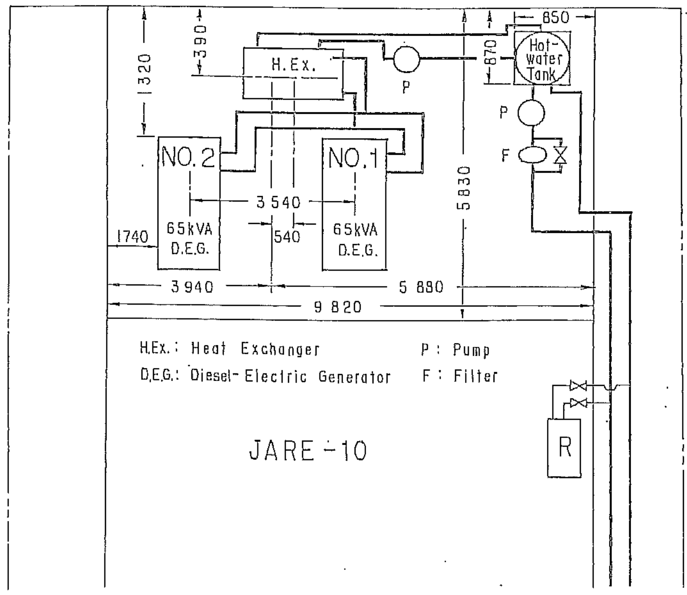

Fig. 14. JARE-9 engine room in JARE-10 (1968/70) containing two 65-kVA diesel-electric generators and rearranged coolant heat recovery system. Direct heating system by exhaust gas for JARE-9 was changed to indirect heating system through a horizontal shell-and-tube type water-to-water heat exchanger.

By this hot-water room-heating system utilizing the coolant heat of the 65-kVA diesel-electric generators, the heating load of the living quarters of the JARE-9 engine room, which was estimated as about 14000 kcal/h excluding the engine room itself, has been supplied satisfactorily from 1968 to 1977. The heating load of the engine room, which was also about 11300 kcal/h could be balanced by the heating loss due to the engine mechanical friction and the mechanical and electrical losses of the generators.

During JARE-10 (1968/70), the coolant heat recovering system was changed from the system of direct heating by engine coolant to an indirect heating system as shown in Fig. 14. A 130-k*l* cylindrical water-storage outdoor tank made of corrugated steel plates and canvas sheets similar to the 10-k*l* tank was built by JARE-11 (1969/71) (Fig. 15) near the JARE-9 engine room.

Shell-and-coil type exhaust-gas heat exchangers were installed in the same engine room as shown in Figs. 16 and 17, and the recovered heat was used for

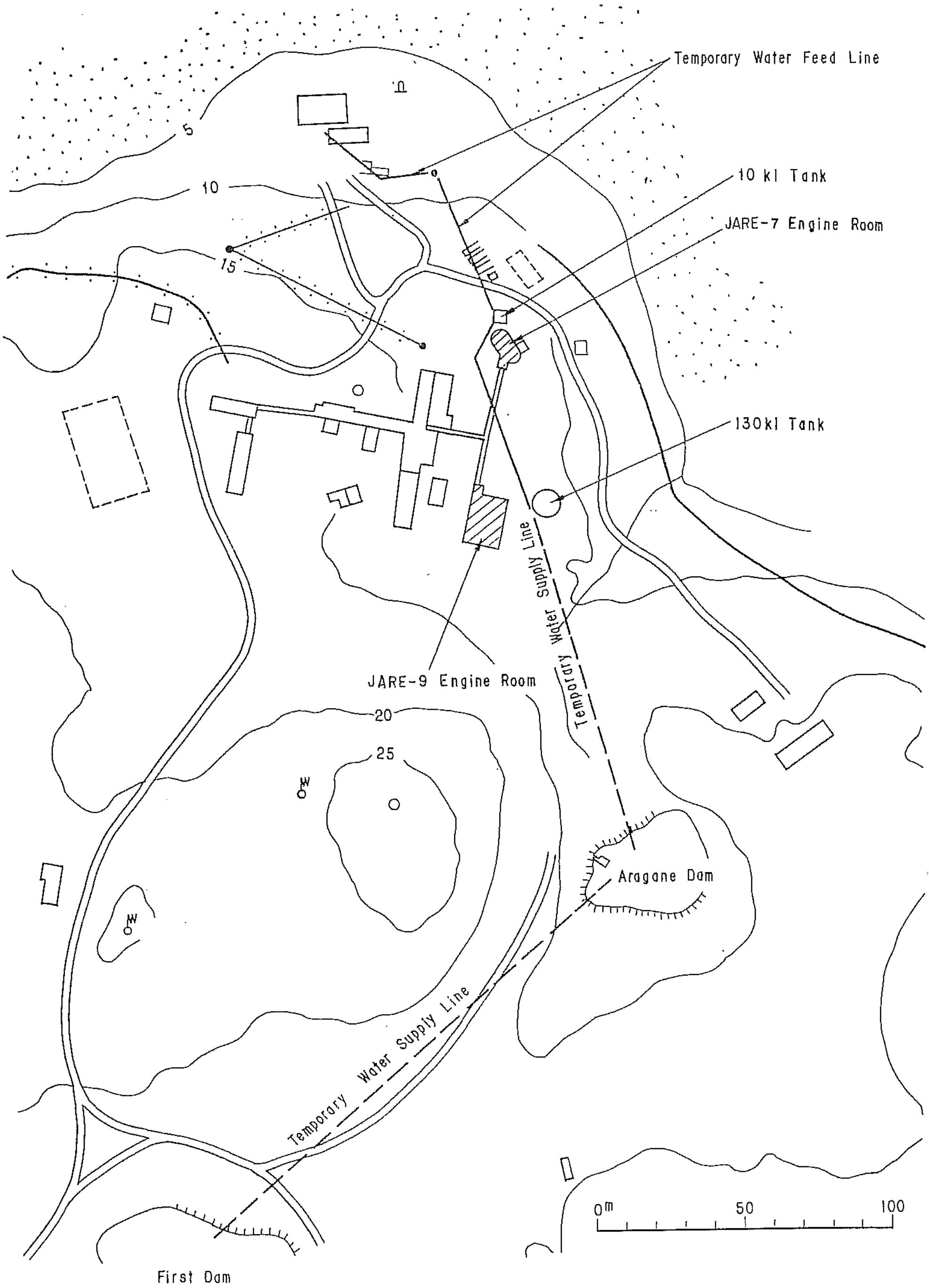

Fig. 15. Water supplying pipelines at Syowa Station during JARE-11 (1969/71).

heating the outdoor 130-k*l* tank through an intermediate vertical water-to-water heat exchanger of shell-and-tube type. By this means, the temperature of the 130-k*l* tank was kept between 6°C and 14°C throughout the year.

The water of the 130-k*l* tank was pumped to the 10-k*l* tank installed near the

JARE-7 engine room. The waste heat recovery system for the 45-kVA diesel-electric generators was not changed. The system shown in Fig. 16 was operated satisfactorily and saved much fuel.

In summer, pond water can be used. In winter, by putting into the pond a 2-kW electric heater, the bottom water could be prevented from freezing. To pump up the water, a fire pump and hoses were used to reduce the pumping time. However, the salt content increased gradually due to growth of the ice. Therefore, only the 10-k*l* tank for melting snow or ice is used from March through November.

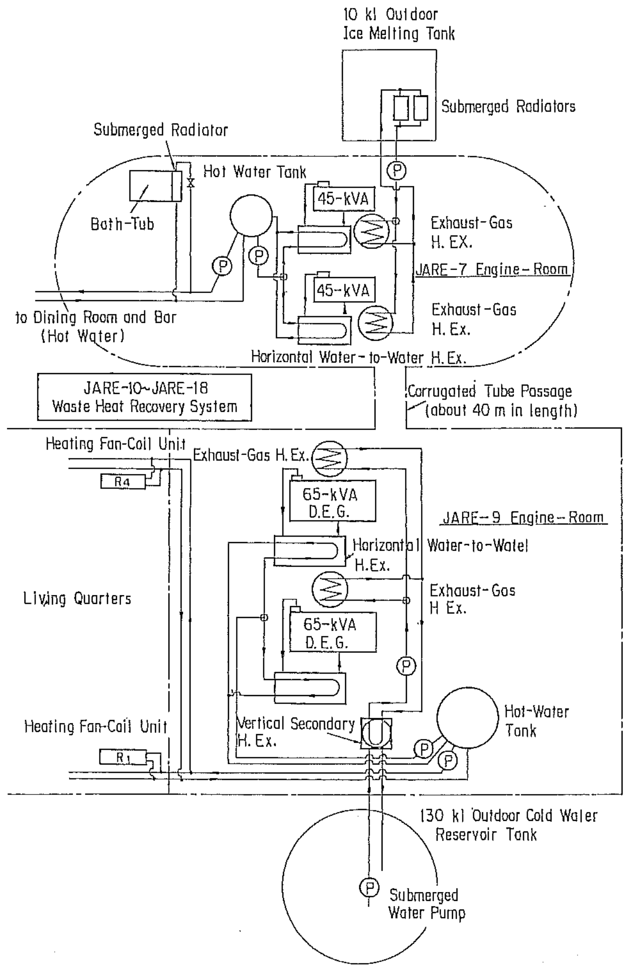

Fig. 16. Waste heat recovery systems in JARE-7 and JARE-9 engine rooms used by JARE-10/18 (1968/76).

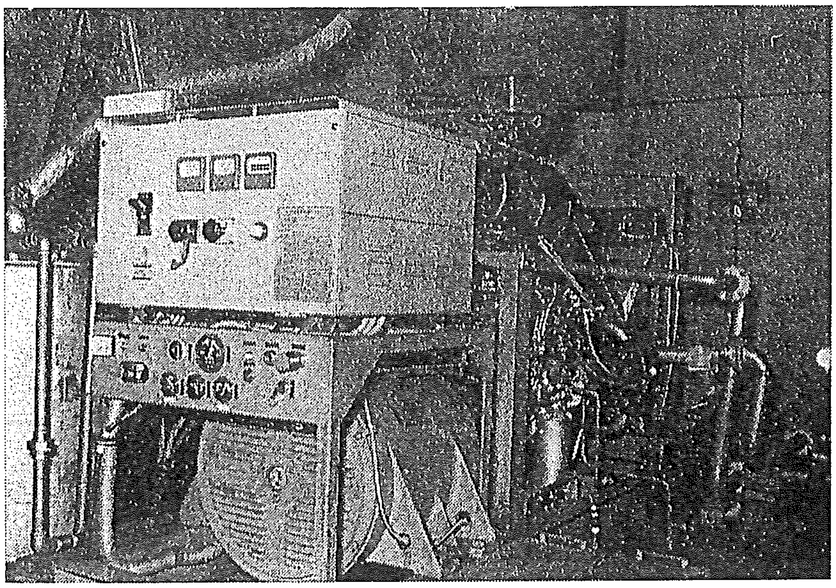

Fig. 17. Inside view of JARE-9 engine room containing 65-kVA diesel-electric generators. Piping for coolant heat recovering system is shown on the right side of the engine and a shell-and-coil type exhaust-gas heat exchanger for heating a 130-kl cold water outdoor tank is shown on the left side.

4. A 110-kVA Diesel-Electric Power Generating Plant with Waste Heat Recovery System

At the commencement of 110-kVA diesel-electric generator operation, the waste heat recovery system was changed from that shown in Fig. 16 to that in Fig. 18. Two photographs show the newly build system (Figs. 19 and 20). In

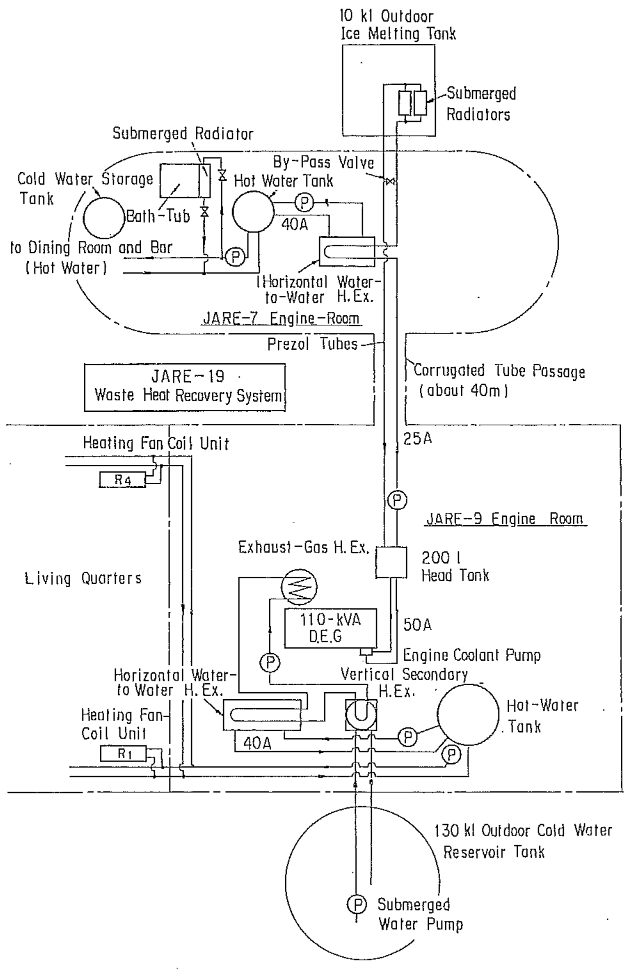

Fig. 18. JARE-9 engine room containing one 110-kVA diesel-electric generator and new heat recovering system combined with that of JARE-7 engine room in JARE-19 (1977/79).

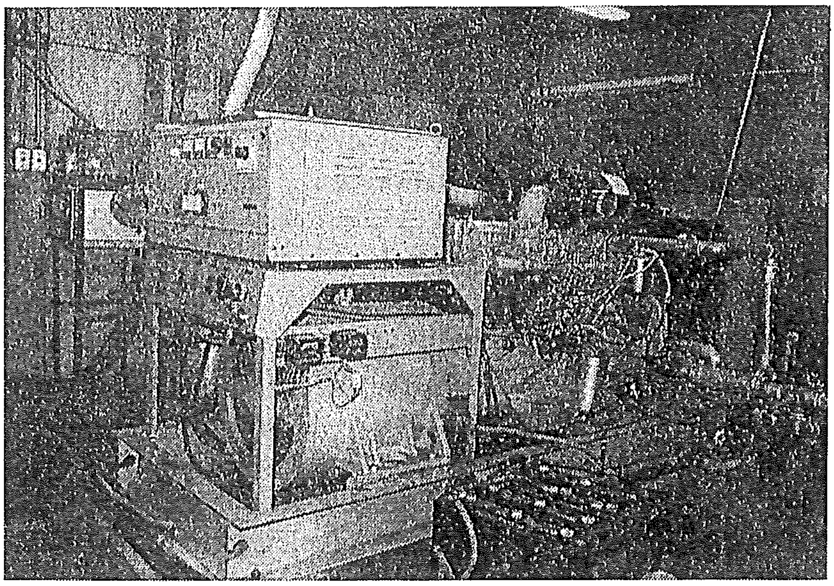

Fig. 19. 110-kVA diesel-electric generator for JARE-19 (1977/79) installed in the JARE-9 engine room. Right side view of engine and piping to and from the headtank for recovery of coolant heat are shown.

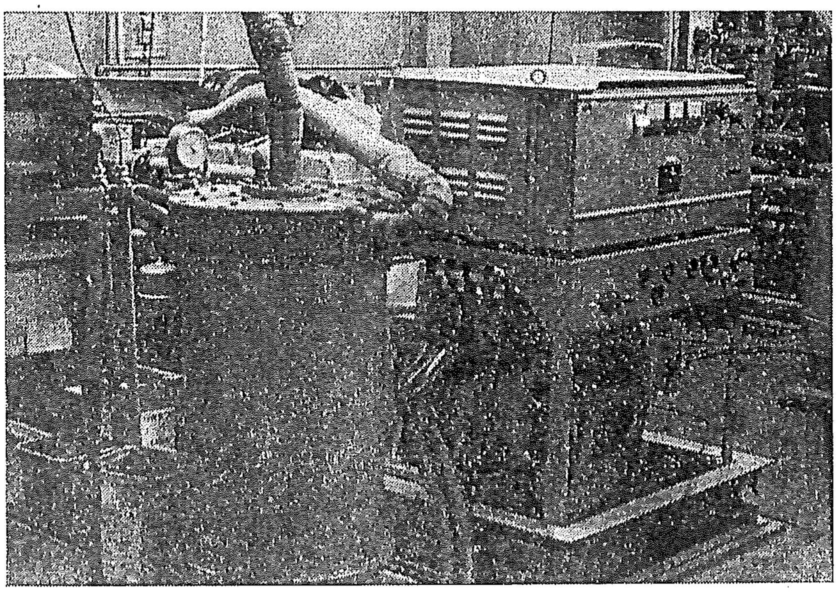

Fig. 20. Left side view of No. 1 110-kVA diesel-electric generator equipped with a fin-tube type exhaust-gas heat exchanger AL-F-1 prepared for JARE-19 (1977/79).

the new system, coolant heat of the 110-kVA diesel engine was pumped to the JARE-7 engine room to provide heat to the horizontal coolant heat exchanger of shell-and-tube type and the 10-k*l* snow-melting tank. The temperature of water in the 10-k*l* tank was kept at 33–45°C. The water heated to 42–50°C by the coolant heat exchanger was reserved in a hot-water tank in the JARE-7 engine room, and supplied to the living quarters and heated water in the bathtub through a submerged radiator.

On the other hand, the exhaust-gas energy of the 110-kVA diesel engines was used to heat water by means of an exhaust-gas heat exchanger. The hot water was recirculated through a horizontal water-to-water heat exchanger (shell-and-tube type) and a vertical water-to-water heat exchanger (shell-and-tube type) as shown in Fig. 18. The hot water heated by the former was reserved in a hot-water tank in the JARE-9 engine room and was used for heating rooms of the JARE-9 engine room. The hot water heated by the latter was used to prevent from freezing of water in the 130-k*l* tank.

On the occasion of installation of another 110-kVA generator, the heat recovery system was changed from that of Fig. 18 to that of Fig. 21. Thus, the electric power source of Syowa Station was simplified from the double sources of 65-kVA and 45-kVA generators separated in two engine rooms to a single source from 110-kVA generators in the JARE-9 engine room. The simplified single

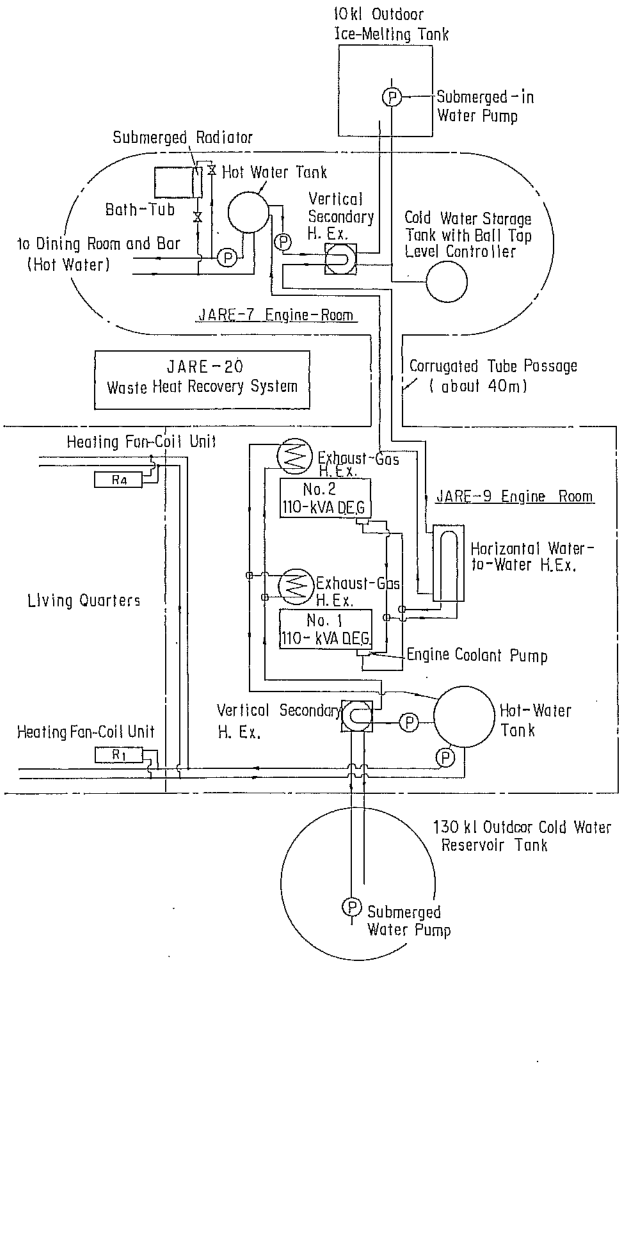

Fig. 21. JARE-9 engine room containing set of two 110-kVA diesel-electric generators and renewed waste heat recovering systems combined with that of JARE-7 engine room in 1979 (JARE-20).

source brought a considerable saving of fuel. In Fig. 22, the dotted lines show the monthly mean and peak loads of the 110-kVA generating system during the period from February 1978 to September 1979. The solid lines show those of the double sources as the mean values for JARE-16, -17 and -18. Both of them are nearly equal, but fuel consumption and specific fuel consumption (SFC) shown in Fig. 23 are very different. The fuel consumption represented by k*l*/month and SFC expressed in *l*/kWh is remarkably decreased by using the single 110-kVA unit. The former decreased from 14.0–15.5 k*l*/month to 9.6–12.6 k*l*/month (68–81%) and the latter decreased from 0.382–0.432 *l*/kWh to 0.282–0.356 *l*/kWh (78–82%). The lowered fuel consumption is mainly due to the decreasing ratio of mechanical loss to output of a 110-kVA diesel engine compared with that of the 45-kVA and 65-kVA diesel engines and with the fact that the equivalence ratio at normal load of 110-kVA diesel engine coincides with its minimum SFC.

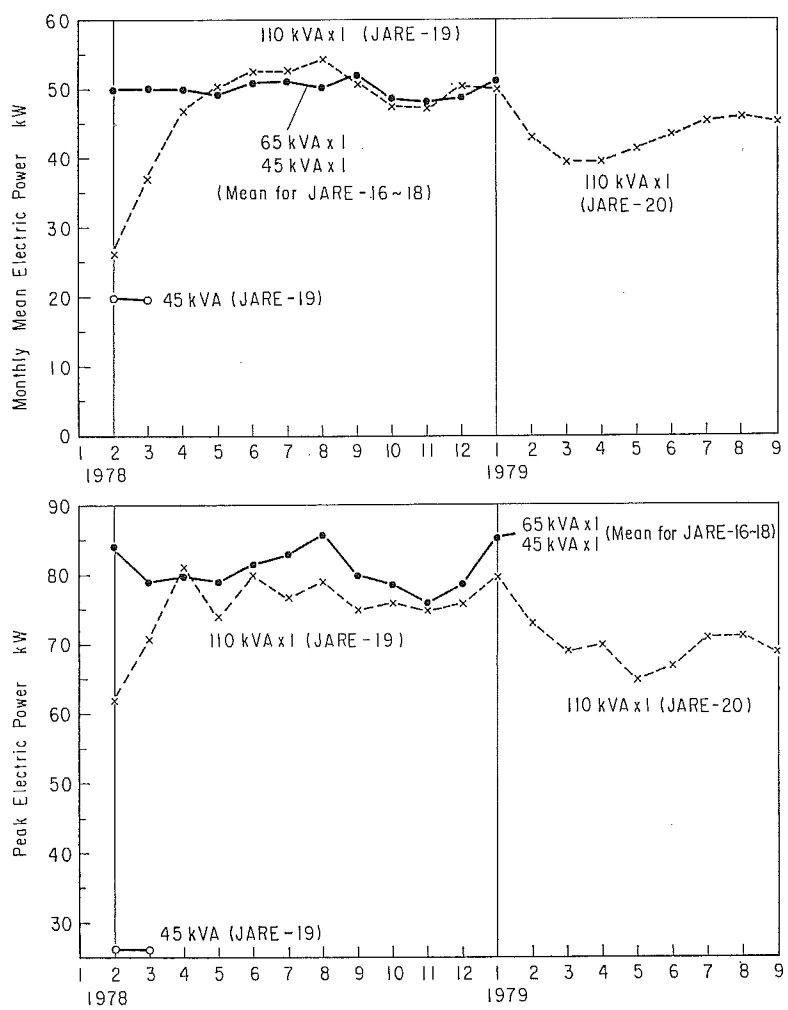

Fig. 22. Monthly mean and peak electric loads at Syowa Station from February 1978 to September 1979.

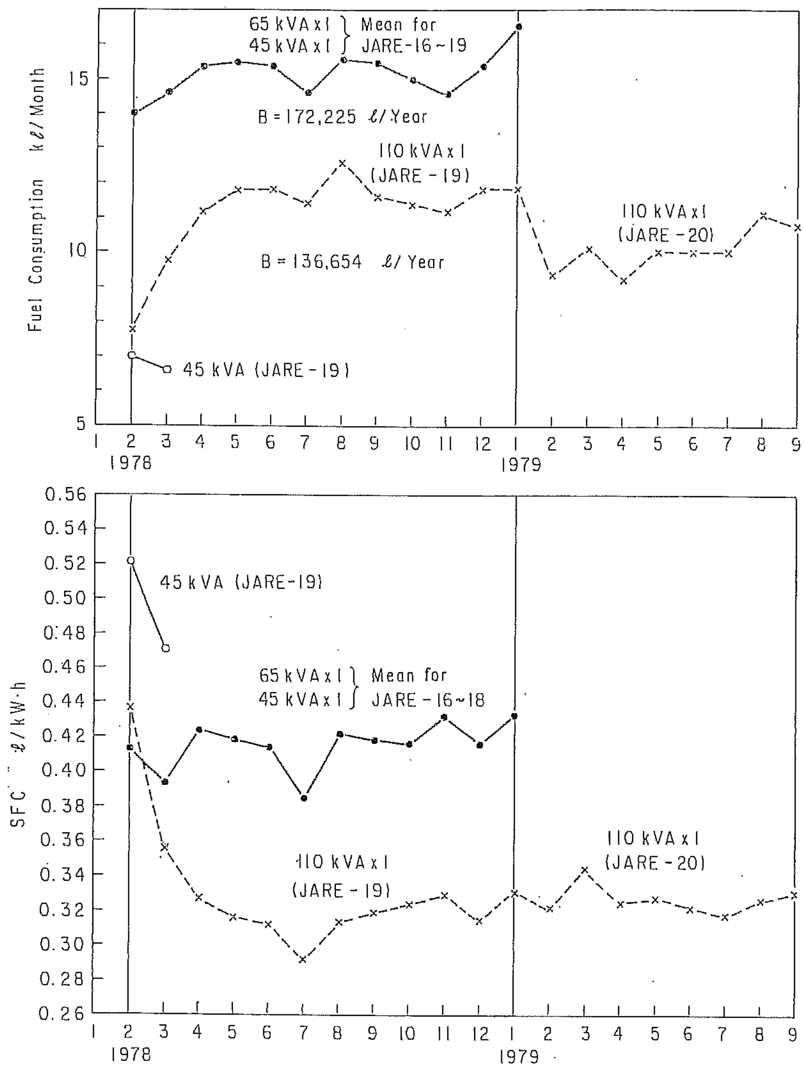

Fig. 23. *Comparison of monthly mean SFC for 110-kVA diesel-electric generator and for two electric generators, that are one 45-kVA generator in JARE-7 engine room and one 65-kVA generator in JARE-9 engine room.*

5. Estimation of Heat Balance of Diesel-Electric Generators

The following subscripts and symbols will be used herein.

Subscript;

- g: exhaust gas of diesel engine,
- a: air aspirated,
- gd: dry exhaust gas without H_2O (its unit weight is represented by kg*),
- w: steam, s: saturated state, ws: saturated steam,
- aw: moisture contained in atmospheric air,
- ad: dry air,
- gw: moisture contained in exhaust gas.

Symbols;

- h: enthalpy, kcal/kg,
- h_{gd}: enthalpy of dry exhaust gas, kcal/kg*,
- h_w: enthalpy of superheated steam, kcal/kg,
- h_w'': enthalpy of saturated steam (including the latent heat of evaporation. The zero point of h_w and h_w'' is set for the saturated water at 0°C), kcal/kg,
- H: calorific value of fuel, kcal/kg or enthalpy flow rate, kcal/h,
- p: atmospheric pressure, mmHg abs.,
- t: temperature, °C,
- t_a: atmospheric air temperature, °C,
- p_{sa}: saturated vapor pressure of steam for t_a, mmHg abs.,
- λ_a: relative humidity,
- x_a: absolute humidity of atmosphere, kg/kg*,
- G_a: weight flow rate of air aspirated by diesel engine, kg/h,
- G_{ad}: weight flow rate of dry air contained in G_a, kg*/h
- G_{aw}: weight flow rate of moisture contained in G_a, kg/h,
- G_g: weight flow rate of exhaust gas, kg/h,
- G_{gd}: weight flow rate of dry exhaust gas contained in G_g, kg*/h,
- G_{gw}: weight flow rate of steam contained in G_g, kg/h,

x_a: absolute humidity of air, kg/kg*,
x_s: saturated absolute humidity of exhaust gas, kg/kg*,
R_{gd}: gas constant for dry exhaust gas, kgm/kg*K,
R_w: gas constant for steam ($R_w=47.05$ kgm/kgK),
G_{ws}: weight flow rate of saturated steam in exhaust gas, kg/h,
$L_{\min}$: stoichiometric air-fuel ratio, $L_{\min}=14.2$ for gas oil,
ϕ: equivalence ratio of exhaust gas,
G_f: weight flow rate of fuel supplied to engine, kg/h,
B: volume flow rate of fuel supplied to engine, l/h,
Q: heat flow rate, kcal/h.

The flow rates of dry air and steam contained in atmospheric air aspirated by a diesel engine are given by

$$G_{ad}=G_a/(1+x_a) \text{ kg*/h}, \tag{1}$$

$$G_{aw}=x_a G_{ad}=x_a G_a/(1+x_a) \text{ kg/h}, \tag{2}$$

$$x_a=0.622\, \lambda_a p_{sa}/(p-\lambda_a p_{sa}) \text{ kg/kg*}. \tag{3}$$

If the fuel supply G_f and air flow rate G_a are measured, the flow rate of the exhaust gas can be given by

$$G_g=G_{ad}+G_f \text{ kg/h}, \tag{4}$$

$$G_a=G_{ad}+G_{aw} \text{ kg/h}, \tag{5}$$

and

$$G_g=G_{gd}+G_{gw} \text{ kg/h}. \tag{6}$$

When the fuel G_f is burned in the engine cylinder with the supply of air G_a, containing dry air G_{ad}, the equivalence ratio of combustion gas is given by

$$\phi=L_{\min}G_f/G_{ad}. \tag{7}$$

The flow rate of the dry exhaust gas can be calculated by

$$G_{gd}=G_g/(1+0.0806\,\phi) \text{ kg*/h}, \tag{8}$$

and the steam content of G_g is

$$G_{gw}=0.0806\,\phi G_{gd} \text{ kg/h}. \tag{9}$$

The total content of steam in the exhaust gas becomes

$$G_w=G_{aw}+G_{gw} \text{ kg/h}. \tag{10}$$

The higher calorific value of the fuel containing the latent heat of steam condensation is represented by H_o and its lower calorific value by H_u, and the values for gas oil are given as

$$H_o = 10837 \text{ kcal/kg}, \qquad H_u = 10162 \text{ kcal/kg}.$$

The heat supplied to an engine by the fuel is

$$Q_f = H_o G_f \text{ kcal/h}. \tag{11}$$

The temperatures of the exhaust gas are defined as

t_{ei}: temperature at the inlet of a turbocharger, °C,
t_{eo}: temperature at the outlet of a turbocharger, °C,
t_{g1}: temperature at the inlet of a heat exchanger, °C,
t_{g2}: temperature at the outlet of a heat exchanger, °C.

If condensation of steam contained in the exhaust gas does not occur, the enthalpy flow rate for these four temperatures can be calculated by

$$H_{ei} = (h_{gd})_{ei} G_{gd} + (h_w)_{ei} G_w \text{ kcal/h}, \tag{12}$$
$$H_{eo} = (h_{gd})_{eo} G_{gd} + (h_w)_{eo} G_w \text{ kcal/h}, \tag{13}$$
$$H_{g1} = (h_{gd})_{g1} G_{gd} + (h_w)_{g1} G_w \text{ kcal/h}, \tag{14}$$
$$H_{g2} = (h_{gd})_{g2} G_{gd} + (h_w)_{g2} G_w \text{ kcal/h}. \tag{15}$$

The heat drop in a turbocharger is

$$Q_T = H_{ei} - H_{eo} \text{ kcal/h}, \tag{16}$$

and the heat flow rate contained in exhaust gas is $Q_r = H_{g1}$ and the available heat recovered by an exhaust-gas heat exchanger is

$$Q_R = H_{g1} - H_{g2} \text{ kcal/h}. \tag{17}$$

If some part of the steam contained in the exhaust gas condenses in the exhaust-gas heat exchanger, eq. (15) should be replaced by

$$H_{g2} = (h_{gd})_{g2} G_{gd} + (h_w'')_{g2} G_{ws} \text{ kcal/h}, \tag{15'}$$

where the saturated steam flow rate can be calculated by

$$G_{ws} = (x_s)_{g2} G_{gd} \text{ kg/h}, \tag{18}$$
$$(x_s)_{g2} = (R_{gd}/R_w)\ [p_s/(p-p_s)]_{g2} \text{ kg/kg*}, \tag{19}$$
$$R_{gd} = 29.27 - 0.93\phi \text{ kgm/kg*K}, \tag{20}$$
$$(x_s)_{g2} = (0.622 - 0.0198\phi)\ [p_s/(p-p_s)]_{g2} \text{ kg/kg*}. \tag{19'}$$

The total flow rate of the outlet gas from the heat exchanger in this case is decreased to

$$G_{\text{out}} = G_{gd} + G_{ws} \text{ kg/h}, \tag{21}$$

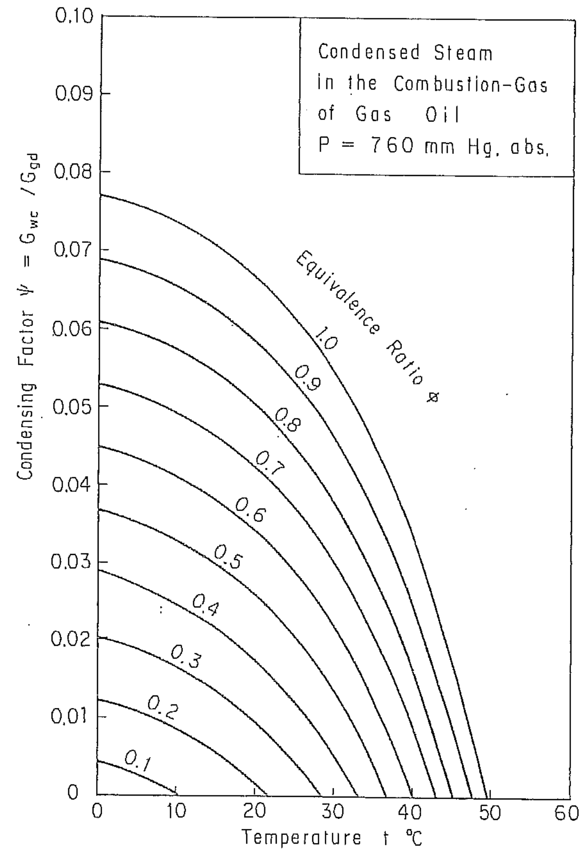

Fig. 24. Condensation factor ψ, the weight ratio of condensed water to 1 kg of dry combustion gas of gas oil at any temperature t and equivalence ratio φ.*

Table 5. Enthalpy of dry combustion gas of gas oil, h_{gd} kcal/kg ($C=0.8584$, $H=0.1269$, $N=0.0147$, $L_{min}=14.2$, Equivalence ratio $\phi=L_{min}G_f/G_{ad}$).*

t°C	ϕ=0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
0	0	0	0	0	0	0	0	0	0	0	0
50	12.02	12.02	12.00	11.97	11.95	11.94	11.92	11.90	11.89	11.87	11.85
100	24.08	24.07	24.05	24.01	23.98	23.96	23.93	23.91	23.89	23.86	23.83
150	36.20	36.19	36.17	36.13	36.10	36.08	36.05	36.03	36.02	35.98	35.95
200	48.40	48.40	48.39	48.36	48.33	48.32	48.30	48.28	48.28	48.25	48.22
250	60.71	60.72	60.72	60.70	60.69	60.69	60.68	60.67	60.69	60.67	60.64
300	73.14	73.17	73.18	73.18	73.18	73.20	73.21	73.21	73.25	73.25	73.23
350	85.70	85.75	85.78	85.80	85.82	85.86	85.89	85.91	85.97	85.99	85.99
400	98.40	98.47	98.53	98.57	98.61	98.67	98.73	98.77	98.85	98.90	98.92
450	111.24	111.34	111.42	111.49	111.55	111.64	111.73	111.79	111.90	111.97	112.02
500	124.22	124.35	124.46	124.56	124.65	124.76	124.88	124.97	125.11	125.21	125.29
550	137.34	137.50	137.64	137.78	137.90	138.04	138.19	138.31	138.48	138.61	138.72
600	150.60	150.79	150.97	151.14	151.29	151.47	151.65	151.80	152.01	152.17	152.31
650	163.99	164.22	164.43	164.64	164.82	165.04	165.25	165.44	165.68	165.88	166.06
700	177.51	177.77	178.02	178.27	178.49	178.74	178.99	179.22	179.49	179.73	179.95

and the condensed steam in the heat exchanger is

$$G_{wc}=G_w-G_{ws}=(G_{aw}+G_{gw})-G_{ws}\ \text{kg/h}. \tag{22}$$

If the steam content in the air, G_{aw}, is negligible compared with G_{gw},

$$G_{wc}=G_{gw}-G_{ws}=\psi G_{gd}\ \text{kg/h}, \tag{23}$$

$$\psi=G_{wc}/G_{gd}=0.0806\phi-(0.622-0.0198\phi)\ [p_s/(p-p_s)]_{g2}. \tag{24}$$

As a standard state, $p=760$ mmHg abs. and $t_{g2}=0°$C, and $p_s=4.6$ mmHg abs. are adopted. Then eq. (24) becomes

$$\psi_0=0.0807\phi-0.00381. \tag{25}$$

The calculated condensing factor ψ from eq. (24) is shown in Fig. 24. It shows that the allowable lowest temperature in an exhaust-gas heat exchanger should be kept higher than 50°C for avoiding condensation of steam and corrosion of heating surfaces.

The maximum recovery efficiency of a heat exchanger can be estimated from eqs. (14) and (17),

$$\eta_{\text{rec}}=Q_R/H_{g1}=1-(H_{g2}/H_{g1}). \tag{26}$$

In these calculations, h_{gd} is shown in Table 5 as a function of temperature and equivalence ratio ϕ, although the effect of the latter is not very great. The enthalpies of steam h_w and h_w'' should be read from a steam table (JSME, 1968).

As an example,

$p=773.0$ mmHg abs., $t=11.0$°C, $\lambda_a=0.77$, $p_s=9.8$ mmHg abs.,
$x_a=0.00613$ kg/kg*,
$G_a=289.8$ kg/h, $G_f=9.04$ kg/h,
$G_{ad}=G_a/(1+x_a)=289.8/1.00613=288.0$ kg*/h,
$G_{aw}=G_a-G_{ad}=289.8-288.0=1.8$ kg/h,
$\phi=L_{\min}G_f/G_{ad}=14.2(9.04)/288.0=0.446$,
$G_g=G_{ad}+G_f=288.0+9.04=297.0$ kg/h,
$G_{gd}=G_g/(1+0.0806\phi)=297.0/(1+0.0806\times0.446)$
$=297.0/1.0359=286.7$ kg*/h,
$G_{gw}=G_g-G_{gd}=297.0-286.7=10.3$ kg/h,
$G_w=G_{aw}+G_{gw}=1.8+10.3=12.1$ kg/h,
$Q_f=H_oG_f=(10837)(9.04)=98000$ kcal/h,
$t_{ei}=375$°C, $h_{gd}=92.25$ kcal/kg*, $h_w=770.7$ kcal/kg,
$t_{eo}=345$°C, $h_{gd}=84.58$ kcal/kg*, $h_w=756.1$ kcal/kg,

$t_{g1}=335°C$, $h_{gd}=82.04$ kcal/kg*, $h_w=751.2$ kcal/kg,
$t_{g2}=85°C$, $h_{gd}=20.36$ kcal/kg*, $h_w''=633.4$ kcal/kg,
$R_{gd}=29.27-0.93\phi=29.27-0.93(0.446)=29.27-0.41=28.86$ kgm/kg*K
at $t_{g2}=85°C$, $p_s=433.6$ mmHg abs., $p_s/(p-p_s)=433.6/(773.0–433.6)=1.278$,
$(x_s)_{g2}=(28.86/47.05)(1.278)=0.784$ kg/kg*,
$G_{ws}=(x_s)_{g2}G_{gd}=0.784(286.7)=224.8$ kg/h.

G_{ws} is greater than $G_w=12.1$ kg/h, so that no condensing occurs. The enthalpy of the exhaust gas can be calculated as follows from eqs. (12), (13), (14) and (15):

$t_{ei}=375°C$, $H_{ei}=(92.25)(286.7)+(770.7)(12.1)$
$=26450+9320=35770$ kcal/h,
$t_{eo}=345°C$, $H_{eo}=(84.58)(286.7)+(756.1)(12.1)$
$=24250+9150=33400$ kcal/h,
$t_{g1}=335°C$, $H_{g1}=(82.04)(286.7)+(751.2)(12.1)$
$=23520+9090=32610$ kcal/h,

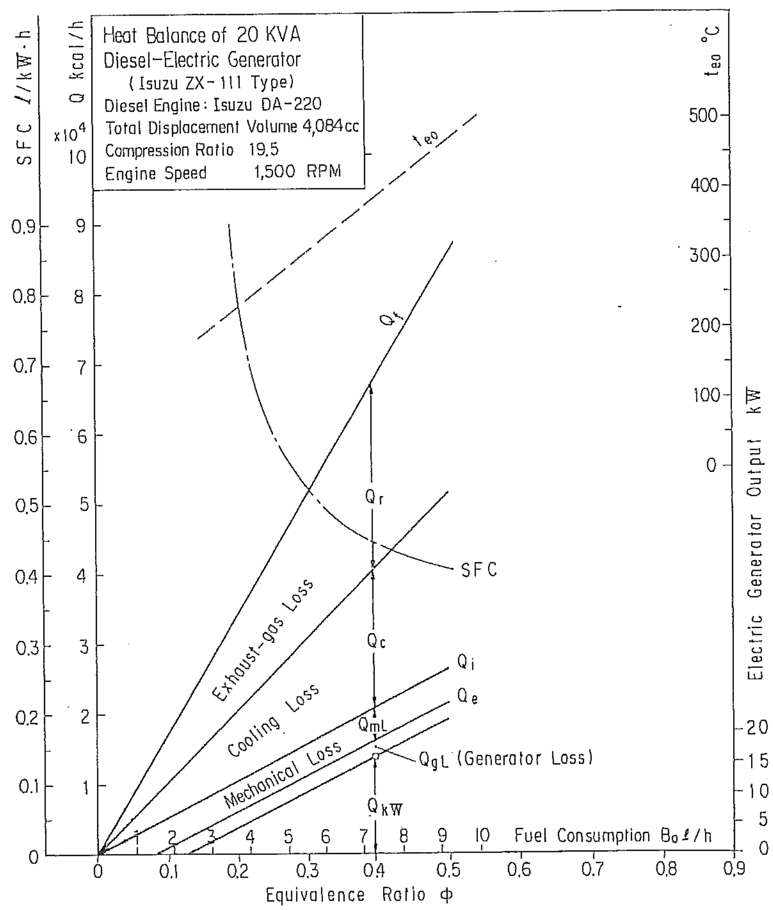

Fig. 25. Heat balance of 20-kVA diesel-electric generator prepared for JARE-1 (1956/58). Meidensha ZX-111 type, diesel engine: DA-220, total piston displacement volume 4084 cc, compression ratio 19.5, 33PS/1500 rpm.

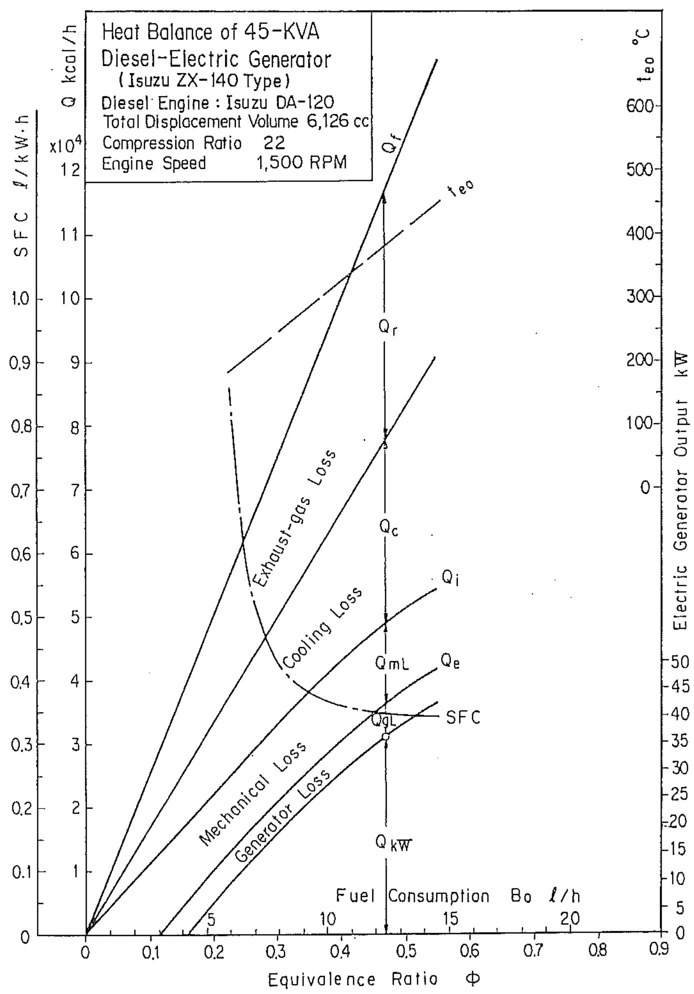

Fig. 26. Heat balance of 45-kVA diesel-electric generator used for JARE-7/9. Meidensha ZX-140 type, diesel engine: Isuzu DA-120, total piston displacement volume 6126 cc, compression ratio 22, 65.5PS/1500 rpm.

$$t_{g2}=85°\text{C}, \qquad H_{g2}=(20.36)(286.7)+(633.4)(12.1)$$
$$=5840+7660=13500 \text{ kcal/h}.$$

The energy absorbed by the turbocharger

$$Q_T=H_{ei}-H_{eo}=35770-33400=2370 \text{ kcal/h } (2.76 \text{ kW}),$$

and the energy recovered by the exhaust-gas heat exchanger

$$Q_R=H_{g1}-H_{g2}=32610-13500=19100 \text{ kcal/h}.$$

The recovery efficiency

$$\eta_{\text{rec}}=Q_R/H_{g1}=19100/32610=0.586.$$

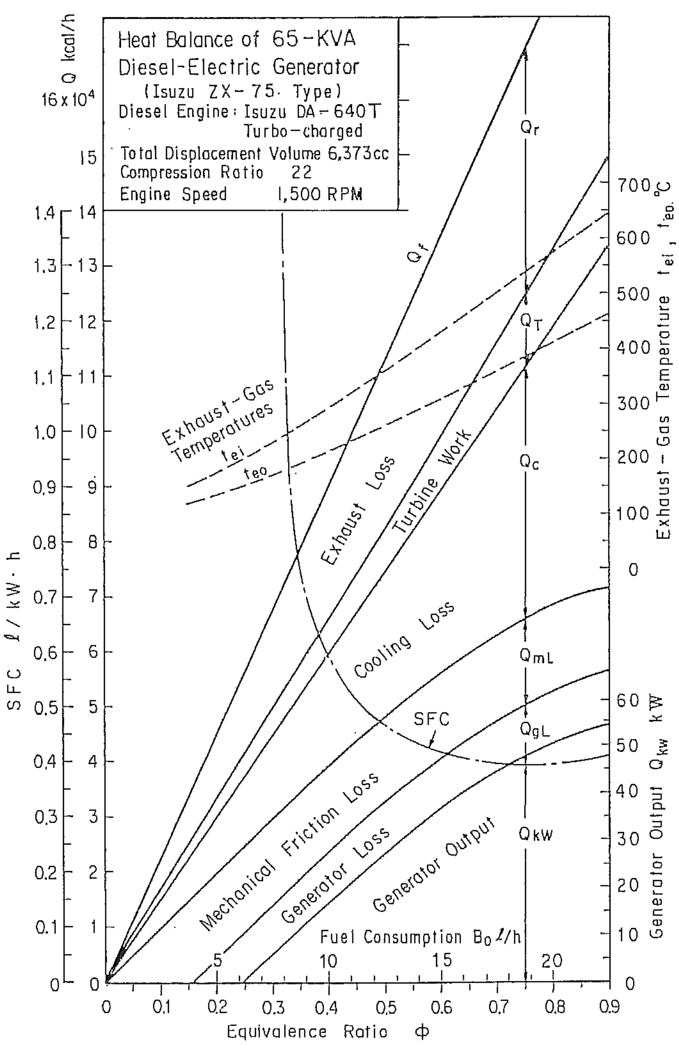

Fig. 27. Heat balance of 65-kVA diesel-electric generator used for JARE-9/19. Meidensha ZX-75 type, diesel engine: Isuzu DA-640T, total piston displacement volume 6373 cc, compression ratio 22, 82PS/1500 rpm.

If the heat supplied by the fuel, Q_f calculated by eq. (11), heat (Q_f-Q_r), and heat $(Q_f-Q_r-Q_T)$ are plotted against the equivalence ratio ϕ as shown in Figs. 25, 26, 27 and 28, these three curves pass through the origin. When the output voltage kV, output current A in ampere and power factor $\cos\Phi$ are given, the output power of the electric generator Q_{kW} can be calculated by

$$Q_{kW}=860\ \text{kVA}\cos\Phi=860\ \text{kW kcal/h}, \tag{27}$$

where the power factor $\cos\Phi$ is 80% and the generator loss Q_{gL} is estimated to be about 17% of Q_{kW}, so that the power necessary for driving the generator, that is, power equal to the brake output of the diesel engine, can be estimated by

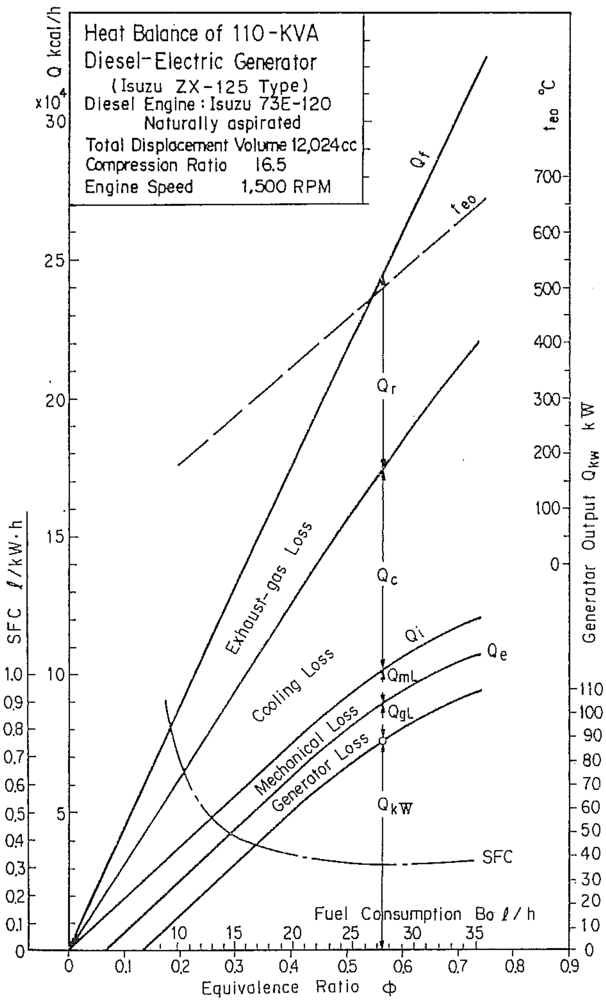

Fig. 28. Heat balance of 110-kVA diesel-electric generator prepared for JARE-19. Meidensha XZ-125 type, diesel engine: Isuzu 73E-120, total piston displacement volume 12024 cc, compression ratio 16.5, 140PS/1500 rpm.

$$Q_e = Q_{kW} + Q_{gL} = 1.17\, Q_{kW} \text{ kcal/h.} \tag{28}$$

As shown in these figures, the curve Q_e does not pass through the origin but cuts the abscissa axis at a point below the origin. The negative value of Q at the origin indicates the mechanical loss Q_{mL} of the diesel engine in kcal/h.

Hence the indicated output of the diesel engine can be obtained by

$$Q_i = Q_e + Q_{mL} = Q_{kW} + Q_{gL} + Q_{mL} \text{ kcal/h.} \tag{29}$$

The curve Q_i should also pass through the origin as shown in these figures, and the cooling loss of the diesel engine can be approximately given by

$$Q_c = Q_f - Q_r - Q_T - Q_i = Q_f - Q_r - Q_T - Q_{kW} - (Q_{mL} + Q_{gL}) \text{ kcal/h.} \tag{30}$$

Table 6. The heat balance of diesel-electric generators prepared for JARE.

a. 20-kVA diesel electric generator (Isuzu ZX-111 type, diesel engine DA-220).

Run No.	Generator output (kVA)	(kW)	ϕ	t_{g1} (°C)	Q_f	Q_{kW}	Q_r (kcal/h)	Q_c	Q_{mL}	Q_{gL}	G_a (kg/h)	SFC (l/kWh)
1	20	16	0.390	380	66430	13760	25530	20400	4400	2340	223	0.446
2	15	12	0.326	327	55490	10320	21550	16880	4400	2340	223	0.498
3	10	8	0.258	274	43890	6880	17630	12640	4400	2340	223	0.590
4	6.8	5.44	0.220	243	37500	4678	15320	10760	4400	2340	223	0.741
5	5	4	0.195	221	33160	3440	13760	9220	4400	2340	223	0.892

b. 45-kVA diesel-electric generator (Isuzu ZX-140 type, diesel engine DA-120).

Run No.	Generator output (kVA)	(kW)	ϕ	t_{g1} (°C)	Q_f	Q_{kW}	Q_r (kcal/h)	Q_c	Q_{mL}	Q_{gL}	G_a (kg/h)	SFC (l/kWh)
1	45	36	0.473	381	117600	30960	39190	29690	12500	5260	326	0.347
2	40	32	0.431	350	107300	27520	35580	26440	12500	5260	326	0.356
3	35	28	0.386	320	95910	24080	32130	21940	12500	5260	326	0.364
4	30	24	0.348	288	86590	20640	28660	19530	12500	5260	326	0.383
5	25	20	0.314	253	78030	17200	24920	18150	12500	5260	326	0.415
6	20	16	0.287	220	71520	13760	21720	18280	12500	5260	326	0.475
7	15	12	0.257	190	63940	10320	18750	17110	12500	5260	326	0.567
8	10	8	0.238	158	59280	6880	15750	18890	12500	5260	326	0.787

c. 65 kVA diesel-electric generator (Isuzu ZX-75 type, diesel engine DA-640-T).

Run No.	Generator output (kVA)	(kW)	ϕ	t_{ei} (°C)	t_{eo} (°C)	Q_f	Q_{kW}	Q_r	Q_c (kcal/h)	Q_T	Q_{mL}	Q_{gL}	G_a (kg/h)	SFC (l/kWh)
1	65	52	0.805	585	400	185300	44720	44100	56990	16590	15300	7600	301	0.392
2	60	48	0.745	535	367	170000	41280	40060	50830	14930	15300	7600	300	0.389
3	55	44	0.685	485	335	156000	37840	36080	46360	12820	15300	7600	299	0.389
4	50	40	0.640	450	310	145000	34400	33000	42900	11800	15300	7600	298	0.399
5	45	36	0.590	410	285	133000	30960	28910	39950	10280	15300	7600	296	0.407
6	40	32	0.550	380	265	125000	27520	27730	37460	9390	15300	7600	296	0.427
7	35	28	0.507	345	240	114000	24080	24940	33660	8420	15300	7600	295	0.446
8	30	24	0.465	315	220	105000	20640	22700	31240	7530	15300	7600	296	0.481
9	25	20	0.427	290	200	96800	17200	20580	29070	7050	15300	7600	297	0.532
10	20	16	0.390	260	185	88600	13760	18890	27220	5830	15300	7600	298	0.608
11	15	12	0.350	235	165	80100	10320	16810	24660	5415	15300	7600	300	0.733
12	10	8	0.313	215	150	71800	6880	15160	21860	5005	15300	7600	301	0.987

t_{ei}: inlet gas temperature of an exhaust-gas turbine, °C, t_{eo}: outlet gas temperature of the exhaust-gas turbine, °C, Q_T: output power of the exhaust-gas turbine, kcal/h.

d. 110 kVA diesel-electric generator (Isuzu ZX-125 type, diesel engine 73E-120).

Run No.	Generator output (kVA)	(kW)	ϕ	t_{g1} (°C)	Q_f	Q_{kW}	Q_r (kcal/h)	Q_c	Q_{mL}	Q_{gL}	G_a (kg/h)	SFC (l/kWh)
1	110	88	0.565	500	248000	75680	91800	55000	12500	13000	575	0.312
2	105	84	0.540	475	237000	72240	87200	52100	12500	13000	577	0.314
3	100	80	0.515	455	228000	68800	83100	49500	12500	13000	579	0.316
4	95	76	0.495	437	219000	65360	79700	48400	12500	13000	580	0.320
5	90	72	0.472	417	209000	61920	75500	46100	12500	13000	581	0.322
6	85	68	0.450	397	199000	58480	71600	43400	12500	13000	582	0.326
7	80	64	0.428	380	191000	55040	68500	42000	12500	13000	584	0.331
8	75	60	0.408	360	182000	51600	64600	40300	12500	13000	585	0.337
9	70	56	0.388	342	173000	48160	61200	38200	12500	13000	586	0.344
10	65	52	0.369	325	166000	44720	57900	37900	12500	13000	587	0.353
11	60	48	0.353	313	158000	41280	55700	35500	12500	13000	588	0.365
12	55	44	0.333	295	150000	37840	52300	34400	12500	13000	589	0.377
13	50	40	0.318	284	143000	34400	50200	32900	12500	13000	590	0.397
14	45	36	0.300	265	135000	30960	46800	31700	12500	13000	591	0.417
15	40	32	0.282	250	128000	27520	44000	31000	12500	13000	593	0.444
16	35	28	0.262	234	119000	24080	40900	28600	12500	13000	594	0.470
17	30	24	0.243	218	111000	20640	37900	26900	12500	13000	595	0.511
18	25	20	0.225	200	102000	17200	34800	24500	12500	13000	596	0.567
19	20	16	0.206	185	93800	13760	32000	22600	12500	13000	597	0.650

ϕ: equivalence ratio, $\phi=1/n$, n: excess-air factor, t_{g1}: inlet gas temperature of an exhaust-gas heat exchanger, °C, Q_f: heat supplied by fuel, kcal/h, Q_{kW}: output power of generator, kcal/h, Q_r: exhaust-gas heat energy, kcal/h, Q_c: coolant heat energy, kcal/h, Q_{mL}: mechanical friction loss of engine, kcal/h, Q_{gL}: generator loss, kcal/h, G_a: weight flow rate of air aspirated by engine, kg/h, SFC: specific fuel consumption, l/kWh.

In the case where the equivalence ratio ϕ is greater than 0.5, the value of Q_c obtained by this method may be slightly greater than its true value because it includes some heat loss due to incomplete combustion of fuel indicated by generation of carbon soot in this region.

Tables 6a, b, c and d show the heat balances for four kinds of diesel-electric generators installed at Syowa Station during JARE-1 and JARE-21.

The specific fuel consumption (SFC) shown in these figures was calculated by

$$\mathrm{SFC}=\mathrm{B/kW} \qquad l/\mathrm{kWh}, \tag{31}$$

and the power plant efficiency can be calculated by

$$\eta_o=860/[(\mathrm{SFC})\rho_f H_o], \tag{32}$$