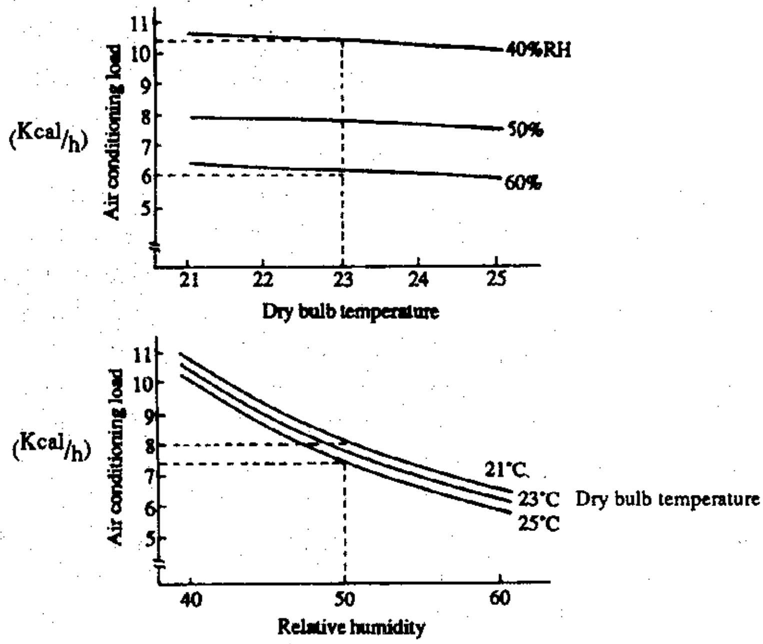
Figure 61 Energies Required for Temperature and Humidity Changes



(8) Improving control method

As mentioned before, control of heat load is available by method which varies flow rate, and by method which varies set temperature with flow rate set constant. Reduction effect of drive power for the fan and pump is larger in the former, and it results in energy conservation.

As control methods for flow rate, there are control of dampers and vanes, and various rotating speed controls. Power consumption is ranked as discharge damper control > inlet vane control > rotating speed control. As rotating speed control by VVVF is easily available for existing motor facilities with considerable effects, it should be studied first among other possibilities.

(9) Regular maintenance

a. Cooling water piping
Accumulation of scale and sludge in piping will increase resistance and require more pump output for the same flow rate. If the water quality is poor, it will naturally accelerate accumulation of scale and sludge. Therefore, control of water quality is necessary. Reference values for the control criterion are shown in Table 40.

Table 40 Quality Standard of Cooling Water

(Japan Refrigeration and Air Conditioning Industrial Association Standard)

	Item		Standard value for makeup water	Standard value for cooling water *1	Tendency *3	
					Corrosion	Scale
Standard item	PH (25°C)		6.0 ~ 8.0	6.0 ~ 8.0	O	O
	Conductivity	(mv/m)	200 or less	500 or less (1,000 or less)	O	
	Chlorine ion Cl	(ppm)	50 or less	200 or less	O	
	Sulfuric acid ion SO_4	(ppm)	50 or less	200 or less	O	
	Total iron Fe	(ppm)	0.3 or less	1.0 or less *2	O	O
	M Alkalinity $CaCO_3$	(ppm)	50 or less	100 or less		O
	Total hardness $CaCO_3$	(ppm)	50 or less	200 or less		O
Reference item	Sulfur ion S	(ppm)	Not be detected	Not be detected	O	
	Ammonium ion NH_4	(ppm)		Not be detected	O	
	Silicon oxide SiO_2	(ppm)		50 or less		O

*1 Cooling water means water passing through condenser for both transient and circulation systems.

*2 Standard value for plastic piping shall be 0.5 ppm or below.

*3 Mark O in "Tendency" column indicates a factor concerning either corrosion or scale tendency.

b. Heat exchanger

When scale, sludge and microbes are generated in the evaporator and condensor by cooling water, they will be accumulated to drop the efficiency of heat exchange, and increase power consumption per refrigeration ton. Therefore, periodical cleaning is necessary.

c. Air duct

When filters are used for cleaning the air, periodical cleaning is inevitable. Clogging of filters increases pressure loss and reduce air volume to degrade cooling capacity. As air conditioners are quickly contaminated when installed at places under poor atmospheric conditions, cleaning is required at least once a week.

d. Others

It is desirable to reduce air-conditioning load, as possible, by performing reviews of air-conditioning zones through studies on unbalanced supercooling/superheating of rooms, review of air-conditioning levels at corridors, etc. in each season.
Also, in installing any air conditioning systems, studies should be made carefully, including appropriateness of installing heat accumulation tanks, appropriateness of using waste heat, selection of the most efficient air conditioning duct system, etc., all of which are realized by placing emphasis on reduction of the running costs.

THE STUDY (AFTER-CARE)
ON THE ENERGY CONSERVATION PROJECT
IN THE KINGDOM OF THAILAND

TEXTBOOK FOR THE ENERGY AUDIT TECHNIQUES WORKSHOP

11. Model Factory Key Sheet

March 1994

Japan International Cooperation Agency (JICA)

The Energy Conservation Center, Japan (ECCJ)

1. No.1 Boiler

Present boiler efficiency

Feed water	
Quantity of feed water	11.1×10^3 kg/h
Feed water temperature	20°C
Enthalpy of feed water	20.03 kcal/kg
Generated steam	
Steam pressure (gauge)	8 kg/cm^{2}g
Dryness of steam	98%
Steam quantity	$11.1 \times 10^3 - 1.1 \times 10^3 = 10.0 \times 10^3$ kg/h
Enthalpy of saturated water	176.67 kcal/kg
Enthalpy of saturated steam	661.95 kcal/kg
Enthalpy of generated steam	$661.95 - (1 - 0.98) \times 485.28 = 652.24$ kcal/kg
Heat of generated steam	$652.24 \times 10 \times 10^3 = 6522.4 \times 10^3$ kcal/h
Blow water	
Blow rate	10%
Quantity of blow water	$11.1 \times 10^3 \times 0.1 = 1.1 \times 10^3$ kg/h
Heat held by blow water	$176.67 \times 1.1 \times 10^3 = 194.3 \times 10^3$ kcal/h
Fuel	
Lower calorific value of A heavy oil	9700 kcal/kg
Consumption of A heavy oil	811.7 kg/h
Input heat	$9700 \times 811.7 = 7874 \times 10^3$ kcal/h
Effective output heat	$6522.4 \times 10^3 - 10.0 \times 10^3 \times 20.03$ $= 6322 \times 10^3$ kcal/h
Boiler efficiency	$(6322 \times 10^3 / 7874 \times 10^3) \times 100 = 80.3\%$

Method of energy conservation

(1) Decreasing the O_2 concentration of exhaust gas

Present O_2 concentration of exhaust gas: 8%

Heat loss by exhaust gas: $L_l = G \cdot c_g (t_g - t_o)$ kcal/kg (fuel) ....(1)

- G: Quantity of exhaust gas (Nm^3/kg)
- c_g: Mean specific heat of exhaust gas 0.33 kcal/Nm$^3 \cdot$°C
- t_g: Exhaust gas temperature (°C)
- t_o: Atmospheric temperature (°C)

Here; $G = G_0 + G_w + (m - 1) A_0$ ....(2)

G_0: Theoretical quantity of dry exhaust gas (Nm^3/kg)

G_w: Quantity of vapor generated by combustion, and quantity of vapor by evaporation of moisture in fuel (Nm^3/kg)

m : Air ratio

A_0: Theoretical quantity of combustion air (Nm^3/kg)

For terms ($G_0 + G_w$) and A_0, approximation formulas based on Boie's formula are available.

In case of liquid fuel

$$A_0 = 12.38 \times \frac{9700}{10000} - 1.36 = 10.65 \ (Nm^3/kg) \quad \text{......(3)}$$

$$G_0 + G_w = 15.75 \times \frac{9700}{10000} - 3.91 = 11.37 \ (Nm^3/kg) \quad \text{......(4)}$$

$$m = \frac{21}{21 - O_2(\%)} = \frac{21}{21 - 8} = 1.62 \quad \text{......(5)}$$

Therefore,

$$G = 11.37 + (1.62 - 1) \times 10.65 = 17.87 \ (Nm^3/kg)$$

Heat loss by exhaust gas: $L_l = 17.87 \times 0.33 \times (300 - 33) = 1574$ (kcal/kg)

The rate of heat loss by exhaust gas to fuel input heat is

$$R_l = \frac{1574}{9700} \times 100 = 16.2\%$$

(Improvement method)

Improve combustion to decrease the O_2 concentration of exhaust gas to 5%

$$m = \frac{21}{21 - 5} = 1.31$$

$$G = 11.37 + (1.31 - 1) \times 10.65 = 14.67 \ (Nm^3/kg)$$

$$L_l = 14.67 \times 0.33 \times (300 - 33) = 1293 \ (kcal/kg)$$

$$R_l = \frac{1293}{9700} \times 100 = 13.3\%$$

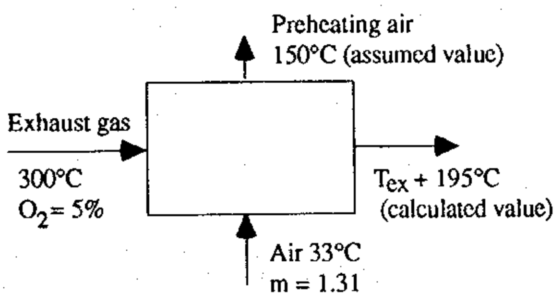
By decreasing the O_2 concentration of exhaust gas from 8% to 5%,

16.2 – 13.3 = 2.9%

It is possible to raise the boiler efficiency by approx. 3%.

(2) Installing the air heater

Normally the rise of boiler efficiency by heat recovery by an air heater is 5%–6%. Suppose the following data are obtaind by installing an air heater;



$$14.67 \times 0.33 \times (300 - T_{ex}) = 10.65 \times 1.31 \times 0.31 \times (150 - 33)$$

$$T_{ex} = 195°C$$

The calorific value of heat collected by air heater is

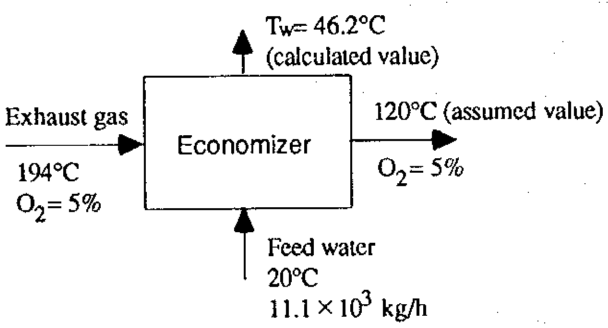
$10.65 \times 1.31 \times 0.31 \times (150 - 33) = 506.0$ kcal/kg (fuel)

The energy conservation efficiency by air heater is

$$\frac{506}{9700} \times 100 = 5.2\%$$

(3) Installing the economizer

Normally the rise of boiler efficiency by heat collection of an economizer is 5% more or less. Suppose the following data are obtaind by installing an economizer;



$$11.1 \times 10^3 \times 1.0 \times (T_w - 20) = 14.67 \times 811.7 \times 0.33 \times (194 - 120)$$

$$T_w = 46.2°C$$

(Exhaust gas after passing the economizer is assumed to be 120°C as the temperature must be keep at a higher than dew point of acid)

The calorific value of heat collected by economizer is

$11.1 \times 10^3 \times 1.0 \times (46.2 - 20) = 290.8 \times 10^3$ kcal/h

The energy conservation efficiency by economizer is

$$\frac{290.8 \times 10^3}{811.7 \times 9700} \times 100 = 3.7\%$$

From (1), (2) and (3), the boiler efficiency of a case all the above-mentioned energy conservation measures are applied is

	Efficiency (%)	Fuel (kg/h)
Efficiency of boiler	80.3	811.7
Improvement of combustion	83.1	784.6
Air heater	88.1	739.8
Economizer	91.6	710.9

(4) Others

a: Raise the feed water temperature by collecting heat from blow water
Increase of efficiency by approx. 1.5%

b: As the boiler surface temperature 150°C is too high, reduce heat loss from the surface by applying insulator.

c: For fuel, eliminate soot blow by changing it to another with better quality such as gas.

d: The blow ratio 10% is for a once-through boiler and is too high, so drop it to 5% or so, if possible. (dependent on the quality of feed water)

Collection of initial cost

• Air heater

Fuel saving by heat recovery by the air heater is,
supposing A heavy oil is $0.40/kg,

$(784.6 - 739.8) \times 3000 \times 0.4 = 53,760$.-US$/year

Supposing the air heater costs $100,000

$$\frac{10,000}{53,760} = 1.9$$

The initial cost can be collected in 1.9 years.

• Economizer

Fuel saving by heat collection of the economizer is

$(739.8 - 710.9) \times 3000 \times 0.4 = 34,680$.-US$/year

Supposing the economizer costs $150,000

$$\frac{150,000}{34,680} = 4.3$$

The initial cost can be collected in 4.3 years.

	Energy conservation efficiency (%)	Initial cost ($)	Fuel saving ($/year)	Period for initial cost collection (year)
Air heater	5.2	100,000	$53,760.-	1.9
Economizer	3.7	150,000	$34,680.-	4.3

2. No.2 Boiler

Present boiler efficiency

Input heat	$9700 \times 406.2 = 3940 \times 10^3$ kcal/h
Effective output heat	$5.0 \times 10^3 \times 652.24 - 5.0 \times 10^3 \times 20.03 = 3161 \times 10^3$ kcal/h
Boiler efficiency	$(3161 \times 10^3 / 3940 \times 10^3) \times 100 = 80.2\%$

Method of energy conservation

(1) Decreasing the O_2 concentration of exhaust gas from 8% to 5%

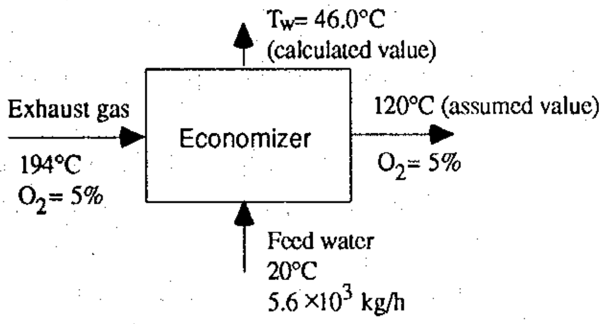
Like the case of the No.1 boiler, the boiler efficiency is raised by 2.9%.

(2) Installing the air heater

Installing an air heater of the same type as that installed for the No.1 boiler, the boiler efficiency is raised by 5.0%.

(3) Installing the economizer

Installing an economizer of the same type as that installed for the No.1 boiler, the boiler efficiency is raised by 3.6%.



From (1), (2) and (3), the efficiency of the energy conservation boiler is

	Efficiency (%)	Fuel (kg/h)
Efficiency of boiler	80.2	406.2
Improvement of combustion	83.0	392.6
Air heater	88.0	370.2
Economizer	91.6	355.7

3. Steam Heater

Improvement

(1) It is important to reduce as much as possible the thermal load of the car, used to take the product to be submitted to heating into and out from the steam heater, comparatively with the product to be submitted to heating.
In other words, such measures as reducing as much as possible the weight of the car.

(2) By making the temperature distribution inside the steam heater even, the volume of the heater can be utilized, and the quantity of the product submitted to heating can be increased.
In order to make the temperature distribution even, the heater must be designed in such a way to attain an uniform flow of the air circulating inside it.

(3) Since the heater is a batch type one, the product must be taken into and out from the heater as quickly as possible, thereby minimizing the temperature drop inside the heater, and minimizing the thermal load peak when starting up the heater.

4. Steamer

Improvement

(1) Since the steamer carries out direct heating of the steam, the peak of the thermal load is large (4.3 t/h) when starting up, thereby exerting decisive influence on the steam generation rate of the boiler. Such being the case, it is important not to start up the 5 steamers at the same time.
After starting up a steamer, it is necessary to confirm that the peak load has returned to the steady load before starting up the next steamer. Thus, it is necessary to prepare in advance the operation program of the steamers.
Both energy saving and safety assurance can be realized during the boiler operation by notifying the work program to the boiler operation group and the number of boilers in operation is controlled in a systematic way.

(2) The temperature inside the steamer can be kept uniform by carrying out the elimination of air for sure. When the temperature distribution inside the steamer is kept even, the volume of the steamer can be utilized, and the quantity of the product submitted to heating can be increased.
The elimination of air can be carried out for sure by installing an O_2 meter or any other kind of detector at the outlet of the steam drain exhaust port. Even when there is no detector, elimination of air can be carried out for sure by measuring in advance the temperature distribution inside the steamer during the start up, and determining the time for exhausting the air.

(3) The thermal load of the tray car must be made as small as possible.

5. Heating Furnace

5.1 Present Condition

5.1.1 Heat balance (Input)

(1) Fuel : $9{,}700 \times 72 = 698{,}400$ kcal/t ($28.8\ \text{kg/h} \times \frac{1}{0.4\ \text{t/h}} = 72\ \text{kg/t}$)

(2) Scale : 7 kg/t × 0.775 (Total Fe in Scale) × 1.335 kcal/kgFe = 7,055 kcal/t

Total : 705,455 kcal/t

5.1.2 Heat balance (Output)

(1) Bar : $\{1{,}000 - (7 \times 0.775)\} \times \Delta h_{950} = 994.7 \times \{150 - (-2.0)\}$
$= 151{,}194$ kcal/t

(2) Scale : $7 \times 0.215 \times (950 - 33) = 1{,}380$ kcal/t

(3) Waste Gas : $G = G_0 + (m - 1) A_0 = 11.37 + (1.6 - 1) \times 10.65 = 17.76\ \text{m}^3/\text{kg}$
$17.76 \times 72 \times 0.33\ (1000 - 33) = 408{,}052$ kcal/t

(4) Wall loss :

$$Q_1 = \frac{2\pi\,(950 - 100)}{\dfrac{\ln\frac{1{,}030}{800}}{1.4} + \dfrac{\ln\frac{1{,}260}{1{,}030}}{0.28} + \dfrac{\ln\frac{1{,}320}{1{,}260}}{0.072}} \times 8\ \text{m} \times \frac{1}{0.4} = 69{,}011\ \text{kcal/t}$$

$$Q_2 = \frac{\pi}{4} \times (0.8)^2 \times 2 \times 940 \times \frac{1}{0.4} = 2{,}361\ \text{kcal/t}$$

$$Q = Q_1 + Q_2 = 69{,}038 + 2{,}361 = 71{,}399\ \text{kcal/t}$$

(5) Cooling Water : ΔT 10°C, $Q = 1\ \text{m}^3/\text{h}$
$1{,}000\ \text{kg/h} \times 1\ \text{kcal/kg·°C} \times 10\text{°C} \times \frac{1}{0.4} = 25{,}000$ kcal/t

(6) Other loss : 48,430 kcal/t

Total : 705,455 kcal/t

5.1.3 Heat Balance Table

Heat Input	kcal/t	%
(1) Calorific value of fuel	698,400	99.0
(2) Formation heat of scale	7,055	1.0
Total	705,455	100.0

Heat Output	kcal/t	%
(3) Heat capacity of extracted steel	151,194	21.4
(4) Sensible heat of scale	1,380	0.2
(5) Heat loss due to waste gas	408,052	57.8
(6) Heat release from wall	71,399	10.1
(7) Heat loss due to cooling water	25,000	3.5
(8) Other heat loss	48,430	7.0
Total	705,455	100.0

Efficiency η_1 = 21.7 %

5.2 Improvement

5.2.1 Improvement of combustion to reduce the air ratio from 1.6 to 1.2

(1) Existing

$$G = 11.37 + (1.6 - 1) \times 10.65 = 17.76\ Nm^3/kg\ fuel$$

$$L_1 = 17.76 \times 0.33 \times (1{,}000 - 33) = 5{,}667\ kcal/kg$$

(2) Improvement

$$G = 11.37 + (1.2 - 1) \times 10.65 = 13.50\ Nm^3/kg\ fuel$$

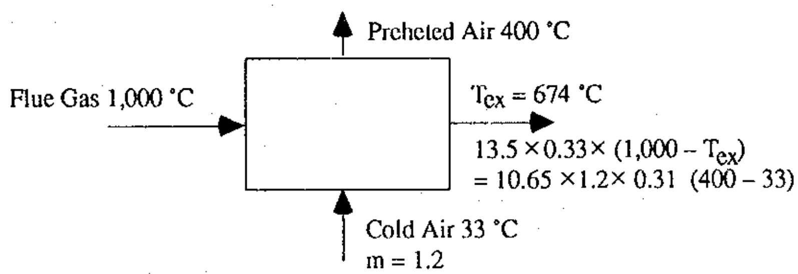
$$L_1 = 13.5 \times 0.33 \times (1{,}000 - 33) = 4{,}308\ kcal/kg$$

(3) Fuel conservation rate by improvement of air ratio

$$\frac{(5{,}667 - 4{,}308)}{(9{,}700 - 4{,}308)} \times 100\% = 25.2\%$$

5.2.2 Installation of recuperator

(1) Suppose the following data are obtained by installation of recuperator



(2) Fuel conservation rate by using recuperator

$$P = mA_0 \times c_p \times t$$
$$= 1.2 \times 10.65 \times 0.32 \times 400 = 1{,}636\ kcal/kg$$

$$S = \frac{1{,}636}{(9{,}700 + 1{,}636 - 4{,}308)} \times 100\% = 23.3\%$$

6. Blower for Dust Collector

From equation of four lines from the bottom on page 67,

Equation (4)
$\eta_t = 0.95$ from Table 17
$\phi = 1.3$ from Table 18

$$L_m = L \times \phi \frac{1}{\eta_t} \text{ (kW)}$$

$$= \frac{\phi}{\eta_F \cdot \eta_t} \times \frac{Q \cdot PT}{6120}$$

$$= \frac{1.3}{0.7 \cdot 0.95} \times \frac{800 \cdot 150}{6120} = 38.3 \text{ (kW)}$$

Real power of Motor L_R (exclude allowance rate)

$$L_R = \frac{1}{0.7 \cdot 0.95} \times \frac{800 \cdot 150}{6120} = 29.5 \text{ (kW)}$$

The ratio of real power to motor rating $= \frac{29.5}{75} = 0.39$

Assumption: (standard motor) motor efficiency $\fallingdotseq$ 0.88

So motor input L_i

$$L_i = 29.5 \times \frac{1}{0.88} = 33.5 \text{ (kW)}$$

(1) Change of motor from 75 kW to 37 kW (standard type)

Assumption: motor efficiency = 0.92
After changing motor, motor input L_i' is

$$L_i' = 29.5 \times \frac{1}{0.92} = 32.0 \text{ (kW)}$$

$\therefore$ Power conservation = 33.5 – 32.0 = 1.5 (kW)

(2) Damper control → Inverter control

1) Existing state

- Motor input at 800 m^3/min = 33.5 (kW)
- Motor input at 300 m^3/min

Ratio to max. flow $= \frac{300}{800} = 0.375$

From power consumption curve of discharge damper control

motor input $\fallingdotseq \frac{0.74}{1.05} \times 33.5 = 23.6$ (kW)

0.74: from Fig. 33, Page 77

2) After improvement

- Motor input at 800 m^3/min = 32.0 (kW)
- Motor input at 300 m^3/min

From power consumption curve of inverter control,

motor input $\fallingdotseq \frac{0.07}{1.05} \times 32.0 = 2.1$ (kW)

0.07: from Fig. 33, Page 77

∴ Power conservation at 800 m^3/min

= 33.5 – 32.0 = 1.5 (kW)

Power conservation at 300 m^3/min

= 23.6 – 2.1 = 22.5 (kW)

7. Pump

(1) Existing state

Real motor input (from equation (3), P.82 $\alpha = 0$)

$$= \frac{0.163 \cdot 1 \cdot 4 \cdot 20}{0.76} \times \frac{1}{0.92}$$

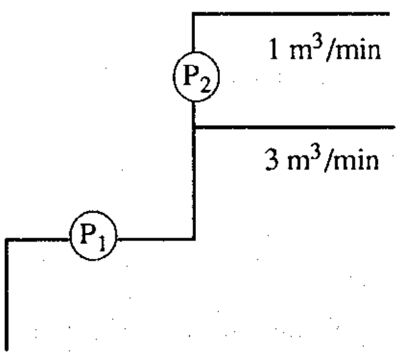
= 18.6 (kW)

Motor efficiency = 0.92

0.76: from Fig. 37, P.82

(2) After improvement -------- (2-motor)

- Establish a booster pump (P_2)
- Change the existing pump to new pump (P_1)



- Real motor input of (P_1)

$$= \frac{0.163 \cdot 1 \cdot 4 \cdot 10}{0.76} \times \frac{1}{0.92} = 9.3 \text{ (kW)}$$

- Real motor input of (P_2)

$$= \frac{0.163 \cdot 1 \cdot 1 \cdot 10}{0.7} \times \frac{1}{0.92} = 2.5 \text{(kW)}$$

Total motor input = 9.3 + 2.5 = 11.8 (kW)

∴ Power conservation = 18.6 – 11.8 = 6.8 (kW)

8. 9. 10. 11. Compressors

1) Adjustment of Run Unit

If the working quantity of compressors is known beforehand, the number of run units of compressors is adjusted according to the necessary air quantity. To decide compressors to be run, the following procedure is used.

(1) Compressors are first classified into the screw type and the reciprocation type, and then arranged in order from ones with larger capacity.

(2) Until a total value of compressors exceeds the necessary flow rate of air, run compressors are decided in order of the screw type and the reciprocation type.

(3) Of compressors, one unit will be run intermittently --- a compressor which will make the off period shortest as possible should be selected.

By deciding run compressors for each hour following the above-mentioned procedure, and by determining the electric power consumed by each compressor, the consumed wattage of the whole can be calculated.

Incidentally, the consumed electric power of a compressor run intermittently can be calculated from the rate of load hours and idling hours.

By setting the following for the compressor run intermittently;

Flow rate:	Q_{max}	m^3/min
Electric power during load hour:	W_l	kW
Electric power during idling hours:	W_u	kW

and by setting an air quantity Q_r to be supplied by this compressor, the load hours t_l and the idling hours t_u are determined as follows.

$$t_l = \frac{Q_r}{Q_{max}} \qquad t_u = \frac{Q_{max} - Q_r}{Q_{max}}$$

Therefore, the consumed electric power during the intermittent hours is

$$W = W_l \frac{Q_r}{Q_{max}} + W_u \frac{Q_{max} - Q_r}{Q_{max}}$$

$$= \frac{Q_r}{Q_{max}} (W_l - W_u) + W_u$$

The result of these calculations are shown in Table 1 and 2.

If the number of run unit is not adjusted, all compressors are run from 8:00 through to 17:00. In this case, compressors run though not required are run under the idling condition, so the electric power used for the idling is a loss. The electric power loss calculated in this way is shown in the bottom stage of Table 1 and 2.

Table 1 Compressor Operation Schedule (Case 1)

time	0 ~ 8	8 ~ 9	9 ~ 12	12 ~ 13	13 ~ 14	14 ~ 16	16 ~ 17	17 ~ 24	
required flow rate	2.5 m³/min	11.0 m³/min	7.0 m³/min	2.5 m³/min	9.0 m³/min	6.0 m³/min	8.0 m³/min	2.5 m³/min	
SCREW TYPE No. 8		continuous	continuous		continuous	continuous	continuous		
Flow rate 4.0 m³/min Load 30 kW Unload 15 kW		4.0 m³/min 30 kW	4.0 m³/min 30 kW	(15 kW)	4.0 m³/min 30 kW	4.0 m³/min 30 kW	4.0 m³/min 30 kW		
SCREW TYPE No. 9		continuous	intermittent		continuous		continuous		
Flow rate 3.2 m³/min Load 25 kW Unload 13 kW		3.2 m³/min 25 kW	3.0 m³/min 24.3 kW	(13 kW)	3.2 m³/min 25 kW	(13 kW)	3.2 m³/min 25 kW		
RECIPRO TYPE No. 10	intermittent	continuous		intermittent		intermittent		intermittent	
Flow rate 3.0 m³/min Load 25 kW Unload 6.0 kW	2.5 m³/min 21.8 kW	3.0 m³/min 25 kW	(6 kW)	2.5 m³/min 21.8 kW	(6 kW)	2.0 m³/min 19.1 kW	(6 kW)	2.5 m³/min 21.8 kW	
RECIPRO TYPE No. 11		intermittent			continuous		intermittent		
Flow rate 1.8 m³/min Load 18 kW Unload 4.0 kW		0.8 m³/min 10.2 kW	(4 kW)	(4 kW)	1.8 m³/min 18 kW	(4 kW)	0.8 m³/min 10.2 kW		
Average Power	21.8 kW	90.2 kW	54.3 kW	21.8 kW	73.0 kW	49.1 kW	65.2 kW	21.8 kW	Total
Power Consumption	174.4 kWh	90.2 kWh	162.9 kWh	21.8 kWh	73.0 kWh	98.2 kWh	65.2 kWh	152.6 kWh	838.3 kWh
Loss of unload	−	(0 kWh)	(30.0 kWh)	(32.0 kWh)	(6.0 kWh)	(34.0 kWh)	(6.0 kWh)	−	(108.0 kWh)

Table 2 Compressor Operation Schedule (Case 2)

time	0 ~ 8	8 ~ 9	9 ~ 12	12 ~ 13	13 ~ 14	14 ~ 16	16 ~ 17	17 ~ 24	
required flow rate	2.5 m³/min	11.0 m³/min	7.0 m³/min	2.5 m³/min	9.0 m³/min	6.0 m³/min	8.0 m³/min	2.5 m³/min	
SCREW TYPE No. 8		continuous	continuous		continuous				
Flow rate 4.0 m³/min Load 30 kW Unload 15 kW		4.0 m³/min 30 kW	4.0 m³/min 30 kW	(15 kW)	4.0 m³/min 30 kW	(15 kW)	(15 kW)		
SCREW TYPE No. 9	intermittent	intermittent		intermittent	intermittent	intermittent	continuous	intermittent	
Flow rate 3.2 m³/min Load 25 kW Unload 13 kW	2.5 m³/min 22.4 kW	2.2 m³/min 21.3 kW	(13 kW)	2.5 m³/min 22.4 kW	2.0 m³/min 20.5 kW	3.0 m³/min 24.3 kW	3.2 m³/min 25 kW	2.5 m³/min 22.4 kW	
RECIPRO TYPE No. 10		continuous	continuous		continuous	continuous	continuous		
Flow rate 3.0 m³/min Load 25 kW Unload 6.0 kW		3.0 m³/min 25 kW	3.0 m³/min 25 kW	(6 kW)	3.0 m³/min 25 kW	3.0 m³/min 25 kW	3.0 m³/min 25 kW		
RECIPRO TYPE No. 11		continuous					continuous		
Flow rate 1.8 m³/min Load 18 kW Unload 4.0 kW		1.8 m³/min 18 kW	(4 kW)	(4 kW)	(4 kW)	(4 kW)	1.8 m³/min 18 kW		
Average Power	22.4 kW	94.3 kW	55.0 kW	22.4 kW	75.5 kW	49.3 kW	68.0 kW	22.4 kW	Total
Power Consumption	179.2 kWh	94.3 kWh	165.0 kWh	22.4 kWh	75.5 kWh	98.6 kWh	68.0 kWh	156.8 kWh	859.8 kWh
Loss of unload	—	(0 kWh)	(51.0 kWh)	(25.0 kWh)	(4.0 kWh)	(38.0 kWh)	(15.0 kWh)	—	(133.0 kWh)

2) Reduction of Discharge Pressure

Power necessary for compressing the air varies by the ratios of pressure at the inlet and outlet of a compressor. As the intake pressure is nearly equal to the atmospheric pressure, the necessary power is reduced by decreasing the discharge pressure. This relationship is shown in Fig. 1 (addition).

The maximum pressure of the working side is 5 kg/cm^{2}g. Therefore, supposing that the outlet pressure of the compressor is decreased from 7 kg/cm^{2}g (8 kg/cm^{2}a) to 6 kg/cm^{2}g (7 kg/cm^{2}a), with a pressure loss in the piping taken into account, the necessary power is, from Fig. 1 , reduced from 4.5 kW/m^3 to 4.15 kW/m^3, and the consumed electric power is reduced by 7.8%.

3) Temperature Reduction of Intake Air

Power necessary for compressing the air is proportionate to the volume of intake air. As the volume of air is proportionate to the absolute temperature, a necessary power for compressing certain volume of air is increased when the temperature of the intake air rises.

When the present temperature of intake air 55°C is dropped to 33°C by taking the atmosphere as is,

$$\frac{33 + 273}{55 + 273} = 0.933$$

namely, the consumed electric power is reduced by 6.7%.

Power necessary for compressing also varies by moisture, besides temperature. When the intake air is cooled down with water, together with removal of moisture content, a large effect can be attained. This relationship is shown in Fig. 2 (addition).

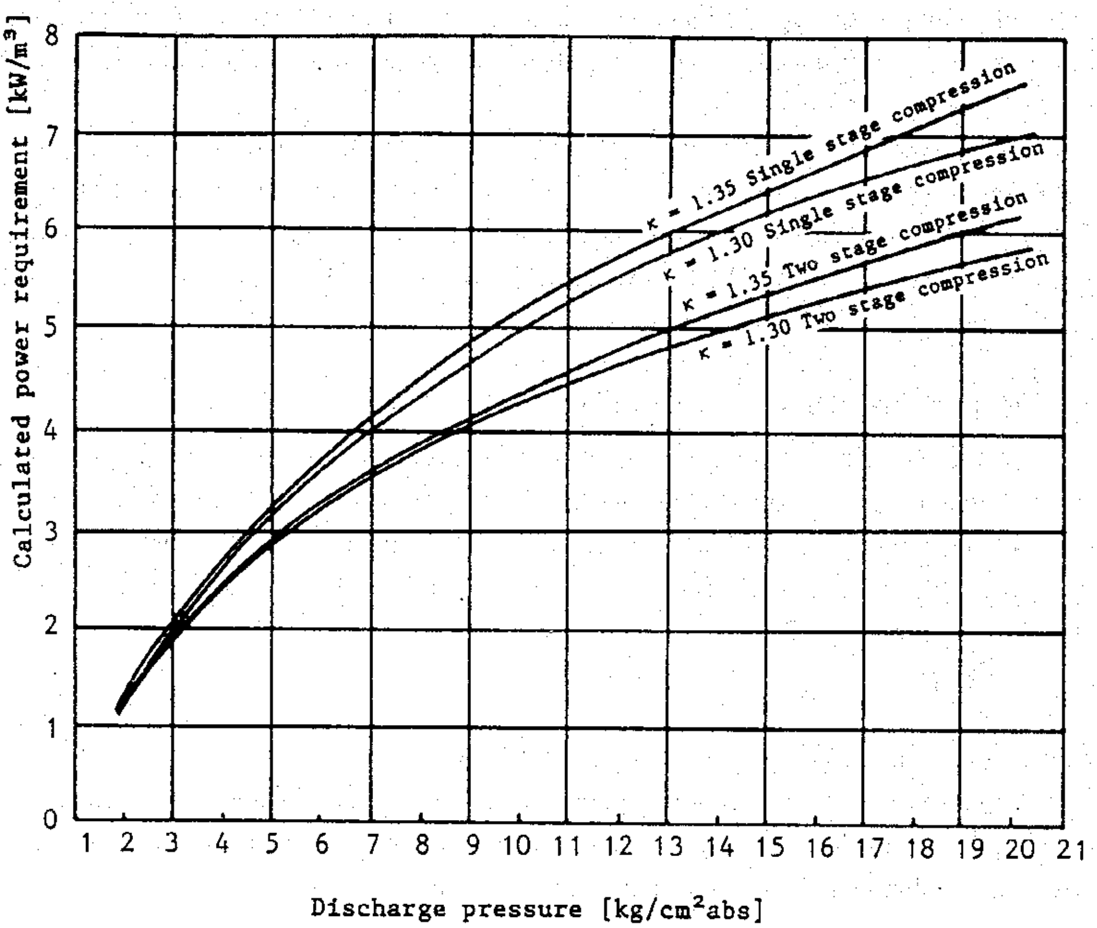


Fig. 1 Required power of air compressor

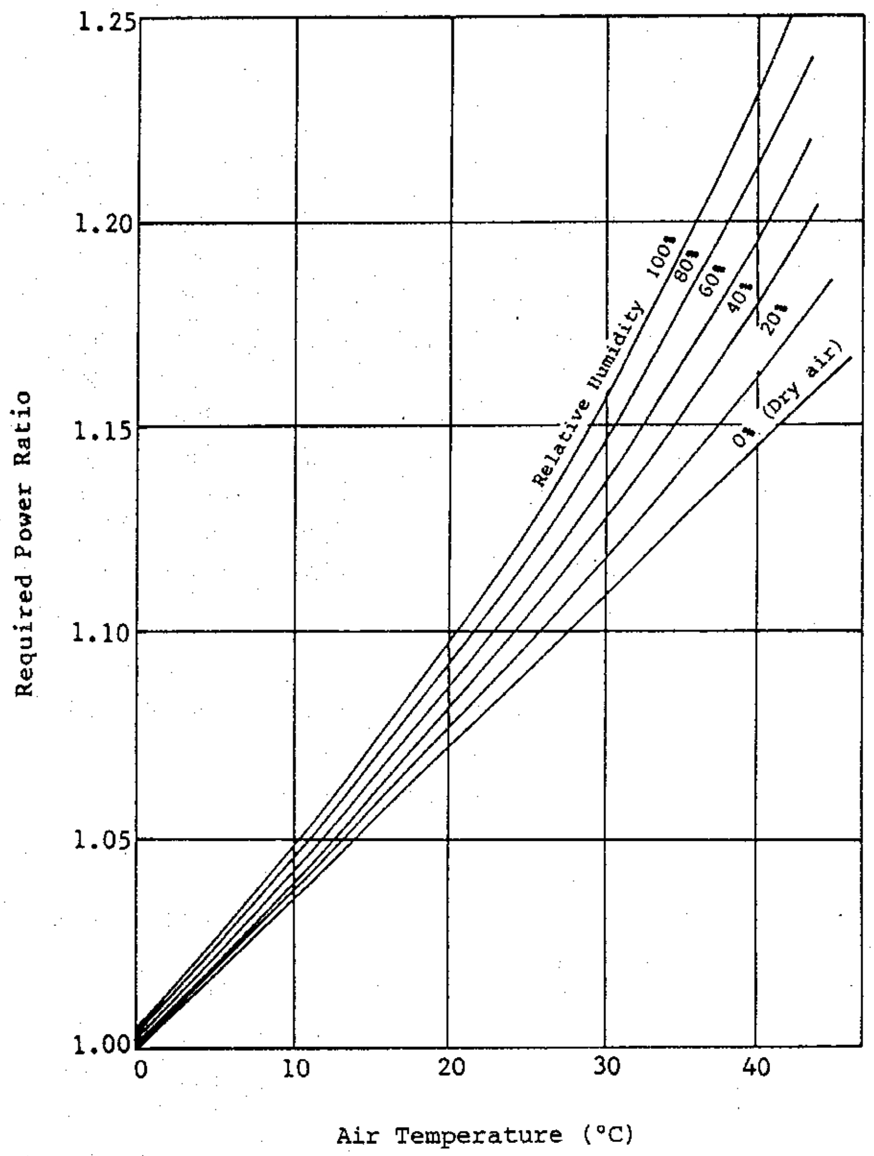


Fig. 2 Effect of humidity on compressor power requirement

12. 13. Transformer

Newly added power conservation is ignored.
(if necessary, you can add the data.)

(1) Existing state

	hour	0–7	7–8	8–12	12–13	13–18	18–19	19–24
Tr (A)	kW	300	300	800	400	800	300	300
	kVA	361	361	941	471	941	361	361
	kVar	200	200	495	249	495	200	200
Tr (B)	kW	200	600	600	300	600	600	200
	kVA	225	698	698	341	698	698	225
	kVar	103	357	357	162	357	357	103
Total at receiving P.F.	kW	500	900	1400	700	1400	900	500
	kVA	520	984	1562	744	1562	984	520
	kVar	143	397	692	251	692	397	143
	P.F.	0.96	0.91	0.90	0.94	0.90	0.91	0.96

(2) After improvement

P.F. at light load on 2ry of Tr (A) and Tr (B) is deregulated to almost 100%.

Condenser of 2ry of Tr (A) 100 kVar → 300 kVar
Condenser of 2ry of Tr (B) 100 kVar → 200 kVar

<Result>

	hour	0–7	7–8	8–12	12–13	13–18	18–19	19–24
Tr (A)	kW	300	300	800	400	800	300	300
	kVA	300	300	853	403	853	300	300
	kVar	0	0	295	49	295	0	0
Tr (B)	kW	200	600	600	300	600	600	200
	kVA	200	653	653	306	653	653	200
	kVar	3	257	257	62	257	257	3
Total Tr (A) + Tr (B)	kW	500	900	1400	700	1400	900	500
	kVA	500	936	1500	709	1500	936	500
	kVar	3	257	552	111	552	257	3
Total at receiving P.F.	kW	500	900	1400	700	1400	900	500
	kVA	500	905	1450	709	1450	905	500
	kVar	3	97	392	111	392	97	3
	P.F.	1.00	0.99	0.97	0.99	0.97	0.99	1.00

1) Exam. of transformer unification

(a) At Max. load

(i) Separate

$$\text{Loss of Tr (A)} = 4.5 + 16.5\left(\frac{853}{1500}\right)^2 = 9.8 \text{ (kW)}$$

$$\text{Loss of Tr (B)} = 2.5 + 12.5\left(\frac{653}{1000}\right)^2 = 7.8 \text{ (kW)}$$

Total loss = 9.8 + 7.8 = 17.6 (kW)

(ii) Unification (Use of Tr (A))

$$\text{Loss of Tr (A)} = 4.5 + 16.5\left(\frac{1500}{1500}\right)^2 = 21.0 \text{ (kW)}$$

From (i), (ii) separate operation is better.

(b) At load 900 kW

(i) Separate

$$\text{Loss of Tr (A)} = 4.5 + 16.5\left(\frac{300}{1500}\right)^2 = 5.2 \text{ (kW)}$$

$$\text{Loss of Tr (B)} = 2.5 + 12.5\left(\frac{653}{1000}\right)^2 = 7.8 \text{ (kW)}$$

Total loss = 5.2 + 7.8 = 13.6 (kW)

(ii) Unification (Use of Tr (A))

$$\text{Loss of Tr (A)} = 4.5 + 16.5\left(\frac{936}{1500}\right)^2 = 10.9 \text{ (kW)}$$

(iii) Unification (Use of Tr (B))

$$\text{Loss of Tr (B)} = 2.5 + 12.5\left(\frac{936}{1000}\right)^2 = 13.5 \text{ (kW)}$$

From (i), (ii), (iii) unification (use of Tr (A)) is better.

(c) At min. load

(i) Separate

$$\text{Loss of Tr (A)} = 4.5 + 16.5\left(\frac{300}{1500}\right)^2 = 5.2 \text{ (kW)}$$

$$\text{Loss of Tr (B)} = 2.5 + 12.5\left(\frac{200}{1000}\right)^2 = 3.0 \text{ (kW)}$$

Total loss = 5.2 + 3.0 = 8.2 (kW)

(ii) Unification (Use of Tr (A))

$$\text{Loss of Tr (A)} = 4.5 + 16.5\left(\frac{500}{1500}\right)^2 = 6.3\ (\text{kW})$$

(iii) Unification (Use of Tr (B))

$$\text{Loss of Tr (B)} = 2.5 + 12.5\left(\frac{500}{1000}\right)^2 = 5.6\ (\text{kW})$$

From (i), (ii),(iii) unification (use of Tr (B)) is better.

14. Air Conditioner

(1) Existing state

Unit (kcal/h)

	Q_{SH}		Q_{LH}	
$q_W = K \cdot A \cdot \Delta t_e =$	24,000			
$q_G = K \cdot A \cdot \Delta t_0 =$	3,080			
$q_I = 0.28 \cdot V_1 \cdot \Delta t_0 =$	4,200	$715 \cdot V_i \cdot \Delta x_0 =$	14,800	Δx_0 (0.0225 (33°C, 70%) / <u>0.0087</u> (23°C, 50%)
$q_L =$	6,000			0.0138
$q_{AS} =$	30,000			
q_d = 15% of RL =	10,092		2,250	
Total	77,372		17,020	

$$SHF = \frac{77{,}372}{77{,}372 + 17{,}020} = 0.82$$

Total supplied air (m^3/h)

$$V = \frac{77{,}372}{1.2 \times 0.24 \times (23 - 15)} = 33{,}581\ m^3/h$$

From wet air diagram, cooling coil load q_C is

$q_C = 1.2 \times 33{,}581 \times (12.9 - 8.5)$

$= 177{,}308$ (kcal/h)

* 12.9 kcal/kgDA: 20% mix air 25 °C, 58%

* 8.5 kcal/kgDA: Cooling air 15 °C, 77%

Driving power P (kW)

$$P = \frac{177{,}308}{(860 \times 3.0)} = 68.7\ (\text{kW})$$

(2) After improvement (Room temperature: 23°C → 28°C)

Unit (kcal/h)

		Q_{SH}	Q_{LH}
q_W	=	12,000	
q_G	=	1,540	
q_I	=	2,100	$715 \times V_i \times \Delta x_0 = 11{,}350$ $\Delta x_0 \left(\begin{array}{l} 0.0225\ (33°C, 70\%) \\ 0.0119\ (28°C, 50\%) \end{array} \right.$
q_L	=	6,000	
q_{AS}	=	30,000	
$q_{d=15\%\ of\ RL}$	=	7,746	1,702
Total		59,386	13,052

$$SHF = \frac{59{,}386}{59{,}386 + 13{,}052} = 0.82$$

Total supplied air (m^3/h)

$$V = \frac{59{,}386}{1.2 \times 0.24 \times (28 - 16)} = 17{,}183\ m^3/h$$

From wet air diagram,

$$q_C = 1.2 \times 17{,}183 \times (15.4 - 10.6)$$
$$= 98{,}974\ (kcal/h)$$

$$P = \frac{98{,}974}{(860 \times 3.0)} = 38.4\ (kW)$$

* 16°C: Change from 15°C
  Because of humidity 100% at 15°C cooling air.

* 15.4 kcal/kgDA: 20% mix air 29°C, 55%

* 10.6 kcal/kgDA: Cooling air 16°C, 95%

(3) Power conservation

Refrigeration driving power = 68.7 – 38.4 = 30.3 (kW)

Others: Circulating fan = 20 kW $\times \frac{17{,}183}{33{,}581}$ = 10.2 (kW)

Cooling water pump

Fresh air fan

15. Electric Charge

	Voltage kV	Demand charge Baht	Energy charge Baht	p.f charge Baht	total Baht
(i)	Below 12	59250	137500	375	197125
(ii)	Below 12	118500	275000	0	393500
(iii)	12-24	105000	267500	0	372500
(iv)	12-24	157500	250000	0	425000
(v)	12-24	252000	353100	0	605100

Regarding cases (ii) and (iii) in the above table, the unit electricity charge (overall electricity charge/electricity consumption in kWh) is cheaper in (iii) than in (ii). This demonstrates the fact that electricity charge decreases as receiving voltage increases.

16. Lighting

(1) Existing state

$$\text{Room index} = \frac{50 \times 80}{7.7 \times (50 + 80)} = 4$$

From utilization factor table

Utilization factor U = 0.69

assume maintenance factor M to be 0.8

$$E = \frac{150 \times 60 \times 400 \times 0.69 \times 0.8}{80 \times 50} \doteqdot 500\ (\text{lx})$$

Power consumption = $400 \times 150 \times 10^{-3} = 60$ (kW)

(2) After improvement (Hg lamp → High pressure Na lamp)

From Table 29 (P.101), efficiency of high pressure Na lamp is 120 lm/W (about twice of Hg lamp).

By using 200W Na lamp, we can get same luminance as Hg lamp.

(3) Power consumption

Half of Hg lamp = 30 kW

(4) Others

Improvement of reflectivity of inner wall floor and ceiling.

17. Heat Insulation

1) No. 1 Boiler

1.1 Present Condition

50 mm unspecified insulation is used to as the hot insulation material, but considering that the surface temperature is still high and is 150°C, the hot insulation cannot be said sufficient.

(Calculation of the quantity of dissipation heat)

If the surface temperature is known, the quantity of dissipation heat per unit area can be determined by equation 3-9 and -12 of Textbook. The quantity of dissipation heat per unit area, when the ambient air temperature is 33°C, emissivity is 0.85, is

From the vertical wall:

$$Q_r = 4.88 \times 0.85 \left(\left(\frac{273 + 150}{100}\right)^4 - \left(\frac{273 + 33}{100}\right)^4 \right)$$

$$= 964 \text{ kcal/m}^2\text{h}$$

$$Q_c = 2.2\,(150 - 33)^{.25} \times (150 - 33) = 847 \text{ kcal/m}^2\text{h}$$

$$Q = Q_r + Q_c = 1811 \text{ kcal/m}^2\text{h}$$

From the cylindrical wall:

$$Q_r = 4.88 \times 0.85 \left(\left(\frac{273 + 150}{100}\right)^4 - \left(\frac{273 + 33}{100}\right)^4 \right)$$

$$= 964 \text{ kcal/m}^2\text{h}$$

$$Q_c = 2.1\,(\,(150 - 33)/2.9\,)^{.25} \times (150 - 33) = 620 \text{ kcal/m}^2\text{h}$$

$$Q = Q_r + Q_c = 1584 \text{ kcal/m}^2\text{h}$$

Surface area is

Vertical wall: $2.9^2 \times 3.14 / 4 \times 2 = 13.2 \text{ m}^2$

Cyrindrical wall: $2.9 \times 3.14 \times 6 = 54.7 \text{ m}^2$

The quantity of dissipation heat is $1811 \times 13.2 + 1584 \times 54.7 = 110{,}550$ kcal/h.

As the annual operting hours are 3000 h/y, the heat loss is

$110{,}550 \times 3000 = 3.3165 \times 10^8$ kcal/y

The thermal conductivity of unspecified insulation is

$$Q_a = \frac{\lambda}{L}(\theta_1 - \theta_0) \quad \lambda = Q_a \frac{L}{(\theta_r - \theta_0)} = 1811 \times 0.05/(300 - 150) \doteqdot 0.6 \text{ kcal/mh°C}$$

1.2 Improvement

Applying hot insulation to the surface to reduce the quantity of dissipation heat

(Selection of hot insulation material)

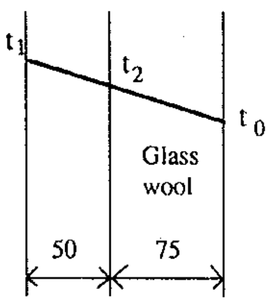
Hot insulation and cold insulation materials stated by JIS are seven types including rock wool, glass wool, cattle hair felt, calcium silicate, polystyrene foam, water repellent pearlite and hard urethane foam. Of these, cattle hair felt, polystyrene foam and hard urethane foam are used mainly for cold insulation, with the maximum temperatures 70°–100°C, so these materials are not usable. Of the remaining four types, calcium silicate and water repellent pearlite are high in the maximum working temperature, but the thermal conductivity is relatively high. Incidentally, ceramic fiber is also available, but it is not used for this sort of low-temperature heat insulation because of its very high cost. Thus, either rock wool or glass wool is most suitable for this purpose.

Physical properties of rock wool and glass wool are very similar, but rock wool is higher in the maximum working temperature and slightly higher in the thermal conductivity. Besides that, for shape as well, bulk fiber, sheet, felt, tubular, belt and blanket are prepared for selection according to the place of use. Moreover, these are classified by density, ones with high density are also high in the maximum working temperature.

As this boiler is 300°C in the maximum working temperature, glass wool is the best suit. For shape, the hot insulation board No.2, 48K is used.

(Calculation of the quantity of dissipation heat)

Using equations 3-1 thru 3-14 of Textbook, the surface temperature and the dissipation heat per unit area are calculated.



The quantity of heat conduction is determined by

$$Q_a = \frac{\lambda_1}{L_1}(t_1 - t_2) = \frac{\lambda_2}{L_2}(t_2 - t_0).$$

As shown in the textbook, the thermal conductivity of glass wool 48K is expressed by cubic equation.

$0.0328 + 8.44 \times 10^{-5}\,\theta + 5.84 \times 10^{-7}\,\theta^2$ W/mk or

$0.0282 + 7.2584 \times 10^{-5}\,\theta + 5.0224 \times 10^{-7}\,\theta^2$ kcal/mh°C

Thickness of the hot insulation material is set to be 75 mm, which is the economical hot insulation thickness as stated in JIS (Table 26). As t_1 = 300°C and t_a = 33°C, $Q_c + Q_r$ is determined by supposing to. From the relationship of $Q_a = Q_c + Q_r$, t_2 and t_0 are calculated as

$$t_2 = t_1 - (Q_c + Q_r)\frac{L_1}{\lambda_1} \qquad t_0 = t_2 - (Q_r + Q_c)\frac{L_2}{\lambda_2}$$

and the calculation is made repeatedly until the determined value is close to the assumed value.

The value of Q_a when the assumed value is almost reached is the quantity of dissipation heat.

In this case the quantity of dissipation heat per unit area is

From the vertical wall: 96.4 kcal/m²h

From the cylindrical wall: 93.8 kcal/m²h

By multiplying this by area

$96.4 \times 14.6 + 93.8 \times 58.9 = 6932$ kcal/h

Heat loss for one year: 2.08×10^7 kcal/y (2.39×10^7 Wh/y)

The heat loss is reduced to 5.4% compared with the condition before improvement.

2) No. 2 Boiler

2.1 Present Condition

50 mm unspecified insulation is used to as the hot insulation material either, the surface temperature is high and is 150°C, and the quantity of dissipation heat per unit area is

From the vertical wall: $Q = Q_r + Q_c$ = 1811 kcal/m²h

From the cylindrical wall: $Q = Q_r + Q_c$ = 1635 kcal/m²h

The surface area is

Vertical wall: $2.1^2 \times 3.14/4 \times 2 = 6.9$ m²

Cylindrical wall: $2.1 \times 3.14 \times 4.8 = 31.7$ m²

The quantity of dissipation heat is

$1811 \times 6.9 + 1635 \times 31.7 = 64{,}325$ kcal/h.

As the annual opeating hours are 3000 h/y, the heat loss is

$64{,}325 \times 3000 = 1.93 \times 10^8$ kcal/y (2.244×10^8 Wh/y)

2.2 Improvement

Like the case of the No. 1 Boiler, the glass wool hot insulation board No.2, 48K is used, and the thickness is set as 75 mm.

Likewise, as the quantity of dissipation heat per unit area after hot insulation is

From the vertical wall: $Q = Q_r + Q_c$ = 96.4 kcal/m²h

From the cylindrical wall: $Q = Q_r + Q_c = 92.9$ kcal/m^{2}h

The surface area is

Vertical wall: $2.25^2 \times 3.14/4 \times 2 = 8.0$ m^2

Cylindrical wall: $2.25 \times 3.14 \times 4.95 = 35.0$ m^2

The quantity of dissipation heat is

$96.4 \times 8.0 + 92.9 \times 35.0 = 4023$ kcal/h.

For one year it is 1.2×10^7 kcal/h (1.4×10^7 Wh/y).

3) Steam Heater

3.1 Present Condition

15 mm glass wool is used to as the hot insulation material, but considering that the recirculated air temperature is high and is 150°C, the hot insulation cannot be said sufficient.

(Calculation of the quantity of dissipation heat)

Using equation 3-9 thru 3-12 of Textbook or Fig. 6, the quantity of dissipation heat per unit area, when the ambient air temperature is 33°C, emissivity is 0.85, is

From the ceiling: 214 kcal/m^{2}h

From the vertical wall: 210 kcal/m^{2}h

There is a heat loss from the bottom as well. However, compared with the ceiling and wall where heat transfer is radiation + convection, heat transfer is only by conduction because the bottom is in contact with the ground and the value is small, so this is considered negligible.

The surface area is

Ceiling: $5.015 \times 3.03 = 15.2$ m^2

Vertical wall: $5.015 \times 3.015 \times 2 + 3.03 \times 3.015 \times 2 = 48.5$ m^2

The quantity of dissipation heat is

$214 \times 15.2 + 210 \times 48.5 = 13{,}438$ kcal/h.

As the annual operating hours are $4 \times 25 \times 12 = 1200$ h/y, the heat loss is

$13{,}438 \times 1200 = 1.61 \times 10^7$ kcal/h

3.2 Improvement

As the temperature of recirculation air is 145°C±5°C, the surface temperature, when no hot insulation is applied to, is assumed as 150°C.

An economical hot insulation thickness at this temperature is 75 mm in the case of glass wool hot insulation board No.2, 48K, thickness of 15 mm glass wool is increased to 75 mm.

By this the quantity of dissipation heat per unit area is

From the ceiling: 57.2 kcal/m^{2}h

From the vertical wall: 56.9 kcal/m^{2}h

By multiplying this by area

$57.2 \times 16.0 + 56.9 \times 50.6 = 3{,}794$ kcal/h

Heat loss for one year: 4.55×10^6 kcal/y

The heat loss is reduced to 24% compared with the condition before improvement.

4) Steamer

4.1 Present Condition

25 mm glass wool is used to as the hot insulation material, but considering that the steam temperature is high and is 150°C, the hot insulation cannot be said sufficient.

(Calculation of the quantity of dissipation heat)

Using equation 3-9 thru 3-12 of Textbook. The quantity of dissipation heat per unit area, when the ambient air temperature is 33°C, emissivity is 0.85, is

From the vertical wall: 135 kcal/m^{2}h

From the cylindrical wall: 134 kcal/m^{2}h

The surface is

Vertical wall: $1.55^2 \times 3.14/4 \times 2 = 3.8$ m^2

Cylindrical wall: $1.55 \times 3.14 \times 11 = 53.6$ m^2

The quantiry of dissipation heat is $135 \times 3.8 + 134 \times 53.6 = 7{,}695$ kcal/m^{2}h.

As the annual operating hours are $4 \times 2 \times 25 \times 12 = 2400$ h/y, the heat loss is

$7{,}695 \times 2400 = 1.8 \times 10^7$ kcal/h

4.2 Improvement

The surface temperature, when no hot insulation is applied to, is assumed as 150°C. An economical hot insulation thickness for this temperature is 75 mm in the case of glass wool hot insulation board No.2, 48K, the thickness of 25 mm glass wool is increased to 75 mm.

In this case the quantity of dissipation heat per unit area is

From the vertical wall: 58.2 kcal/m^{2}h

From the cylindrical wall: 55.5 kcal/m^{2}h

By multiplying this by area

$58.2 \times 4.3 + 55.5 \times 57 = 3414$ kcal/h (3864W)

Heat loss for one year: 8.2×10^6 kcal/y

The heat loss is reduced to 37.5% compared with the condition before improvement.

5) Heating Furnace

5.1 Present Condition

Of heat loss of industrial heating furnaces, heat losses related to refractory materials composing the furnace wall include one that accumulated within the refractory materials, and one dissipated from the surface of furnace wall through refractory materials.

In the case of a batch furnace (one which is run intermittently) of which the industrial heating furnace is run by 8 h/day, the accumulation heat loss is large and it sometimes present a ratio higher than that of dissipation heat loss.

Besides that, in the case of a continuous furnace, where accumulation heat loss occurs only at the starting of the furnace, the ratio is as very low as negligible.

In the case of this furnace, a continuous furnace, the accumulation heat loss is considered negligible.

(Calculation of the quantity of dissipation heat)

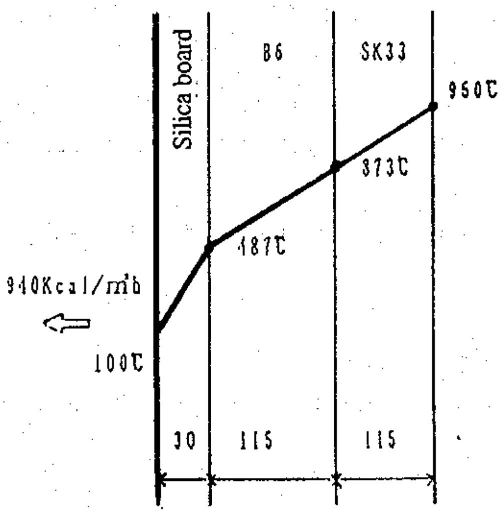
The quantity of dissipation heat is calculated using equations 3-2 thru 3-16 of the textbook. As the furnace is tubular in shape, calculation is made separately for the tubular portion and flat portion. Physical properties necessary for calculation are quoted from the textbook.

Thermal conductivity of fire clay brick:	Fig. 1
Thermal conductivity of insulating brick B6:	Fig. 3
Thermal conductivity of silica board:	Table 27
Thermal conductivity of ceramic fiber:	Fig. 4

Furnace temperature	
(=inner surface temperature of furnace wall):	950°C
Ambient air temperature:	33°C
Emissivity of outer surface:	0.95
Inside diameter of furnace:	800 mm

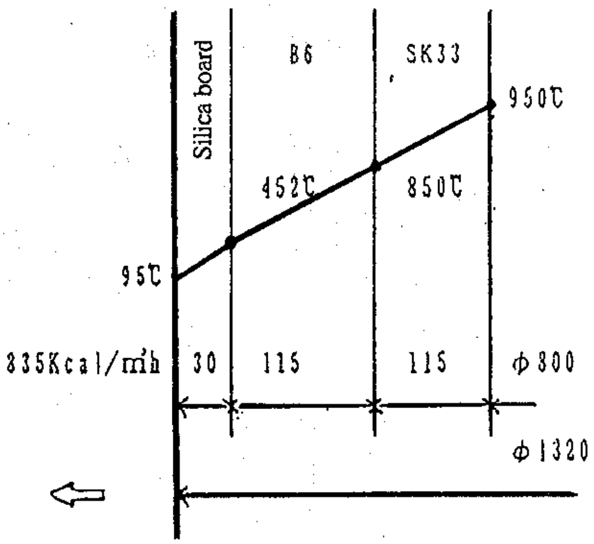
The result of calculation is the following:

(1) Flat wall portion



	Thermal conductivity (kcal/hm°C)
SK33	1.4
B6	0.28
Silica board	0.072

(2) Tubular portion



(3) Total of heat loss from furnace wall

Tubular wall portion	$835 \times (1.32 \times \pi \times 8) = 27690$
Flat wall portion	$940 \times (0.8)^2 \times \pi/4 \times 2 = 945$
Total	28635 kcal/h

5.2 Improvement

To reduce the quantity of dissipation heat from the outer surface of furnace wall, the use of refractory materials with low thermal conductivity is effective. In the case of this furnace, of which the furnace temperature is so high as 950°C, the refractory material used inside the furnace must be one with high heat resistance temperature (950°C and higher). Besides that, as this furnace is an atmosphere furnace, the refractory material is in contact with gas. A refractory material ideal for these conditions is ceramic fiber, and in our country as well ceramic fiber is widely used for furnaces of this type.

For the configuration of furnaces using ceramic fiber, the following methods are typical:

A method which, leaving the existing refractory lining, applies ceramic fiber to the inner face of the furnace, called the "veneering."

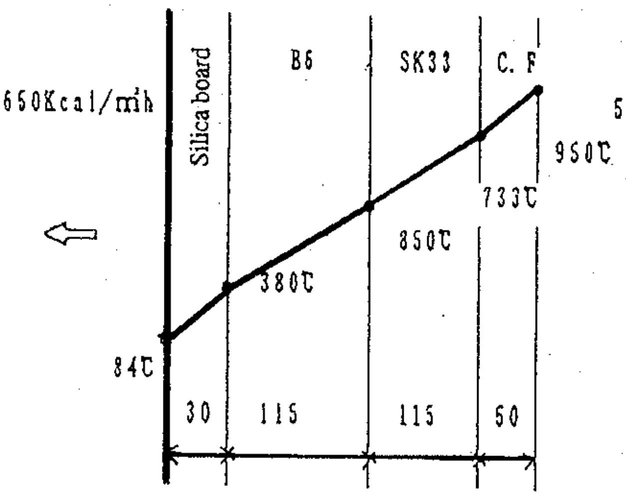
This method is simple in construction and low cost, but there is a limit in the reduction of heat loss.

The reduction effects of heat loss are calculated as follows.

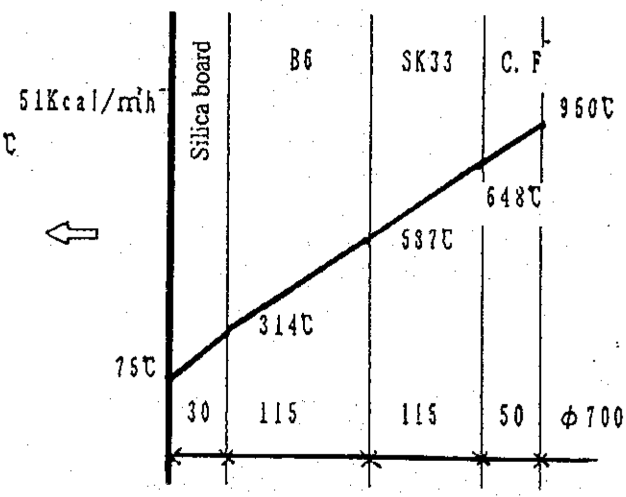
Here, in the "veneering" of the method (1), 50 mm ceramic fiber for 1260°C is applied to the inner face of the furnace wall.

(Calculation of the quantity of dissipation heat)

(1) Flat wall portion



(2) Tubular wall portion



	Thermal conductivity (kcal/hm°C)
Ceramic fiber	0.15
SK33	1.4
B6	0.25
Silica board	0.066

(3) Total of heat loss from furnace wall

Tubular wall portion	$510 \times (1.32 \times \pi \times 8) =$	16910
Flat wall portion	$650 \times (0.8)^2 \times \pi / 4 \times 2 =$	650
Total		17560 kcal/h

6) Steam Piping

6.1 Present Condition

20mm glass wool is applied to as a hot insulation material, but of the pipeline having a total extension of 200 m, a 10 m portion misses the hot insulation material. Though the temperature of steam is not clearly mentioned, supposing a saturation temperature under 8 kg/cm^2G, it is 175°C, and considerable heat loss is expected, since the surface temperature of the pipe without hot insulation is nearly the same. Referring to Table 26, an economical hot insulation thickness for a 12" diameter pipe at this temperature is 65 mm or so, and thus hot insulation cannot be said sufficient with the present 20 mm glass wool.

(Calculation of the quantity of dissipation heat)

A. Portion missing hot insulation

Supposing the surface temperature 175°C, ambient temperature 33°C, the quantity of dissipation heat per unit area is determined using equation.

$\theta = \pi * d * \alpha \, (t_1 - t_a)$, $\alpha = 18$ $kcal/m^2h°C$

2757 $kcal/m^2h$ (3206 W/m^2h)

The 12" pipe length is 10 m.

The quantity of dissipation heat is, $2757 \times 10 = 27{,}570$ kcal/h (32,058 W)

B. Portion with sound hot insulation

As the hot insulation material of the pipe is tubular in shape, the equation 3-15 of the textbook must be used.

Supposing a value of to with $r_0 = 0.16$ m, $r_1 = 0.18$ m, $t_1 = 175$ and $t_a = 33°C$, the quantity of dissipation heat is determined by calculating repeatedly, like the case of plains.

As the result of calculation by computer, $t_0 = 56°C$, and the quantity of dissipation heat 252 $kcal/m^2h$ (293 W/m^2h), are obtained.

The pipe length is $200 - 10 = 190$ m, so the surface area is $0.36 \times \pi \times 190 = 215$ m^2, and the quantity of dissipation heat is $252 \times 215 = 54{,}180$ kcal/h (63,000 W).

6.2 Improvement

The total length is renewed with glass wool with economical hot insulation thickness of 65 mm.

(Calculation of the quantity of dissipation heat)

Supposing r_0 = 0.16 m and r_1 = 0.225 m, the quantity of dissipation heat is determined.
By calculation using a computer, t_o = 41 and the quantity of dissipation heat per unit area = 75 kcal/m^2h, are obtained.
The surface area is, 0.45 × π × 200 = 283 m^2.
The quantity of dissipation heat is, 75 × 283 = 21,225 kcal/h (24,680 W), or reduced to 30% compared with the condition before improvement.
The approximate value of the quantity of dissipation heat can be determined from Table 26, too.

THE STUDY (AFTER-CARE)
ON THE ENERGY CONSERVATION PROJECT
IN THE KINGDOM OF THAILAND

TEXTBOOK FOR THE ENERGY AUDIT TECHNIQUES WORKSHOP

12. Model Building Key Sheet

March 1994

Japan International Cooperation Agency(JICA)

The Energy Conservation Center, Japan(ECCJ)

I. REDUCTIONS IN COOLING LOAD

1. Determination of maximum cooling load

See Table 1 and Fig.1 for the hourly cooling load of the Model Building
Peak load : 2,065 [Mcal/h]
Daily load: 20,946 [Mcal/day]
Refrigerator capacity : 2,065 [Mcal/h] = 683 [USRT]--->700 [USRT]

See Table 2 and Fig.1 for the hourly cooling load of the Energy Conservation Building
Peak load : 1,137 [Mcal/h] (-44.9[%])
Daily load: 10,390 [Mcal/day](-50.4[%])
Refrigerator capacity : 1,137 [kcal/h] = 376 [USRT]--->400 [USRT]

Adopted techniques for energy saving :

(1) insulation of walls & roof	: 25[mm]--------->50[mm]
(2) reflective film on window glass	
(3) high efficiency lighting etc.	: 20[W/m²]------->15[W/m²]
(4) reduction of fresh air	: 30[m³/h/P]---->20[m³/h/P]
(5) fresh air intake control	: CO_2 control
(6) reduction of infiltration	: 0.2[1/h]------>0.1[1/h]
(7) change room air set temperature	: 24[℃],50[%]-->26[℃],50[%]
(8) total heat exchanger	: efficiency = 50[%]

The following is the effect of each energy saving technique.
Only results are shown.

2. Changing temperature and humidity settings

24[℃] 50 [%] -------> 26[℃] 50[%]
Peak load : 1,913 [Mcal/h] (-7.4[%])
Daily load : 19,273 [Mcal/day] (-8.0[%])

3. Changing fresh air intake volume

30 [m³/h/P] ---------> 20 [m³/h/P]
Peak load : 1,802 [Mcal/h] (-12.7[%])
Daily load : 18,132 [Mcal/day] (-13.4[%])

4. Changing the fresh air intake control system

Constant fresh air intake -------> Intake fresh air proportionally to personal occupancy (CO_2 control)
Peak load : 1,907 [Mcal/h] (- 7.7[%])
Daily load : 16,855 [Mcal/day] (-19.5[%])
Daily fresh air load: 8,440 [Mcal/day] (Model Building)
Daily fresh air load: 4,349 [Mcal/day] (CO_2 control) (-48.5[%])

5. Adopting total heat exchanger(efficiency=50[%])

50[%] of sensible heat and latent heat from exhaust air is recovered.

Peak load : 1,670 [Mcal/h] (-19.1[%])

Daily load : 16,726 [Mcal/day] (-20.1[%])

Daily fresh air load: 8,440 [Mcal/day] (Model Building)

Daily fresh air load: 4,220 [Mcal/day] (total heat exchanger) (-50[%])

6. Stopping fresh air intake during precooling

Peak load : 2,065 [Mcal/h] (0[%])

Daily load : 20,221 [Mcal/day] (- 3.5[%])

Daily fresh air load: 8,440 [Mcal/day] (Model Building)

Daily fresh air load: 7,715 [Mcal/day] (- 8.6[%])

7. Minimizing fresh air load

See Table 2 for the hourly cooling load of the Energy Conservation Building

30 [m^3/h/P] ----------------------> 20 [m^3/h/P]

Constant fresh air intake -------> Intake fresh air proportionally to personal occupancy (CO_2 control)

50[%] of sensible heat and latent heat from exhaust air is recovered.

Daily fresh air load: 8,440 [Mcal/day] (Model Building)

Daily fresh air load: 1,284 [Mcal/day] (-84.8[%])

8. Reducing air infiltration

0.2 [1/h]--------------> 0.1 [1/h] (by the improvement of door control, etc.)

Peak load : 1,994 [Mcal/h] (- 3.4[%])

Daily load : 20,186 [Mcal/day] (- 3.6[%])

9. Changing illumination intensity

20 [W/m^2]--------------> 15 [W/m^2] (by high efficiency lighting, etc.)

Peak load : 1,980 [Mcal/h] (- 4.1[%])

Daily load : 20,014 [Mcal/day] (- 4.4[%])

10. Strengthening exterior wall thermal insulation

Insulation : 25 [mm] ------->50 [mm]

K-roof : 0.757---------->0.496[kcal/m^2hdegC]

K-walls : 1.080---------->0.617[kcal/m^2hdegC]

Peak load : 2,024 [Mcal/h] (- 2.0[%])

Daily load : 20,563 [Mcal/day] (- 1.8[%])

11. Strengthening window glass thermal insulation

SC-glass : 0.67 ----------->0.525[ND] (by reflective film)

Peak load : 1,986 [Mcal/h] (- 3.8[%])

Daily load : 20,172 [Mcal/day] (- 3.7[%])

Table 1 Calculation sheet of the cooling load (Model Building)

COOLING LOAD														
TIME			8	9	10	11	12	13	14	15	16	17	18	TOTAL
WALLS	AREA	K-VALUE												
ROOF	1970	0.757	10290	13571	16702	21624	26545	30870	35194	37431	39668	39370	38923	310188
N-WALL	1400	1.08	10584	12852	14969	16027	16934	17086	17086	16330	15574	16330	16934	170705
E-WALL	700	1.08	19127	17842	16556	12550	8543	8543	8543	8089	7636	6728	5746	119902
S-WALL	1400	1.08	10282	16330	22226	24797	27367	26006	24494	19958	15422	13457	11491	211831
W-WALL	700	1.08	5141	6350	7484	8014	8467	13003	17464	19580	21697	18446	15120	140767
TOTAL			55423	66944	77938	83011	87857	95508	102781	101389	99997	94331	88214	953393
WINDOW1	AREA	SC												
N	1191	0.67	31121	33515	38303	41494	42292	41494	38303	33515	31121	82989	94160	508307
E	596	0.67	225616	202455	155335	92642	21164	20765	19167	16771	13976	10382	5990	784264
S	1191	0.67	27929	93362	150816	188321	201088	188321	150816	93362	27929	20747	11970	1154663
W	596	0.67	13976	16771	19167	20765	21164	92642	155335	202455	225616	214834	144554	1127280
TOTAL			298642	346104	363622	343222	285709	343222	363622	346104	298642	328953	256674	3574514
WINDOW2	AREA	K-VALUE												
N	1191	4.9	34432	42018	49022	51940	54857	55441	56025	53690	51356	46104	40268	535152
E	596	4.9	17230	21027	24531	25992	27452	27744	28036	26868	25700	23071	20151	267801
S	1191	4.9	34432	42018	49022	51940	54857	55441	56025	53690	51356	46104	40268	535152
W	596	4.9	17230	21027	24531	25992	27452	27744	28036	26868	25700	23071	20151	267801
TOTAL			103324	126091	147106	155862	164618	166370	168121	161116	154111	138350	120837	1605905
TOTAL-W			401966	472195	510728	499084	450327	509592	531743	507220	452753	467302	377511	5180420
HUMAN-S	54		0	42552	53190	53190	63828	74466	85104	85104	74466	63828	0	595728
LIGHTING	20		338840	338840	338840	338840	338840	338840	338840	338840	338840	338840	338840	3727240
FR.AIR.S	30		100423	122550	142975	151485	159996	161698	163400	156591	149783	134464	117444	1560807
INFIL.-S	0.2		18076	22059	25735	27267	28799	29106	29412	28186	26961	24204	21140	280945
HUMAN-L	48		0	37824	47280	47280	56736	66192	75648	75648	66192	56736	0	529536
FR.AIR.L	30		625396	625396	625396	625396	625396	625396	625396	625396	625396	625396	625396	6879358
INFIL/-L	0.2		112571	112571	112571	112571	112571	112571	112571	112571	112571	112571	112571	1238284
S.TOTAL			914728	1065140	1149406	1152878	1129646	1209209	1251279	1217331	1142800	1122969	943148	12298534
L.TOTAL			737968	775792	785248	785248	794704	804160	813616	813616	804160	794704	737968	8647179
G.TOTAL			1652696	1840931	1934654	1938125	1924350	2013368	2064895	2030946	1946959	1917673	1681115	20945712
Δθe														
ROOF			7	9	11	15	18	21	24	25	27	26	26	
N			7	9	10	11	11	11	11	11	10	11	11	
E			25	24	22	17	11	11	11	11	10	9	8	
S			7	11	15	16	18	17	16	13	10	9	8	
W			7	8	10	11	11	17	23	26	29	24	20	
Δθo			6	7	8	9	9	10	10	9	9	8	7	
I (SOLAR)														
N			39	42	48	52	53	52	48	42	39	104	118	
E			565	507	389	232	53	52	48	42	35	26	15	
S			35	117	189	236	252	236	189	117	35	26	15	
W			35	42	48	52	53	232	389	507	565	538	362	
P(%)			0	40	50	50	60	70	80	80	70	60	0	
FA(%)			100	100	100	100	100	100	100	100	100	100	100	

X-ROOM 0.0092 [kg/kg']
X-OUT 0.024 [kg/kg']
T-24= 0

Table 2 Calculation sheet of the cooling load (Energy Conservation Building))

COOLING LOAD	[kcal/h]													
TIME			8	9	10	11	12	13	14	15	16	17	18	TOTAL
WALLS	AREA	K-VALUE												
ROOF	1970	0.496	4788	6938	8990	12214	15438	18272	21106	22571	24037	23842	23549	181744
N-WALL	1400	0.617	4319	5615	6824	7429	7947	8033	8033	7601	7170	7601	7947	78519
E-WALL	700	0.617	10063	9329	8595	6306	4017	4017	4017	3758	3498	2980	2419	58998
S-WALL	1400	0.617	4146	7601	10970	12439	13907	13130	12266	9675	7083	5960	4837	102015
W-WALL	700	0.617	2073	2764	3412	3714	3973	6565	9113	10322	11532	9675	7774	70918
TOTAL			25390	32247	38791	42101	45283	50017	54535	53927	53320	50058	46526	492194
WINDOW1	AREA	SC												
N	1191	0.525	24386	26262	30013	32514	33140	32514	30013	26262	24386	65029	73782	398300
E	596	0.525	176789	158640	121718	72593	16584	16271	15019	13142	10952	8135	4694	614536
S	1191	0.525	21885	73157	118177	147565	157569	147565	118177	73157	21885	16257	9379	904773
W	596	0.525	10952	13142	15019	16271	16584	72593	121718	158640	176789	168340	113270	883317
TOTAL			234010	271201	284927	268943	223876	268943	284927	271201	234010	257761	201125	2800925
WINDOW2	AREA	K-VALUE												
N	1191	4.9	22760	30347	37350	40268	43186	43769	44353	42018	39684	34432	28596	406762
E	596	4.9	11390	15186	18691	20151	21611	21903	22195	21027	19859	17230	14310	203552
S	1191	4.9	22760	30347	37350	40268	43186	43769	44353	42018	39684	34432	28596	406762
W	596	4.9	11390	15186	18691	20151	21611	21903	22195	21027	19859	17230	14310	203552
TOTAL			68299	91066	112081	120837	129593	131345	133096	126091	119086	103324	85812	1220628
TOTAL-W			302309	362266	397008	389780	353470	400287	418023	397292	353096	361086	286937	4021554
HUMAN-S	54		0	42552	53190	53190	63828	74466	85104	85104	74466	63828	0	595728
LIGHTING	15		254130	254130	254130	254130	254130	254130	254130	254130	254130	254130	254130	2795430
FR.AIR.S	20		0	11801	18156	19574	25191	29786	34495	32680	27006	20085	0	218774
INFIL.-S	0.1		5974	7966	9804	10570	11336	11489	11642	11029	10417	9038	7506	106771
HUMAN-L	48		0	37824	47280	47280	56736	66192	75648	75648	66192	56736	0	529536
FR.AIR.L	20		0	76062	95077	95077	114093	133108	152123	152123	133108	114093	0	1064864
INFIL/-L	0.1		51342	51342	51342	51342	51342	51342	51342	51342	51342	51342	51342	564758
S.TOTAL			587803	710962	771078	769345	753237	820176	857930	834162	772435	758224	595098	8230451
L.TOTAL			51342	165227	193699	193699	222170	250642	279113	279113	250642	222170	51342	2159158
G.TOTAL			639145	876189	964777	963044	975407	1070817	1137043	1113275	1023077	980395	646440	10389609
Δθe														
ROOF			7	9	11	15	18	21	24	25	27	26	26	
N			7	9	10	11	11	11	11	11	10	11	11	
E			25	24	22	17	11	11	11	11	10	9	8	
S			7	11	15	16	18	17	16	13	10	9	8	
W			7	8	10	11	11	17	23	26	29	24	20	
Δθo			6	7	8	9	9	10	10	9	9	8	7	
I (SOLAR)														
N			39	42	48	52	53	52	48	42	39	104	118	
E			565	507	389	232	53	52	48	42	35	26	15	
S			35	117	189	236	252	236	189	117	35	26	15	
W			35	42	48	52	53	232	389	507	565	538	362	
P(%)			0	40	50	50	60	70	80	80	70	60	0	
FA(%)			0	20	25	25	30	35	40	40	35	30	0	
X-ROOM	0.0105	[kg/kg']												
X-OUT	0.024	[kg/kg']												
T-24=	2													

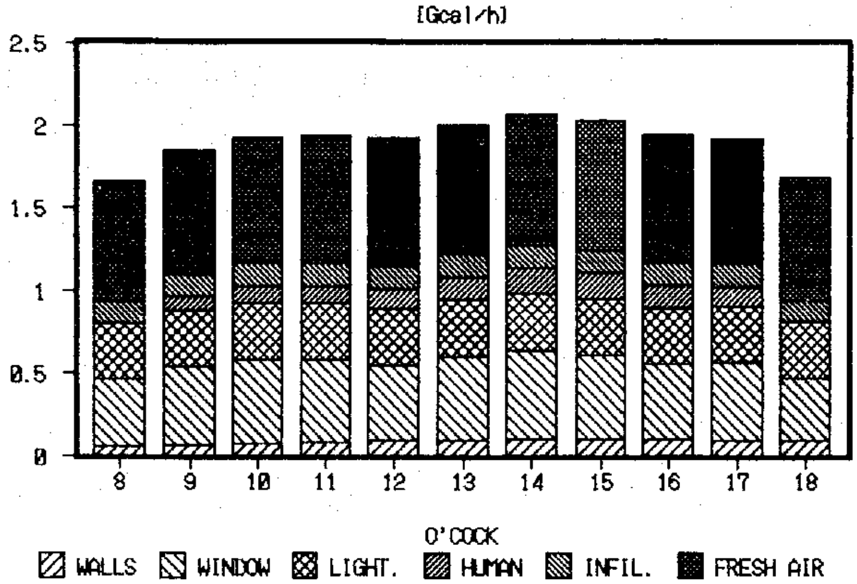


Fig. 1 Hourly cooling load of Model Building

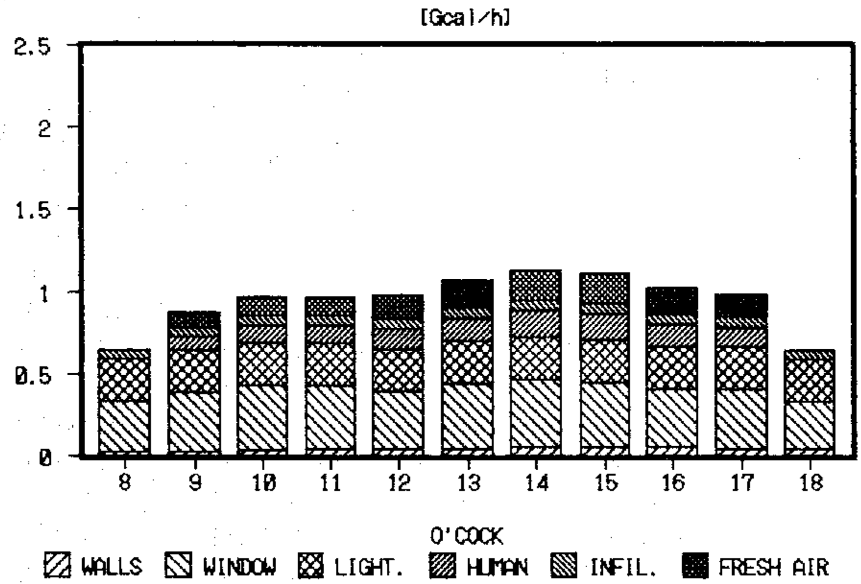


Fig. 2 Hourly cooling load of Energy Conservation Building

II. HEAT SOURCE SYSTEM

1. Energy consumption by refrigerator

(1) Hourly load on a non maximum load operation day

Peak load : 2155 [Mcal/h]

Time	Part.load [%]	Cooling load [Mcal/h]
8- 9	40	862
9-10	40	862
10-11	50	1077
11-12	50	1077
12-13	60	1293
13-14	70	1508
14-15	80	1724
15-16	80	1724
16-17	70	1508
17-18	60	1293
	TOTAL	12928

(2) Electricity consumption

Refrigerator capacity : 2155/3.024=713 --->750 [USRT]=2268[Mcal/h]
Refrigerator power input : 750*3.024/4.5/0.86=586 [kW]

Time	Cooling load [Mcal/h]	Load factor	Number of Operating Machine	Power input relative to standard input	Power input [kW]
8- 9	862	0.38(0.76)	1(1)	0.49(0.76)	287(223)
9-10	862	0.38(0.76)	1(1)	0.49(0.76)	287(223)
10-11	1077	0.48(0.96)	1(1)	0.54(0.95)	316(278)
11-12	1077	0.48(0.96)	1(1)	0.54(0.95)	316(278)
12-13	1293	0.57(0.57)	1(2)	0.61(0.61)	357(357)
13-14	1508	0.67(0.67)	1(2)	0.69(0.69)	404(404)
14-15	1724	0.76(0.76)	1(2)	0.77(0.77)	451(451)
15-16	1724	0.76(0.76)	1(2)	0.77(0.77)	451(451)
16-17	1508	0.67(0.67)	1(2)	0.69(0.69)	404(404)
17-18	1293	0.57(0.57)	1(2)	0.61(0.61)	357(357)
				TOTAL	3630(3426) (-5.6%)

NOTE : Numbers in ( ) shows the data when the refrigerator is divided into 2 machines. (375 [USRT] 2 UNITS)

2. Heat storage system

(1) Hourly load on a maximum load day

Time	Part.load [%]	Cooling load [Mcal/h]
8- 9	60	1293
9-10	60	1293
10-11	70	1508
11-12	80	1724
12-13	80	1724
13-14	90	1939
14-15	100	2155
15-16	100	2155
16-17	90	1939
17-18	80	1724
	TOTAL	17454

(2) Capacity of heat storage tank and refrigerator

Capacity of refrigerator : R [Mcal/h]

$R = Qo\ (1+\alpha)/(\varepsilon T)$

Qo = 17454	: Maximum daily load	[Mcal/day]
α = 0.1	: heat loss of the tank	
ε = 0.9	: load factor of the refrigerator	
T = 10.5	: Operation hours of the refrigerator [h/day] <---- CASE 1 21:30---8:00	

R=2032[Mcal/h]=672[USRT] ----->700[USRT](547[kW])

Capacity of the tank : Vo [m^3]

$Vo = Qs/(\eta\ \Delta\theta)$

Qs=17454	: Cooling load to be stored	[Mcal/day]
η =0.8	: Efficiency of the heat storage tank	
$\Delta\theta$ =5.0	: Temperature difference of water [degC]	

Vo = 4364 [m^3]

(3) Electricity charges for CASE 1,2,3, & 4

Given condition : Hourly load on a non maximum load day (100 %----->2155 [Mcal/h])

Time	Part.load [%]	Cooling load [Mcal/h]
8- 9	40	862
9-10	40	862
10-11	50	1077
11-12	50	1077
12-13	60	1293
13-14	70	1508
14-15	80	1724
15-16	80	1724
16-17	70	1508
17-18	60	1293
	TOTAL	12928

ASSUMPTION : 20 DAYS OF A MONTH HAVE THE COOLING LOAD SHOWN HERE.

THE REST OF THE DAYS HAVE NO COOLING LOAD.

Calculation of the electricity charges per month for CASE 1,2,3 & 4

CASE 1 :

Refrigerator operation : 10.5 [h/ day] (21:30--->8:00)

Refrigerator capacity : R = 17454*1.1/(0.9*10.5) = 2032 [Mcal/h]
= 672 [USRT]----> 700 [USRT](INPUT : 547 [kW])

Refrigerator input : G = 12928*1.1/(4.5*0.86)*20
= 73493 [kWh/Month]

CASE 2 :

Refrigerator operation : 21.0 [h/ day] (21:30--->18:30)

Refrigerator capacity : R = 17454*1.1/(0.9*21.0) = 1016 [Mcal/h]
= 336 [USRT]----> 350 [USRT](INPUT : 274 [kW])

Refrigerator input : G = 12928*1.1/(4.5*0.86)*20
= 73493 [kWh/Month]

CASE 3 :

Refrigerator operation : 21.0 [h/ day] (21:30--->21:30)

Refrigerator capacity : R = 17454*1.1/(0.9*24.0) = 889 [Mcal/h]
= 294 [USRT]----> 300 [USRT](INPUT : 234 [kW])

Refrigerator input : G = 12928*1.1/(4.5*0.86)*20
= 73493 [kWh/Month]

CASE 4 :

Refrigerator operation : Not heat storage system

Refrigerator capacity : R = 2155 [Mcal/h]
= 713 [USRT]----> 750 [USRT](INPUT : 586 [kW])

Refrigerator input : G = 12928/(4.5*0.86)*20
= 66811[kWh/Month]

Electricity charges per month (for refrigerator only)

CASE 1 :

Time period	Demand [kW]	Demand charge [Baht/Month]	Energy [kWh/Month]	Energy charge [Baht/Month]	Total [Baht/Month]
8:00---18:30	0	0	0	0	
18:30---21:30	0	0	0	0	
21:30--- 8:00	547	0	73493	78638	(78638)

CASE 2 :

Time period	Demand [kW]	Demand charge [Baht/Month]	Energy [kWh/Month]	Energy charge [Baht/Month]	Total [Baht/Month]
8:00---18:30	274	17262	0	0	
18:30---21:30	0	0	0	0	
21:30--- 8:00	274	0	73493	78638	(95900)

CASE 3 :

Time period	Demand [kW]	Demand charge [Baht/Month]	Energy [kWh/Month]	Energy charge [Baht/Month]	Total [Baht/Month]
8:00---18:30	234	14742	0	0	
18:30---21:30	0	0	0	0	
21:30--- 8:00	234	0	73493	78638	(93380)

CASE 4 :

Time period	Demand [kW]	Demand charge [Baht/Month]	Energy [kWh/Month]	Energy charge [Baht/Month]	Total [Baht/Month]
8:00---18:30	586	36918	0	0	
18:30---21:30	0	0	0	0	
21:30--- 8:00	0	0	66811	71488	(108406)

III. DISTRIBUTION SYSTEM

1. VAV System

A/C zone : 5th Floor(West zone)

Cooling load : max load at 16:00

Wall	1784	
Window	25090	
Human	1337(S)	1188(L)
Light	4257	
Infilt.	339(S)	1510(L)
TOTAL	32807(S)	2698(L)

Max air flow volume : V [m^3/h]

$V = Qs / (Cp\ \gamma\ \Delta t)$

Qs : Sensible heat load [kcal/h]

Cp : Specific heat of air[kcal/kgdegC]

γ : Specific weight of air[kg/m^3]

Δt: Temp.diff. of supply & return air[degC]

V = 32807/(0.24*1.2*10) = 11391 [m^3/h]

Fan input : Wf [kW]

$Wf = V\ \Delta P / (6120\ \eta)$

V : Air flow volume [m^3/min]

ΔP: Fan pressure [mmAq]

η : Fan efficiency

Wf = 190*30 / (6126*0.6) = 1.55[kW]

Comparison of Fan energy consumption :

TIME	PARTIAL LOAD	FAN INPUT CAV		VAV1		VAV2		VAV3		VAV4	
	[%]	[%]	[kW]	[%]	[kW]	[%]	[kW]	[%]	[kW]	[%]	[kW]
8- 9	40	100	1.55	68	1.05	45	0.70	30	0.47	26	0.40
9-10	40	100	1.55	68	1.05	45	0.70	30	0.47	26	0.40
10-11	50	100	1.55	75	1.16	52	0.81	38	0.59	32	0.50
11-12	50	100	1.55	75	1.16	52	0.81	38	0.59	32	0.50
12-13	60	100	1.55	82	1.27	59	0.91	44	0.68	42	0.65
13-14	70	100	1.55	88	1.36	66	1.02	56	0.87	52	0.81
14-15	80	100	1.55	90	1.40	75	1.16	66	1.02	64	0.99
15-16	80	100	1.55	90	1.40	75	1.16	66	1.02	64	0.99
16-17	70	100	1.55	88	1.36	66	1.02	56	0.87	52	0.81
17-18	60	100	1.55	82	1.27	59	0.91	44	0.68	42	0.65
TOTAL	[kWh/Day]		15.50 (100%)		12.5 (81%)		9.2 (59%)		7.3 (47%)		5.7 (37%)

2. VWV System

Zone : Whole Building

Max cooling load : 2155 [Mcal/h]
Max water flow volume : V [m^3/h]

$V = QT / \Delta t$

QT : Max cooling load of the whole building [Mcal/h]
Δt: Temp.diff. of supply & return air [degC]

$V = 2155/5 = 431$ [m^3/h]
$= 7183$ [l/min]

Pump input : Wp [kW]

$Wp = V H / (6120 \eta)$

V : Water flow volume [l/min]
H : Pump head [mAq]
η : Pump efficiency

Wp = 7183*30 /(6120*0.6)=58.7[kW]

Comparison of pump energy consumption :

TIME	PARTIAL LOAD	PUMP CWV		INPUT VWV1		VWV2		VWV3		VWV4	
	[%]	[%]	[kW]	[%]	[kW]	[%]	[kW]	[%]	[kW]	[%]	[kW]
8- 9	50	100	58.7	80	47.0	50	29.4	60	35.2	32	18.8
9-10	50			80	47.0	50	29.4	60	35.2	32	18.8
10-11	50			80	47.0	50	29.4	60	35.2	32	18.8
11-12	60			84	49.3	84	49.3	63	37.0	42	24.7
12-13	70			88	51.7	88	51.7	88	51.7	52	30.5
13-14	80			92	54.0	92	54.0	92	54.0	66	38.7
14-15	70			88	51.7	88	51.7	88	51.7	52	30.5
15-16	70			88	51.7	88	51.7	88	51.7	52	30.5
16-17	60			84	49.3	84	49.3	63	37.0	42	24.7
17-18	50	100	58.7	80	47.0	50	29.4	60	35.2	32	18.8
TOTAL			587		496		425		424		255 [kWh/day]
			(100%)		(84%)		(72%)		(72%)		(43%)

JICA