The operation of six traps is shown in Table 5.1.7. Two traps out of six traps are normal, but four traps indicate abnormality of some kind. The contents of abnormality are;

① Leakage : Steam is leaking together with the condensate, and loss occurs as a result.

② Block : The condensate for eight cylinders cannot be discharged perfectly, and the cylinder surface temperature cannot be maintained.

Table 5.1.7 Operation of traps (No. 85 machine)

System		Type	Size	Pressure	Judgment
System 1	Cylinder	Thermodynamic	20 mm	0.8 bar(G)	Leak
	Steam feed line	Thermodynamic	20 mm	0.8 bar(G)	Good
System 2	Cylinder	Thermodynamic	20 mm	0.8 bar(G)	Block
	Steam feed line	Thermodynamic	20 mm	0.8 bar(G)	Good
System 3	Cylinder	Thermodynamic	20 mm	0.8 bar(G)	Leak
	Steam feed line	Thermodynamic	20 mm	0.8 bar(G)	Leak

Although the cylinder surface temperature is secured, trap operation is inferior at many points, and steam leakage is observed at a number of points. Furthermore, the steam pressure before trap was 0.8 bar(G) or less due to leakage and pressure loss. It is estimated that if no steam leakage from the pipeline occurs, the steam pressure in the cylinders rises to 1.0 bar(G) or higher, and that the leakage rate from traps will increase. The leakage rate caused by faulty trap operation is estimated as about 40 kg/h.

B) Countermeasures for improvement and their effect

a. Improvement of steam feed method

The steam to No. 85 machine and No. 86 machine is supplied from a low pressure steam line of 1.2 bar(G). The low pressure steam is also used at other points, and the steam feed pressure is fluctuating (1.2 ~ 1.05 bar), and accordingly, it is imagined that the cylinder surface temperature is fluctuating.

The following requirements should be satisfied in order to upkeep the drying efficiency of the cylinder dryer.

- The steam feed pressure should be constant.

- The condensate should be completely separated from the feed steam, and it should not be allowed to make inflow to cylinders.

- The condensate generated in the cylinders should be discharged completely and quickly.

It is recommended that the method for steam feed is improved as described below, in order to satisfy these requirements.

a) To stabilize the steam feed pressure

Steam is supplied to the cylinders from the low pressure line. But change it to the high pressure line, and provide a reducer valve so that the pressure on the secondary side is kept constant, even if the pressure on the primary side varies to a certain extent. (Figure 5.1.8)

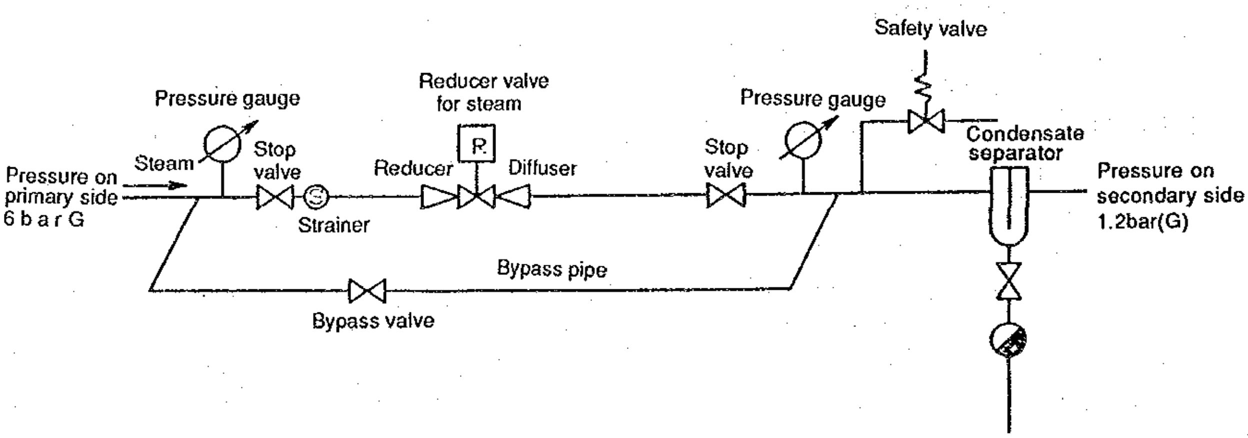


Figure 5.1.8 Improvement of steam feeding method

b) To separate and discharge condensate from the steam feed line

The condensate generated in the steam feed line should not be allowed to make inflow to the cylinders. If the condensate makes inflow into the cylinders, it may form films on the cylinder internal surfaces and may cause reduction of the cylinder surface temperature. Mount a trap and a condensate separator along the steam feed line in order to completely separate and discharge the condensate in the fine water-drop state. (Figure 5.1.8)

c) Effect

The following effect can be anticipated as a result of the countermeasures stated above.

- Rise of cylinder surface temperature
- Increase of drying capacity (increase of cloth speed)
- Improvement of steam unit consumption

The expenses required for the countermeasures stated above are around ¥160,000 (equivalent to about 80,000 Ft).

b. Elimination of steam leakage

Steam is leaking from flexible tubes and pipe joints of the steam feed line for the cylinder dryer. Leakage is observed with 23 cylinder steam feed lines out of 24 cylinders. It is estimated that the leakage per system in total is equivalent to the leakage from a hole of diameter 5 mm.

a) Calculation of steam leaking rate

The rate of steam leakage from a hole of diameter 5 mm can be expressed by equation (1).

$$G = 71.64 \times 10^4 \times A \times \sqrt{(P / v")} \qquad (1)$$

where; G : Steam leaking rate kg/h

A : Area of leaking hole m^2

P : Absolute pressure of steam kg/cm^2 abs

$$P = P_1 \times 1.0197 + 1.033$$

P_1 : Steam pressure in the pipeline bar(G)

v" : Steam specific volume in the pipeline m^3/kg

$$\therefore G = 71.64 \times 10^4 \times 0.785 \times 0.005^2 \times \sqrt{\frac{1.2 \times 1.0197 + 1.033}{0.8054}}$$

$$\fallingdotseq 23.5\ kg/h$$

b) Effect

The steam leaking rate per system calculated by equation (1) is 23.5 kg/h, and the rate of leakage from the entire No. 85 machine is $23.5 \times 3 = 70.5$ kg/h.

The annual saving amount achieved by elimination of leakage can be expressed by equation (2).

$$Q = Gm \times 1/S \times H \times @ \qquad (2)$$

where;	Q :	Annual saving amount	Ft/Y
	Gm :	Rate of leakage from the entire No. 85 machine (69.6 kg/h)	kg/h
	S :	Boiler evaporation multiple (11.39 kg/Nm³)	kg/Nm³-Fuel
	H :	Annual operation time of No. 85 machine (4800 h)	h
	@ :	Boiler fuel unit price (13.5 Ft/Nm³)	Ft/Nm³-Fuel

$$\therefore Q = 70.5 \text{ kg/h} \times 1/11.39 \text{ Nm}^3/\text{kg} \times 4800 \text{ h/y} \times 13.5 \text{ Ft/Nm}^3$$
$$= 29{,}700 \text{ Nm}^3/\text{y} \times 13.5 \text{ Ft/Nm}^3$$
$$= 401{,}000 \text{ Ft/Y}$$

c. Positioning of steam traps and change to steam trap type

a) Positioning of traps

One trap only is provided for eight cylinders of a system. With this arrangement, discharge of the generated condensate may be easy from some cylinders, but it is hard from other cylinders, and such a difference causes a drop in the drying capacity. It is therefore usual to provide a trap for each cylinder.

b) Change to steam trap type

The following points are important in the running of the cylinder dryer.

① To quickly activate running.

② To equalize the cylinder surface temperature.

Discharge of the condensate is particularly important to satisfy two points indicated above, and it is necessary to pay attention to the following points.

① To feed dry steam to cylinders.

To provide a condensate separator along the steam feed line.

② To prevent air locking.

To open the trap bypass valve at the beginning of running.

③ To quickly discharge the generated condensate.

To provide one trap for each cylinder.

④ To select traps of suitable capacity.

To select traps having a discharge capacity that is 5~8 times of the condensate generation rate.

The type of the currently used traps is thermodynamic type. But from the viewpoint stated above, it is recommended that traps of mechanical type (downward bucket type) with air locking eliminators (air vent) which discharge the condensate continuously are adopted.

The contents of implementation of items a) and b) above are shown in Figure 5.1.9.

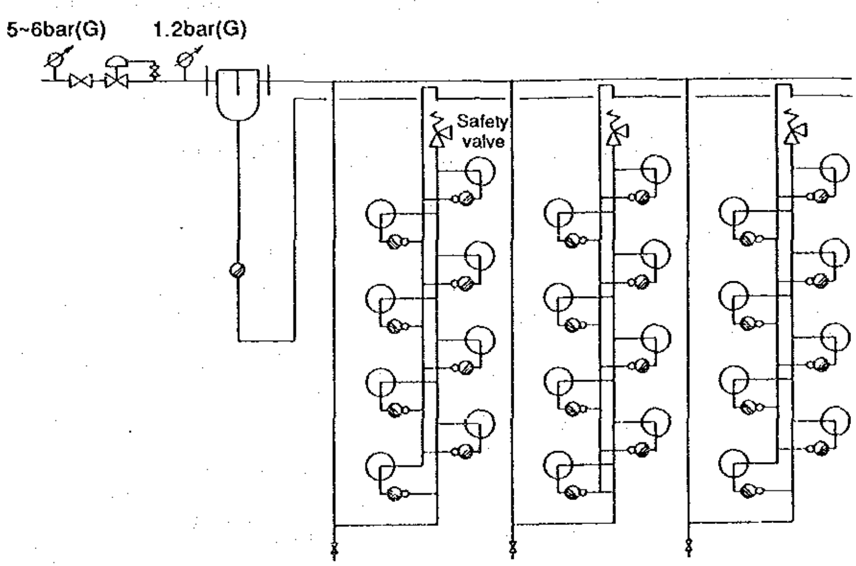


Figure 5.1.9 Positioning of steam traps and change to steam trap type

The expenses required for the countermeasures stated above are considered to be about ¥840,000 (equivalent to about 420,000 Ft) in Japan including purchase of traps and check valves and change of layout of pipelines.

c) Effect

The current rate of leakage of the condensate caused by faulty traps is estimated as 40 kg/h per trap from the results of measurement [A) b. c]. As leakage from three traps is observed, the total rate of leakage is as follows.

$$Gm = 40 \times 3 = 120 \text{ kg/h}$$

The annual saving amount can be expressed by equation (2).

$$\begin{aligned} G &= Gm \times 1/S \times H \times @ \\ &= 120 \text{ kg/h} \times 1/11.39 \text{ Nm}^3\text{/kg} \times 4{,}800 \text{ h/y} \times 13.5 \text{ Ft/Nm}^3 \\ &= 50{,}600 \text{ Nm}^3\text{/y} \times 13.5 \text{ Ft/Nm}^3 \\ &= 683{,}000 \quad \text{Ft/y} \end{aligned}$$

d. Reutilization of the condensate

The condensate from the cylinder dryer is currently discharged to the space underneath the dryer with the bypass valve opened, because the traps are faulty. It is recommended that a condensate recovery tank is provided and the condensate is reutilized for the equipment (such as the washing equipment in the previous process) that use hot water.

a) Condensate recovery system

The condensate recovery flow is shown in Figure 5.1.10.

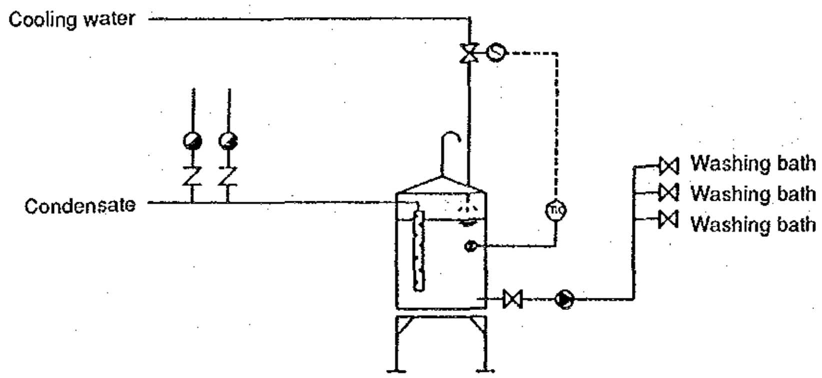


Figure 5.1.10 Condensate recovery flow

① What require attention with the condensate recovery line are as follows.

- The back pressure should not exceed the level of 40 ~ 45% of the steam working pressure.
- The recovery line should be heat insulated.
- Provisions should be made so that the condensate blows into the liquid through many small holes, so that vibration and noise will be hardly produced.
- A cooling line should be provided so that the temperature in the recovery tank will not exceed 90°C.

② The diameter of the recovery line is expressed by equation (3).

$$d = \sqrt{\frac{3.53 \times W \times Ve}{V}} \qquad (3)$$

where;

d : Pipe inside diameter — cm

W : Condensate flow — kg/h

V : Flow velocity in the pipe — m/s
Open recovery: 10 ~ 15 m/s,
Closed recovery: 5 ~ 10 m/s

Ve: Equivalent specific volume

$Ve = v'(1 - f) + v''f$ — m³/kg

v' : Specific volume of saturated water under the pressure in the recovery line — m^3/kg

v'' : Specific volume of saturated steam under the pressure in the recovery line — m^3/kg

f : Re-evaporation rate

$$f = \frac{h_1 - h_2}{r_2}$$

h_1 :	Enthalpy of the condensate on the trap inlet side	kJ/kg
h_2 :	Enthalpy of the condensate under the pressure in the recovery line	kJ/kg
r_2 :	Vaporization latent heat under the pressure in the recovery line	kJ/kg

The expenses required for the countermeasures stated above are considered to be about ¥1,000,000 (equivalent to about 500,000 Ft) in the case of Japan.

Breakdown	Condensate reservoir 5 m^3 (SUS304)	¥300,000
	Temperature controller	¥150,000
	Tank stand (SS)	¥50,000
	Water delivery pump	¥100,000
	Set of pipeline and heat insulation	¥400,000
	Total	¥1,000,000

b) Effect

It is said that the condensate generated per cylinder of a cylinder dryer is 40 kg/h in general. The condensate of $24 \times 40 = 960$ kg/h can be recovered from 24 cylinders. Saving of fuel and water can be achieved by making effective use of this condensate at other places.

The recovered quantity of heat is expressed by equation (4).

$$Q = G \times (t - t_1) \times C \qquad (4)$$

where;

Q :	Recovered quantity of heat	kJ/h
G :	Recovered condensate flow (960 kg/h)	kg/h
t :	Recovered condensate temperature (105°C)	°C
t_1 :	Water temperature (25°C)	°C
C :	Specific heat of water	kJ/kg

$$\therefore Q = 960 \text{ kg/h} \times (105 - 25)^\circ\text{C} \times 4.186 \text{ kJ/(kg}\cdot{}^\circ\text{C)}$$
$$= 341{,}600 \quad \text{kJ/h}$$

The amount of effect through the reutilization of condensate can be calculated by equation (2).

Here, it is assumed that the effective quantity of heat of the steam is 2,257 kJ/kg.

① Reduction of fuel gas consumption

$= 341.600 \text{ kJ/h} \times 4{,}800 \text{ h/y} \times 1/2{,}257 \text{ kg/kJ} \times 1/11.39 \text{ Nm}^3\text{/kg} \times 13.5 \text{ Ft/Nm}^3$
$= 63{,}800 \text{ Nm}^3\text{/y} \times 13.5 \text{ Ft/Nm}^3$
$= 861{,}000 \text{ Ft/y}$

② Reduction of water consumption

$= 960 \text{ kg/h} \times 4{,}800 \text{ h/y} \times 1/1{,}000 \times 19.8 \text{ Ft/t}$
$= 4{,}600 \text{ t/y} \times 19.8 \text{ Ft/t}$
$= 91.000 \text{ Ft/y}$

① + ② = 861,000 + 91,000 = 952,000 Ft/y

Therefore, the equipment expenses for improvement can be recovered in six months.

c. Others

Deterioration of the driving part of the cylinder dryer is excessive, and thorough consolidation is required together with repair to the steam leaking points. When the countermeasures stated above are implemented, the balance in the throughput rate with the washer is improved by the increase of the capacity in addition to the steam loss reduction effect. Besides, the effect brought by increase of production and stabilization of the quality can also be anticipated.

(2) Washer

A) Current situations

a. Situations of equipment

The washer is composed of ten baths, and the cloth flows as water washing → hot water washing → water washing. Every hot water washing bath is equipped with a TIC (Temperature Indicating Controller), and the hot water temperature is automatically controlled by regulating the flow rate of the steam for indirect heating. However, since the heater tubes are damaged at the present time, live steam is directly blown into the baths, and the control mechanism is not functioning.

In the line of washing baths, the liquid makes counterflow with cloth, and water drainage is made in the final bath to constitute an energy saving system. The cloth speed is around 40 m/min with the balance with the next process taken into account. (See Figure 5.1.11)

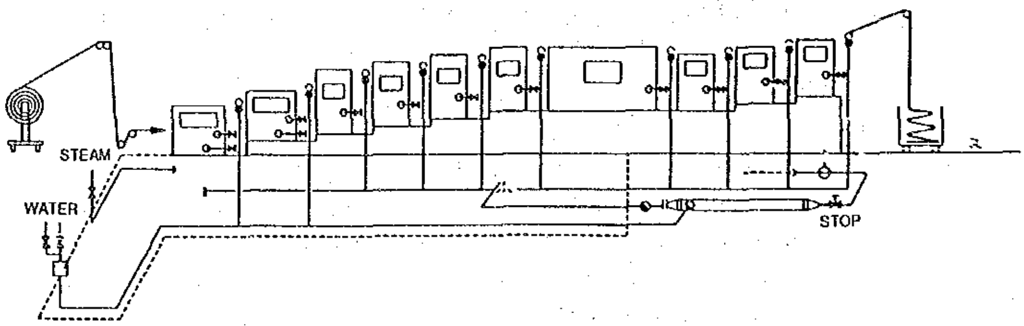


Figure 5.1.11 Washer

The rates of water delivery to the washing baths and mangles are manually controlled by operators, and the rates of water delivery to the washing baths are not stable. The rates of water delivery to No. 1 and No. 2 baths in particular are large. The specification of baths is shown in Table 5.1.8.

Table 5.1.8 Specification of washing baths

Bath		1	2	3	4	5	6	7	8	9	10
Set temperature (°C)		20	20	80	80	80	80	80	60	60	20
Washing water valve		-	Open	-	-	-	Open	Open	-	Open	-
Bath size (mm)	H	1,420	→	1,560	→	→	→	→	1,400	→	→
	L	1,470	→	1,200	→	→	→	2,700	1,200	→	→
	D	2,270	→	→	→	→	→	→	→	→	→
	h	400	→	500	→	→	→	300	400	→	→

b. Situations of running

a) Investigation of water flow rate

The water delivery mains is of 3", but reduction is made to 2-1/2" on the way. The water flow rate of the entire washer and that of the mangle water pipe (0.5") for No. 5 bath were measured using an ultrasonic flowmeter. The flow rate of the entire machine is shown in Figure 5.1.12, and the mangle flow rate is shown in Figure 5.1.13. Furthermore, the rate of use of water by bath was measured by receiving the water discharged from the bottom of the bath in a fixed container. It is converted into the flow rate per hour and is shown in Table 5.1.9.

The leakage from some pipe joints and baths was neglected.

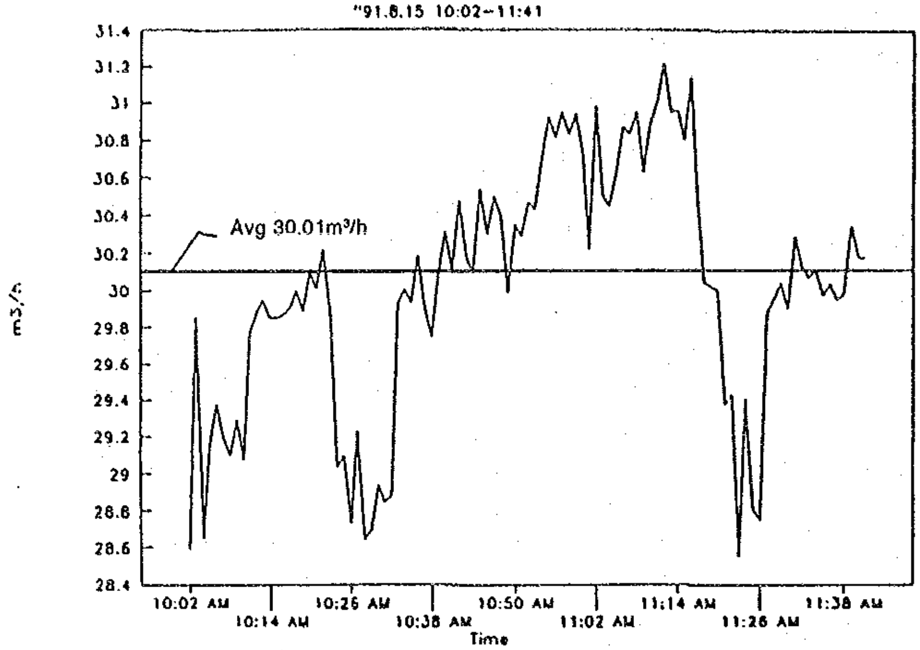


Figure 5.1.12 Water Flow for Washer

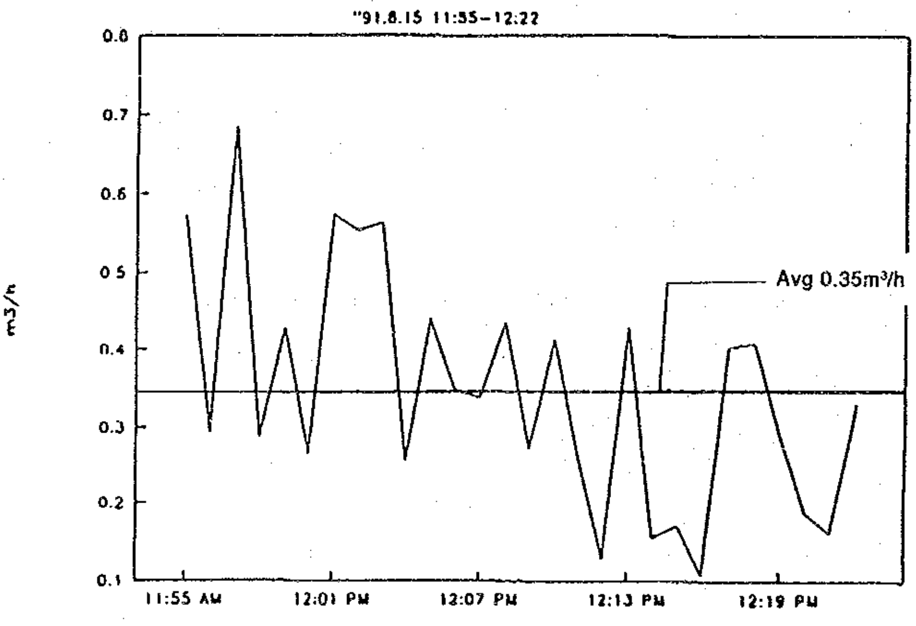


Figure 5.1.13 Water Flow for Mangle

The flow rate is fluctuating in time as shown in Figure 5.1.12 and Figure 5.1.13, and it is estimated that the water delivery pressure is fluctuating. The flow rate for the entire washer was 31.2 m^3/h at maximum, 28.6 m^3/h at minimum, and 30.0 m^3/h at average. The flow rate for the mangle was 0.69 m^3/h at maximum, 0.11 m^3/h at minimum, and 0.35 m^3/h at average. The flow rate is fluctuating.
The drainage from each bath is divided into five draining systems, i.e., counterflow drainage from No. 2 bath to No. 1 bath, counterflow drainage from No. 6 bath to No. 3 bath, drainage to No. 7 bath, drainage to No. 8 bath, and drainage to No. 10 bath.

The waste water discharged out of the bottom of baths makes inflow to the waste water treatment plant through the drainage channel. The hot waste water used in the counterflow from No. 6 bath to No. 3 bath is also abandoned to the waste water treatment plant without being used.

Table 5.1.9 Rate of drainage and set temperature of each bath

Bath		No.1	No.3	No.7	No.8	No.10	Total
Flow	ℓ/h	18,400	3,200	1,550	4,500	2,320	30,000
Temperature	°C	20	80	80	60	20	

b) Surface temperature and heat insulation of washing baths

The surface temperature of each bath was measured using a thermocouple type surface thermometer. The results of measurement are shown in Figure 5.1.14. As for heat insulation, the portions indicated by bold lines, i.e., 20 m of 3" mains and 7 m of 1-1/2" branch line, are not heat insulated. (Figure 5.1.15) A part of the steam feed line is located in the pit, and such a phenomenon that the water in the pit is boiling as heated by the steam line is observed.

The washer main units are not heat insulated at all, and heat radiation is made from surfaces of stainless steel plates.

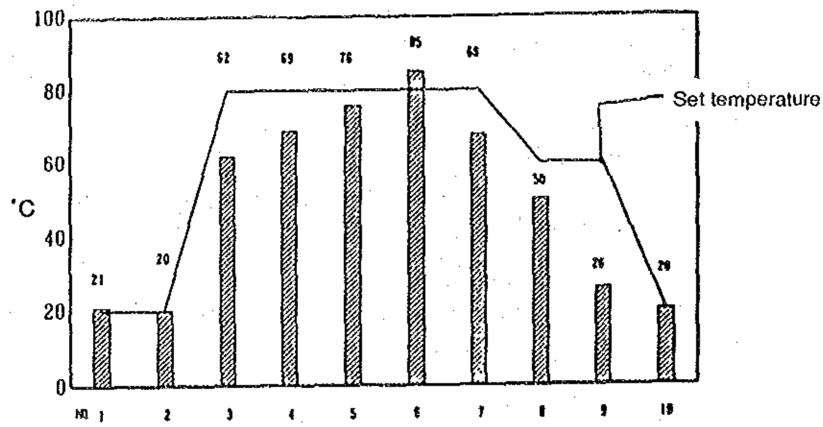


Figure 5.1.14 Bath surface temperature

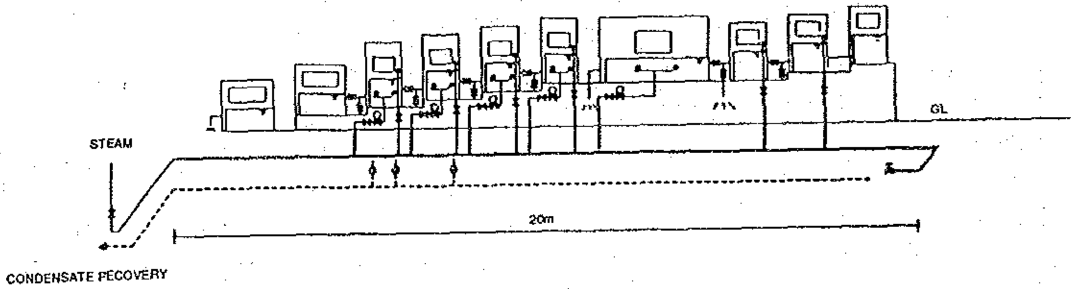


Figure 5.1.15 Heat Insulation

B) Countermeasures for improvement and their effect

a. Saving of washing water

a) To make the flow rate to each bath constant

The flow rates of water fed to baths and mangle are controlled manually by the operators, and the flow rates are varying daily. It is recommended that a reference value of water flow rate is determined by product type and a valve type that facilitates adjustment is adopted.

Furthermore, it is recommended that the following improvement is made in Figure 5.1.16 so that the water feed pressure to the washing baths is kept constant even when the water feed source pressure varies.

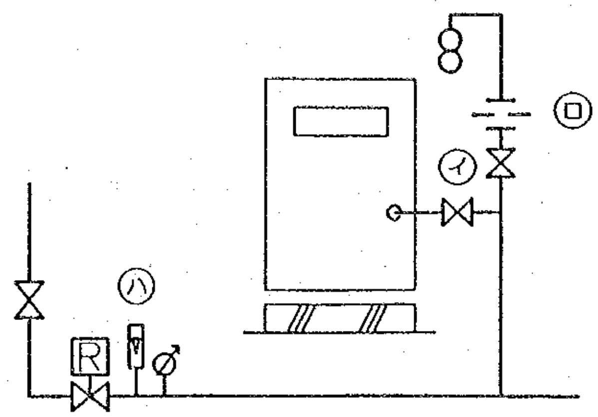


Figure 5.1.16 Washing bath water feed system

a: To change the type of handles of water valves from round type to lever type so that the opening can be set at a fixed level.

b: To provide orifice plates (hole diameter 5 ~ 8 mm) in mangle line to make the flow rate constant (about 0.30 m^3/h).

c: To provide a pressure regulator valve so as to keep the pressure on the secondary side at a fixed level. Furthermore, to provide a flow indicator and a pressure gauge to permit management upon eye observation of these instruments.

The drainage rate of No. 1 bath can be reduced from 18,400 ℓ/h to 5,000 ℓ/h, and the mangle water flow rate can be reduced from 350 ℓ/h to 250 ℓ/h as a result of the countermeasures of items a, b and c above. As viewed from the drainage rate of No. 3 and No. 8 baths, the value of 5,000 ℓ/h is considered feasible for No. 1 bath, and 250 ℓ/h is considered to be sufficient for the mangle as observed from Figure 5.1.13.

b) Equipment investment expenses ¥435,000 (equivalent to 220,000 Ft)

Breakdown

a.	Lever type ball valve	¥10,000 × 12 = ¥120,000
b.	Replacement of orifices	¥ 4,000 × 10 = ¥40,000
c.	Set of pressure regulator valves	¥150,000
	Set of flow cells	¥120,000
	Pressure gauge	¥5,000
	Total	¥435,000

c) Effect

Reduction of flow rate to No. 1 bath	18,400 – 5,000 = 13,400 ℓ/h
Reduction of flow rate for mangle	100 × 10 = 1000 ℓ/h
Total	13,400 + 1,000 = 14,400 ℓ/h

The annual saving amount can be expressed by equation (5).

$$Q = G \times H \times @ \qquad (5)$$

where; Q : Annual saving amount — Ft/y

G : Reduced water flow rate (14.4 m^3/h) — m^3/h

H : Annual operation time of washer (4,800 h) — h

@ : Water unit price (19.8 Ft/m^3) — Ft/m^3

$$\begin{aligned} \therefore Q &= 14.4\ m^3/h \times 4800\ h/y \times 19.8\ Ft/m^3 \\ &= 69{,}100\ m^3/y \times 19.8\ Ft/m^3 \\ &= 1{,}369{,}000\ Ft/y \end{aligned}$$

Besides, reduction of the waste water treating cost can be anticipated.

Therefore, the expenses required for improvement can be recovered within a short period of time.

b. Recovery and reutilization of waste heat

a) Recovery of heat from washing hot waste water

It was told that this is being planned at the factory. But when the hot waste water from No. 3, No. 7 and No. 8 baths is recovered in a reservoir and heat exchange is made with fresh water using a heat exchanger, reduction of the heating steam flow of the hot water baths can be achieved. Since the waste water contains ravelings, it is necessary to remove them using a screen or alike. The flow of recovery of heat is shown in Figure 5.1.17.

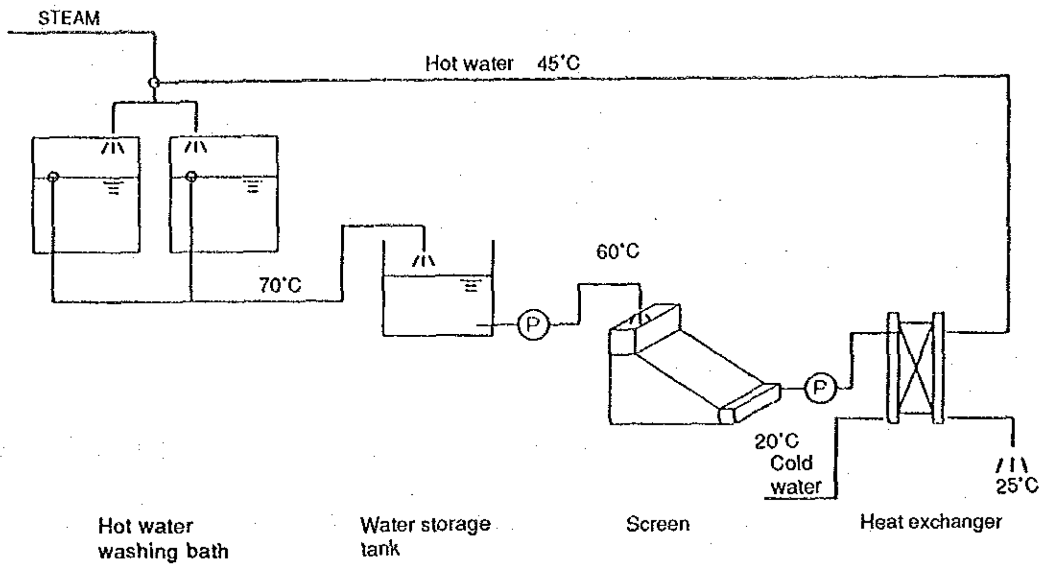


Figure 5.1.17 Flow of recovery of heat from waste water

It is assumed that a plate type heat exchanger is used, and that the temperature and flow rates of the waste water and fresh water are as shown in Table 5.1.10.

Table 5.1.10 Temperature and flow rates of waste water and fresh water

	Temperature at inlet of heat exchanger	Temperature at outlet of heat exchanger	Flow rate
Waste water	60°C	25°C	9.25 t/h
Fresh water	20°C	45°C	9.25 t/h

Heat exchange rate $Q = 9.25 \times (60 - 25) \times 4.186 = 1{,}355$ MJ/h

The required heat transfer area (A) can be calculated by equation (6).

$$A = \frac{Q}{K \cdot \Delta t_m} \qquad (6)$$

where; A : Heat transfer area m^2

Q : Heat exchange rate (1,355 MJ/h) kJ/h

K : Total heat transfer coefficient (6,700 kJ/•m²h•°C) kJ/(m²•h•°C)

Δt_m : Mean temperature difference °C

$$\Delta t_m = \frac{(60-45)-(25-20)}{2.3\log\dfrac{(60-45)}{(25-20)}} = 9.1°C$$

$$\therefore A = \frac{1,355 \times 10^3}{6,700 \times 9.1} = 22\ m^2$$

b) Equipment investment expenses ¥4,200,000 (equivalent to 2,100,000 Ft)

Breakdown	Water storage tank 10 m³ (made of RC)	¥700,000
	Screen	¥400,000
	Heat exchanger	¥2,500,000
	Set of pipeline	¥600,000
	Total	¥4,200,000

c) Effect

The hot waste water drainage rates are as follows from Table 5.1.9.

No. 3 bath = 3,200 ℓ/h (80°C)
No. 7 bath = 1,550 ℓ/h (80°C)
No. 8 bath = 4,500 ℓ/h (60°C)
Total of 3 baths = 9,250 L/h (70°C)

Furthermore, when the hot water temperature is set at 45°C and the hot water flow rate is same as the waste water drainage rate, the recovered quantity of heat is as follows.

$(45 - 20) \times 9,250 \times 4.186 = 968$ MJ/h

The annual saving amount can be expressed by equation (7).

$$Q = M \times H \times 1/h'' \times 1/S \times @ \qquad (7)$$

where; Q : Saved amount Ft/y

M : Recovered quantity of heat (968 MJ/h) MJ/h

H : Operation time per year (4,800 h) h

h":	Vaporization latent heat (2,257 kJ/kg)	kJ/kg
S :	Boiler evaporation multiple (11.39 kg/Nm³)	kg/Nm³
@:	Boiler fuel unit price (13.5 Ft/Nm³)	Ft/Nm³

$$\therefore Q = 968{,}000 \text{ kJ/h} \times 4{,}800 \text{ h/y} \times 1/2{,}257 \text{ kg/kJ} \times 1/11.39 \text{ Nm}^3\text{/kg} \times 13.5 \text{ Ft/Nm}^3$$
$$= 180{,}800 \text{ Nm}^3\text{/y} \times 13.5 \text{ Ft/Nm}^3$$
$$= 2{,}441{,}000 \text{ Ft/y}$$

Therefore, the equipment expenses can be recovered within a year.

c. Execution of heat insulation work

a) Surface heat insulation of washing baths

It is so designed that the hot water temperature in washing baths is 60 ~ 80°C. But heat loss is occurring because the bath surfaces are not heat insulated.

For heat insulation of such equipment that use water, foam plastic in plate form is used, and heat insulation work is executed with caulking applied so that washing water will not enter the heat insulation material.

The area requiring heat insulation is as follows when it is assumed that heat insulation work is executed at the portion lower than the liquid level in each bath.

80°C washing bath	37.6 m^2
60°C washing bath	12.3 m^2

b) Expenses for heat insulation work ¥500,000 (equivalent to 250,000 Ft)

Breakdown	Foam plastics	50 m^2 × ¥4,000/m^2 = ¥200,000
	Working expenses	50 m^2 × ¥5,000/m^2 = ¥300,000
	Total	¥500,000

c) Effect

The results of calculation of heat loss before and after heat insulation are shown in Table 5.1.11. (See the Guideline for the method for calculation.)

Table 5.1.11 Dispersed quantity of heat before and after heat insulation of washing baths

Inner Temp °C	Surface		Present Heat Loss		Heat Loss after Insulation Improved					Saved Energy	Note
	Area m^2	Direction	$kJ/m^2/h$	MJ/h		Thick mm	Surface Temp	$kJ/m^2/h$	MJ/h		
80	20.6	Side	2,507	51.7	Plastics	15	41	402	8.3	43.4	
80	17.0	Bottom	1,842	31.4	Plastics	15	45	362	6.2	25.2	
60	6.9	Side	1,423	9.9	Plastics	8	40	372	2.6	7.3	
60	5.4	Bottom	1,059	5.8	Plastics	8	43	319	1.7	4.0	
	50.0			98.7					18.8	80.0	

Method for calculation : See Guideline.

Preconditions for calculation: 1. The room temperature is 25°C

2. The surfacre emissivity is 0.9 at bare portions and is 0.5 at heat insulated portions.
3. The representative length is 0.5 m

Heat loss is reduced by 80.0 MJ/h as a result of heat insulation.

The annual saving amount achieved by heat insulation can be expressed by equation (7).

$$
\begin{aligned}
Q &= 80{,}000\ \mathrm{kJ/h} \times 4{,}800\ \mathrm{h/y} \times 1/2{,}257\ \mathrm{kg/kJ} \\
&\quad \times 1/11.39\ \mathrm{Nm^3/kg} \times 13.5\ \mathrm{Ft/Nm^3} \\
&= 14{,}900\ \mathrm{Nm^3/y} \times 13.5\ \mathrm{Ft/Nm^3} \\
&= 201{,}200\ \mathrm{Ft/y}
\end{aligned}
$$

Therefore, the expenses required for heat insulation can be recovered in about 1.2 years.

(3) Frame dryer

A) Current situations

a. Situations of equipment

City gas is combusted in the dryer body and the cloth is dried by the heat of the combusted gas. The air required for combustion is fed by two blowers installed on top of the dryer. (Figure 5.1.18)

① Temperature control in the dryer

The temperature in the dryer can be set in the range of 50~250°C, and the combustion rate is controlled with the difference between internal temperature and the set temperature.

② Automatic control of air-fuel ratio (Figure 5.1.19)

The flow of the gas for combustion and the flow of the air for combustion are controlled by automatically opening and closing linked valves of lever type, to prevent incomplete combustion caused by lack of air.

③ Control of exhaust rate (Figure 5.1.20)

A damper is provided on the suction side of the exhaust fans, and its opening can be remote controlled between 0% and 100%. When the opening is reduced to 40% or less, however, interlock is applied and combustion of the burner is stopped by the safety system. Besides, functions such as cloth speed control and dryness check are provided.

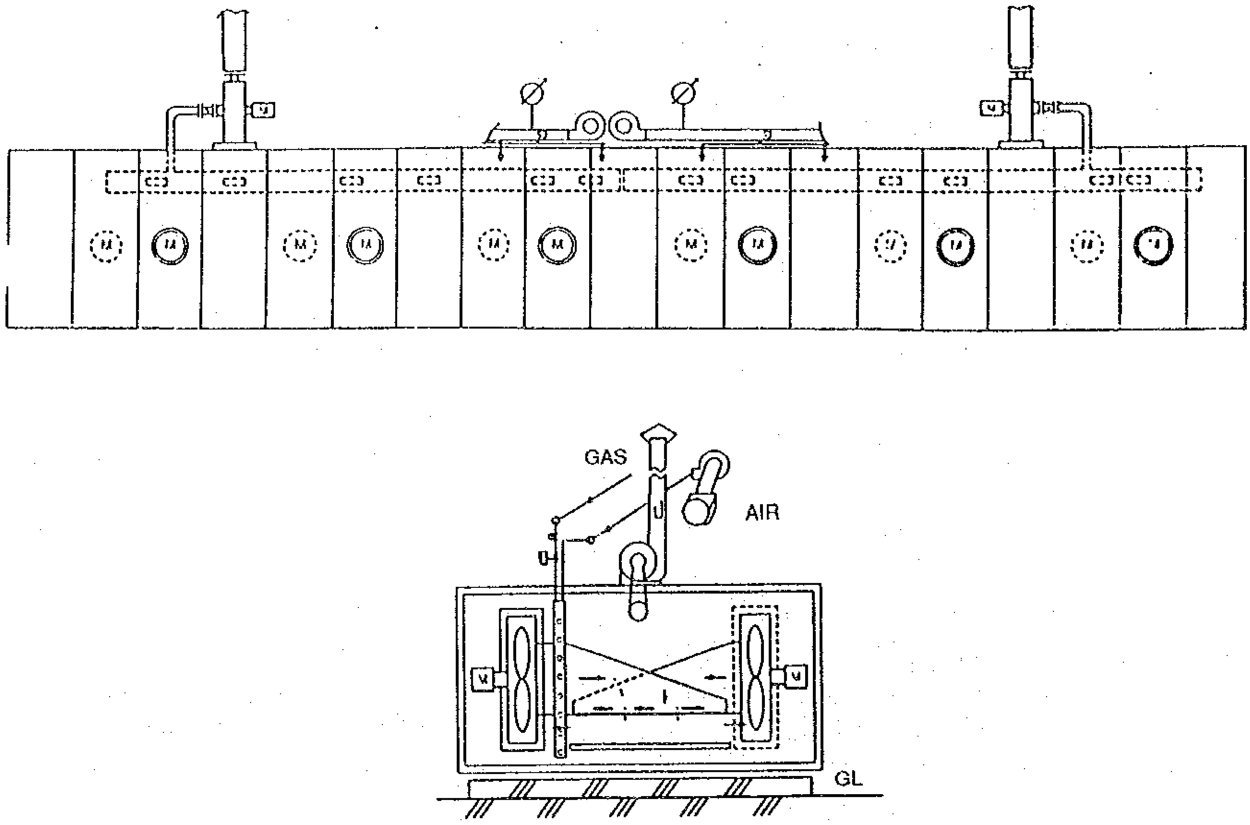


Figure 5.1.18 Frame Dryer

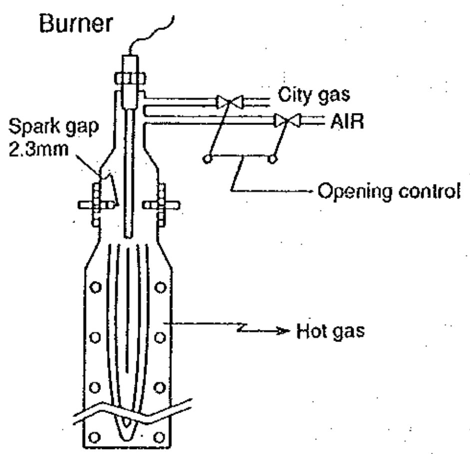


Figure 5.1.19 Burner

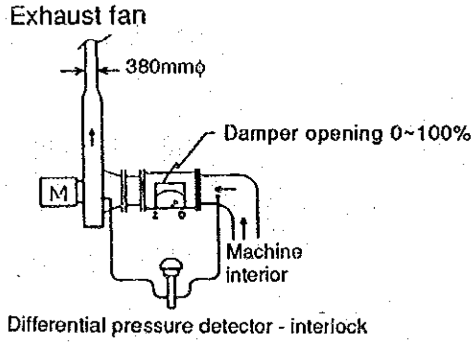


Figure 5.1.20 Exhaust Fan

b. Situations of running

Two exhaust fans are provided. The O_2%, CO_2%, CO%, exhaust gas temperature and gas flow velocity were measured in the duct at the outlet of the left side fan, and the O_2% and exhaust gas temperature were measured in the duct at the outlet of the right side fan. In addition, the city gas consumption was measured with the gas meter located along the gas line.

The results of measurement of components of the exhaust gases are shown in Table 5.1.12, and the exhaust gas flow and consumed quantity of heat are shown in Table 5.1.13.

Table 5.1.12 Properties of exhaust gases

Time	Damper opening		O_2 %		CO_2 %	CO %	Exhaust gas temperature °C		Gas flow velocity m/S	Dryness
	L	R	L	R			L	R		
15:35	70	90	18.2	17.9	3.0	0.03	127	-	15.3	Good
15:55	60	60	17.2	15.6	3.6	0.02	129	136	5.8	Good
16:25	60	60	16.5	16.1	3.8	0.01	131	147	5.3	Good

The exhaust gas flows in the cases where the openings of suction dampers are 70% on the left and 90% on the right and 60% on the left and 60% on the right are calculated below.

The exhaust gas flow can be expressed by equation (8).

$$G = \frac{\pi}{4} \times D^2 \times V \times 3,600 \times \frac{273}{273+t} \times \frac{760+p}{760} \qquad (8)$$

where; G : Exhaust gas flow — Nm^3/h

D : Exhaust duct inside diameter — m

V : Exhaust gas flow velocity — m/S

t : Exhaust gas temperature — °C

p : Exhaust gas pressure — mmHg

However, the pressure of the exhaust gas is neglected because it was not measured this time.

Case of opening of 70% on the left and 90% on the right:

$$G = \frac{\pi}{4} \times 0.380^2 \times 15.3 \times 3,600 \times \frac{273}{273+127}$$
$$= 4,260 \text{ Nm}^3/\text{h}$$

Case of opening of 60% on the left and 60% on the right:

$$G = \frac{\pi}{4} \times 0.380^2 \times 5.3 \times 3,600 \times \frac{273}{273+131}$$
$$= 1,460 \text{ Nm}^3/\text{h}$$

It is assumed that the total flow of exhaust gases on the left side and right side is twice as much as the above figures.

The city gas consumption, exhaust gas flow and its quantity of heat are shown in Table 5.1.13.

Table 5.1.13 Exhaust gas flow and consumed quantity of heat

Damper opening % L R	Exhaust gas flow Nm^3/h	City gas consumption m^3/h	Input quantity of heat (a) MJ/h	Exhaust gas quantity of heat (b) MJ/h	Ratio b/a %
70 – 90	8,520	110.7	3,955	1,202	30
60 – 60	2,920	38.4	1,372	447	33

Note: It is assumed that the exhaust gas specific heat is 1.32 kJ/(Nm^3•C) and the reference temperature is 20°C.

Even when the suction damper openings of exhaust fans were reduced, there was no abnormality in the dryness of the cloth, and reduction of exhaust gas flow, reduction of city gas consumption and rise of dryer internal temperature were observed.

B) Countermeasures for improvement and their effect

a. Reduction of exhaust gas losses

a) Adjustment of exhaust fan suction damper openings

A dryness sensor is mounted at the dryer outlet, and its result is indicated by a lamp. It is recommended that standardization of optimum damper openings is made by repeating the test to adjust the exhaust fan suction damper openings to such an extent that insufficient dryness or excessive dryness will not occur with different conditions such as cloth type, treating speed and extent of preparatory drying.

b) Effect

According to test results (Table 5.1.13), the city gas consumption decreased from 110.7 Nm^3/h to 38.4 Nm^3/h by adjustment of the damper openings.

The annual saving amount achieved by reduction of city gas consumption can be expressed by equation (9).

$$Q = G \times H \times @ \qquad (9)$$

where;	Q :	City gas saving amount	Ft/y
	G :	City gas consumption reduction (110.7 – 38.4 = 72.3 Nm^3/h)	Nm^3/h
	H :	Annual running time (4,800 h/y)	h/y
	@ :	City gas unit price (13.5 Ft/Nm^3)	Ft/Nm^3

$$\begin{aligned} \therefore Q &= (110.7 - 38.4)\ Nm^3/h \times 4{,}800\ h/y \times 13.5\ Ft/Nm^3 \\ &= 347{,}000\ Nm^3/y \times 13.5\ Ft/Nm^3 \\ &= 4{,}685{,}000\ Ft/y \end{aligned}$$

b. Heat recovery from exhaust gases

a) Heat exchange of exhaust gases

As the exhaust gas temperature at the exhaust fan outlet is 127~147°C, it is recommended that heat is recovered from exhaust gases to be used as combustion air. (Figure 5.1.21)

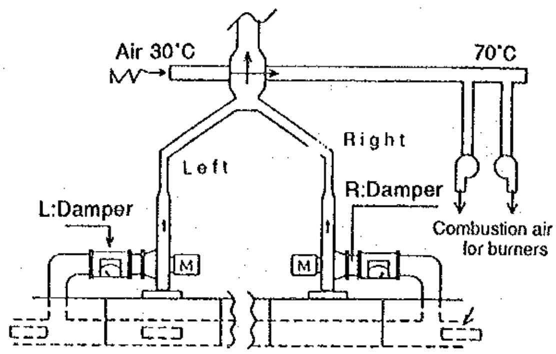


Figure 5.1.21 Flow of heat recovery from frame dryer exhaust gases

The required heat transfer area is calculated assuming that heat recovery is made using a multi-tube type heat exchanger under the condition of after damper adjustment (Table 5.1.14). 2,100 Nm³/h, which is equivalent to the capacity of the forced draft fan, is adopted as the supply air flow.

Table 5.1.14

	Flow rate Nm³/h	Temperature at inlet °C	Temperature at outlet °C
Exhaust gases	2,920	140	110
Combustion air	2,100	30	70

Heat transfer area A is obtained by equation (6).

Heat transfer rate $Q = (140 - 110)\ ^\circ C \times 1.32\ kJ/(Nm^3 \cdot {}^\circ C) \times 2,920\ Nm^3/h = 115,600\ kJ/h$

Exhaust gas specific heat $1.32\ kJ/(Nm^3 \cdot {}^\circ C)$

Total heat transfer coefficient $K = 630\ kJ/(m^2 \cdot h \cdot C)$

Logarithmic mean temperature difference (1 was adopted as the correction factor, although the type is cross flow type.)

$$\Delta t_m = \frac{(140-30)-(110-70)}{2.3\ \log \dfrac{110}{40}}$$
$$= 69°C$$

$\therefore$ Heat transfer area $\quad A = \dfrac{115,600}{630 \times 69} = 2.6\ m^2$

b) Equipment investment expenses ¥1,000,000 (equivalent to 500,000 Ft)

Breakdown	Heat exchanger	¥500,000
	Ducts	¥500,000
	Total	¥1,000,000

c) Effect

The fuel saving rate achieved by preheating of the air is given by equation (10).

$$S = \frac{P}{F - Q + P} \times 100 \qquad (10)$$

where; S : Fuel saving rate %

P : Quantity of heat brought in by preheated air kJ/Nm³-Fuel

F : Calorific value of fuel 35,700 kJ/Nm³-Fuel

Q : Quantity of heat brought away by combustion gases (see Table 5.1.13) kJ/Nm³-Fuel

Air specific heat 1.30 kJ/(Nm³·°C)

$P = (70-30)°C \times 2,100\ Nm^3/h \times 1.30\ kJ/(Nm^3 \cdot °C)/38.4\ Nm^3/h$
$= 2,840$ kJ/Nm³-Fuel

$F = 35,700$ kJ/Nm³-Fuel

$Q = 447,000/38.4 = 11,600$ kJ/Nm³-Fuel

$$S = \frac{2,840}{35,700 - 11,600 + 2,840} \times 100 = 10.5\%$$

Annual fuel saving amount

$$38.4\ Nm^3/h \times 0.105 \times 4{,}800\ h/y \times 13.5\ Ft/Nm^3$$
$$= 19{,}400\ Nm^3/y \times 13.5\ Ft/Nm^3$$
$$= 262{,}000\ Ft/y$$

c. Cleaning of filter for hot gas circulation fan

The cotton dust and ravelings, etc generated out of cotton fibers, tend to clog the filter and drop in the drying rate occurs due to reduction of airflow. It is therefore necessary to secure the circulation airflow by periodically cleaning the filter.

(4) Boilers

A) General description of private thermal power generation plant

The entire steam consumed at this factory and electric power of the system independent from the commercial power line are supplied by its own thermal power generation plant. The steam system is composed of a 6 bar extraction steam line and a 1.5 bar exhaust steam line. Make-up water produced by ion exchange softening treatment of a part of the factory water obtained by treating river water and factory condensate are used for the boilers after treatment deaeration.

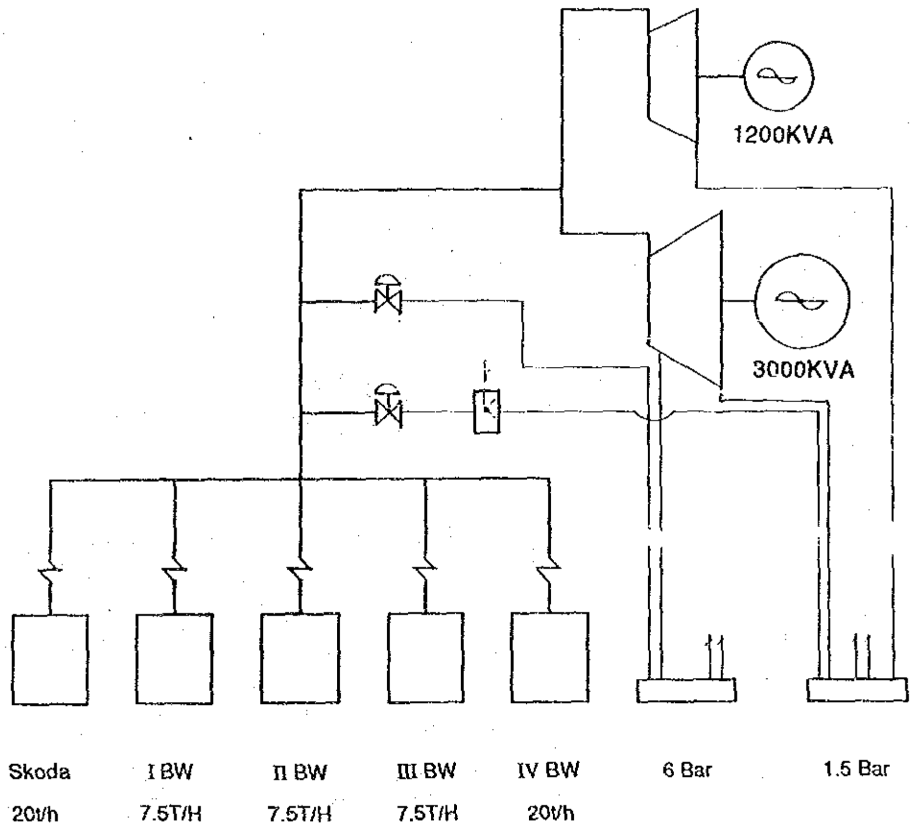


Figure 5.1.22 Flow sheet of private thermal power generation plant

Particulars of principal components of the private thermal power generation plant are shown in Table 5.1.15.

Table 5.1.15 Particulars of principal components

① Boiler (performance test boiler)	
Manufacturer	Rock Gépgyár (B/W)
Max. continuous evaporation rate	20 t/h
Normal pressure × temperature	20.6 bar(G) × 307°C
Fuel	City gas
Number	1
② Boiler	
Manufacturer	Rock Gépgyár (B/W)
Max. continuous evaporation rate	7.5 t/h
Normal pressure x temperature	20.6 bar(G) × 307°C
Fuel	City gas
Number	3 (No. 3 boiler was run at the time of data sampling)
③ Boiler	
Manufacturer	Skoda
Max. continuous evaporation rate	20 t/h
Normal pressure x temperature	20.6 bar(G) × 350°C
Fuel	City gas
Number	1 (Running is suspended due to failure)
④ Turbine generator	
Manufacturer	Lang
Type	Back pressure extraction turbine
Generator output	3,000 kVA × 380 V × 50 Hz × 2P
Number	1
⑤ Turbine generator	
Manufacturer	Brown-Boveri
Type	Back pressure turbine
Generator output	1,200 kVA × 380 V × 50 Hz × 4P
Number	1 (reserve)

B) Heat balance

a. Particulars of boiler

The particulars of the performance tested boiler are shown in Table 5.1.16, and a flow sheet is shown in Figure 5.1.23.

Table 5.1.16 Particulars of boiler

Manufacturer's name	Rock Gépgyár (B/W)	
Date of production/modification	1928/1987	
Type	Water tube type	
Max. continuous evaporation rate	20	t/h
Max. working pressure	21.58	bar(G)
Max. working temperature	350	°C
Normal pressure	20.6	bar(G)
Normal temperature	307	°C
Boiler heat transfer area	372.4	m^2
Superheater heat transfer area	233	m^2
Economizer heat transfer area	1,125	m^2
Air preheater heat transfer area	-	m^2
Combustion chamber volume	71.7	m^3
Calorific capacity of combustion chamber	856	$MJ/(m^3 \cdot h)$
Fuel	City gas	
Burner type x number	Ring + Center Firing Gas Burner × 2	
Drafting method	Balanced draft	
Smokestack (top bore x height)	2 m × 50 m (common to 5 boilers)	
Control system	Air ratio control : Motor linkage system Others : Manual	

b. Flow sheet

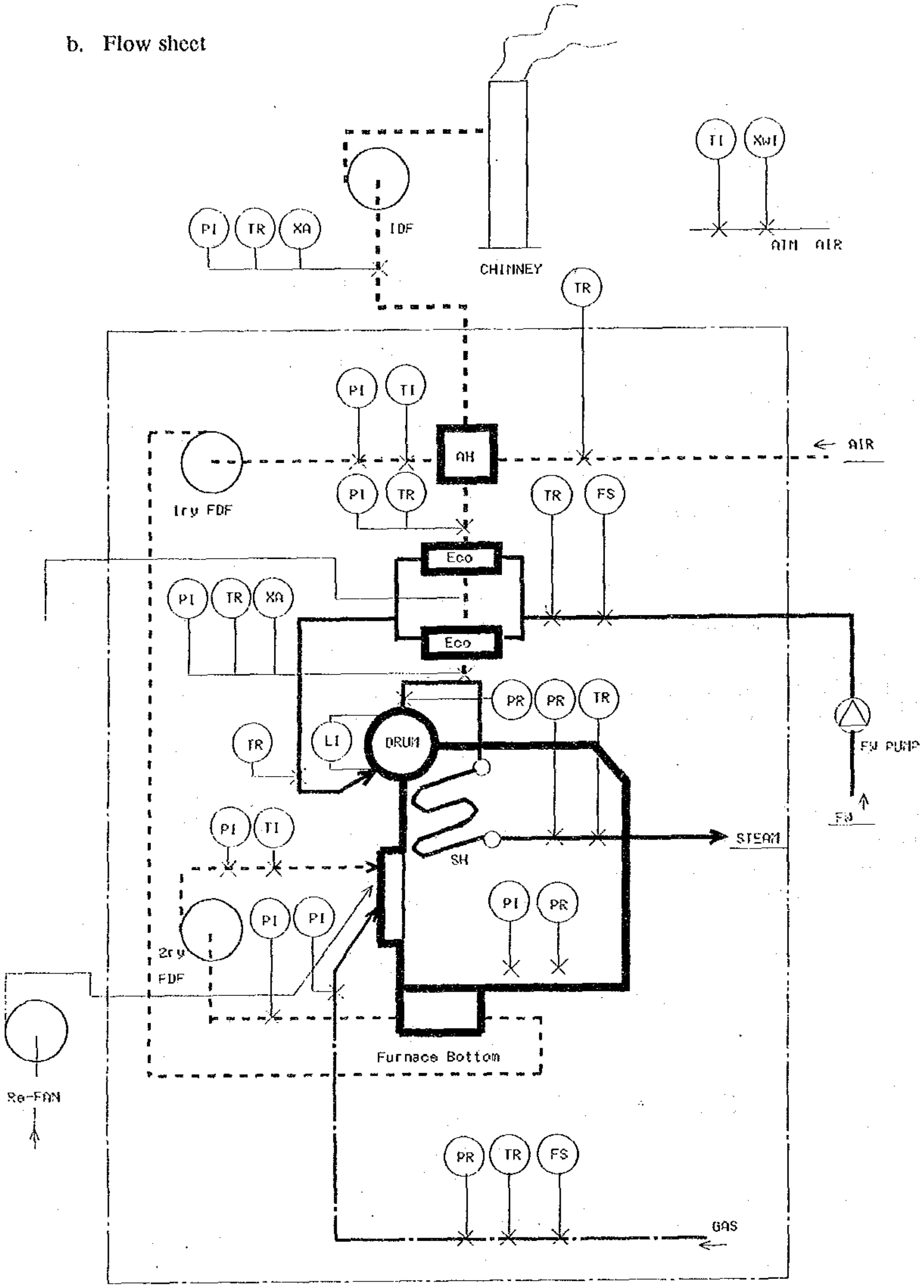


Figure 5.1.23 Boiler flow sheet

c. Results of measurement

The results of measurement are shown in Table 5.1.17.

Table 5.1.17 Results of measurement

Test date and hour	August 15, 1991 (11:10 ~ 12:30)	
Weather and outdoor air temperature	Fine, dry bulb temperature 23°C, wet bulb temperature 18.5°C	
Load factor	103.5	%
Fuel	City gas	
Industrial analyzed value (vol%)	CH_4 98.54, C_2H_6 0.40, C_3H_8 0.14 C_4H_{10} 0.05, C_5H_{12} 0.01, N_2 0.83 CO_2 0.03	
Temperature	23.76	°C
Low calorific value	35.73	MJ/Nm^3
Consumption	1,817	Nm^3/h
Feed water		
Water feed rate (measured value)	20,690	kg/h
Water feed rate (calculated value)	20,700	kg/h
Water feed rate per unit fuel	11.39	kg/Nm^3
Economizer inlet temperature	104.1	°C
Boiler inlet temperature	154.5	°C
Generated steam		
Boiler drum pressure	21.52	$kg/cm^2(G)$
Superheater outlet pressure	20.88	$kg/cm^2(G)$
Superheater outlet temperature	299.6	°C
Combustion air		
Air flow per unit fuel	14.19	Nm^3/Nm^3
Air preheater inlet temperature	30.4	°C
Air preheater outlet temperature	69.2	°C
Air preheater outlet pressure	−10	mmAq
Primary forced draft fan outlet temperature	69.2	°C
Primary forced draft fan outlet pressure	55	mmAq
Wind box (right, left) temperature	77.9, 89	°C
Wind box (right, left) pressure	- , 69	mmAq
Air ratio (boiler furnace outlet)	1.466	
(induced draft fan inlet)	(1.925)	

Table 5.1.17 Results of measurement (Cont'd)

Exhaust gases		
Exhaust gas flow per unit fuel	15.19	Nm^3/Nm^3
Furnace pressure	–9.4	mmAq
Boiler furnace outlet temperature	312.7 (415)	°C
Boiler furnace outlet pressure	–14.6	mmAq
Primary economizer outlet temperature	118.8	°C
Primary economizer outlet pressure	–26.4	mmAq
Secondary economizer outlet temperature	122.4 (307)	°C
Air preheater outlet temperature	179.8 (274)	°C
Air preheater outlet pressure	–39.5	mmAq
Induced draft fan inlet temperature	144.8	°C
Gas analysis		
Boiler furnace outlet	CO_2 4.32, O_2 7.46, CO 0.03	%
Induced draft fan inlet	O_2 10.09	%

Note 1: The following measured data were used for the heat balance.

Water feed rate : Hybrid recorder (integrated value)
Fuel gas flow : City gas flowmeter (integrated value) and No.3 boiler orifice flowmeter
Outdoor air dry/wet bulb temperature: Psychrometer (mean value)
Other data : Hybrid recorder (mean value)

Note 2: The industrial analysis values of fuel were obtained from the gas company.

Note 3: The air ratio at induced draft fan inlet was calculated by equation (11).

$$m = \frac{21}{21 - (O_2)} \qquad (11)$$

Note 4: The air preheater outlet exhaust gas temperature is a calculated value.
(See consideration ②)

Note 5: The values in ( ) of the exhaust gas temperature are corrected values calculated from the heat balance of steam, water and combustion air side.

d. Heat balance chart

A heat balance chart is shown in Table 5.1.18.

Table 5.1.18 Heat balance chart

Heat input		kJ/Nm³	%
① Calorific value of fuel	Hl	35,732	99.62
② Sensible heat of air	Q_a	137	0.38
Total		35,869	100.00
Heat output		**kJ/Nm³**	**%**
Effective heat output (heat absorbed by generated steam)			
Heat absorbed in boiler furnace	Q_b	24,443	68.15
Heat absorbed at economizer	Q_{ec}	2,449	6.83
Heat absorbed at superheater	Q_{sh}	2,494	6.95
Sub total	Q_s	29,386	81.93
Heat loss			
① Loss by exhaust gas retention heat (including water vapor)	L_1	3,287 (5,262)	9.16 (14.67)
② Heat loss caused by incomplete combustion	L_2	50	0.14
③ Radiated heat loss	L_3	339	0.95
④ Other heat losses	L_4	2,807 (832)	7.83 (2.32)
Sub total		6,483	18.07
Total		35,869	100.00
Boiler efficiency			**%**
① Heat input/output method	$y_1 = \{Q_s/(H_1 + Q_a)\} \times 100$		81.91
② Heat loss method	$y_2 = \{1 - (L_1 + L_2 + L_3)/(H_1 + Q_a)\} \times 100$		89.74 (84.23)

Note: The figures in ( ) are of the cases where corrected value (274°C) is used as the exhaust gas temperature.

e. Calculation of heat balance (per fuel of 1 Nm^3)

[Heat input]

① Lower calorific value of fuel, Hl

$$\begin{aligned} Hl &= 357.9CH_4 + 644.6C_2H_6 + 937.7C_3H_8 + 1{,}234.9C_4H_{10} + 1{,}460.9C_5H_{12} \\ &= 357.9 \times 98.54 + 644.6 \times 0.40 + 937.7 \times 0.14 + 1{,}234.9 \times 0.05 + 1{,}460.9 \\ &\quad \times 0.01 \\ &= 35{,}732 \end{aligned}$$ kJ/Nm³-Fuel

② Sensible heat of air, Q_a

$Q_a = A \times C_a \times (t_a - t_o)$ kJ/Nm³-Fuel

$A = m \times A_o \times (1 + 1.61z)$ Nm³/Nm³-Fuel

$$\begin{aligned} A_o &= (2CH_4 + 3.5C_2H_6 + 5C_3H_8 + 6.5C_4H_{10} + 8C_5H_{12})/21 \\ &= (2 \times 98.54 + 3.5 \times 0.4 + 5 \times 0.14 + 6.5 \times 0.05 + 8 \times 0.01)/21 \\ &= 9.504 \end{aligned}$$ Nm³/Nm³-Fuel

$$\begin{aligned} m &= \frac{21}{21 - 79 \times \dfrac{(O_2) - 0.5 \times (CO)}{(N_2)}} \times \left[1 + \{(O_2)/(N_2)\} \times \{n_2/(21A_o)\}\right] \\ &= \frac{21}{21 - 79 \times \dfrac{7.46 - 0.5 \times 0.03}{88.19}} \times \left[1 + \{(7.46/88.19)\} \times \{0.83/(21 \times 9.504)\}\right] \\ &= 1.466 \end{aligned}$$

$$\begin{aligned} A &= 1.466 \times 9.504 \times (1 + 1.61 \times 0.0115) \\ &= 14.19 \end{aligned}$$ Nm³/Nm³-Fuel

$Q_a = 14.19 \times 1.30 \times (30.4 - 23) = 137$ kJ/Nm³-Fuel

where;

A	:	Airflow per fuel of 1 Nm³ (including water vapor)	Nm³/Nm³-Fuel
C_a	:	Mean specific heat of air (1.30 kJ/(Nm³·°C))	kJ/(Nm³·°C)
t_a	:	Air temperature at inlet of air preheater	°C
t_o	:	Ambient temperature	°C
m	:	Air ratio	

z	:	Absolute humidity of outdoor air (0.0115 kg/kg per dry air of 1 kg, from the air diagram)	kg/kg
A_0	:	Theoretical dry airflow	Nm^3/Nm^3-Fuel
(CO_2)	:	Volume % of CO_2 gas in dry combustion exhaust gases	%
(O_2)	:	Volume % of O_2 gas in dry combustion exhaust gases	%
(CO)	:	Volume % of CO gas in dry combustion exhaust gases	%
(N_2)	:	Volume % of N_2 gas in dry combustion exhaust gases $= 100 - \{(CO_2) + (O_2) + (CO)\}$	%
n_2	:	Volume % of N_2 gas in fuel	%

[Effective heat output]

Heat absorbed by generated steam, Q_s

$Q_S = W_2 \times (h_2 - h_1) = 11.39 \times (3{,}018 - 438)$
$= 29{,}386$ kJ/Nm^3-Fuel

$W_2 = W_1 = W/F = 20{,}700/1{,}817 = 11.39$ kg/Nm^3-Fuel

$W = 20{,}690 \times 1.0006 = 20{,}700$ kg/h

$F = F_1 - F_2 = 2{,}116 - 299 = 1{,}817$ Nm^3-Fuel/h

$F_1 = (8{,}947 - 8{,}665) \times 10/1.333 = 2{,}116$ Nm^3-Fuel/h

$F_2 = 3{,}337/10.75 \times 0.9646 = 299$ Nm^3-Fuel/h

① Heat absorbed in boiler furnace, Q_b

$Q_b = W_1 \times (h_4 - h_3) = 11.39 \times (2799 - 653)$
$= 24{,}443$ kJ/Nm^3-Fuel

② Heat absorbed at economizer, Q_{ec}

$Q_{ec} = W_1 \times (h_3 - h_1) = 11.39 \times (653 - 438)$
$= 2{,}449$ kJ/Nm^3-Fuel

③ Heat absorbed at superheater, Q_{sh}

$$Q_{sh} = W_1 \times (h_2 - h_4) = 11.39 \times (3{,}018 - 2{,}799)$$
$$= 2{,}494 \qquad \text{kJ/Nm}^3\text{-Fuel}$$

where;

W_1	:	Water feed rate per fuel of 1 Nm^3		kg/Nm^3-Fuel
W_2	:	Superheater outlet steam flow per fuel of 1 Nm^3		kg/Nm^3-Fuel
W	:	Water feed rate per hour		kg/h
F	:	Fuel gas flow per hour		Nm^3-Fuel/h
F_1	:	City gas flowmeter integrated fuel gas flow per hour		Nm^3-Fuel/h
F_2	:	No. 3 boiler integrated gas flow per hour (calculated from the value indicated in the boiler log)		Nm^3-Fuel/h
h_1	:	Enthalpy of feed water at economizer inlet	438	kJ/kg
h_2	:	Enthalpy of steam at superheater outlet	3,018	kJ/kg
h_3	:	Enthalpy of feed water at boiler inlet	653	kJ/kg
h_4	:	Enthalpy of generated saturated steam	2,799	kJ/kg

: Density correction factor of water feed rate measuring orifice

$$(v_d/v)^{0.5} = (0.0010462/0.0010450)^{0.5} = 1.0006$$

v_d : Specific volume of feed water at orifice data m^3/kg

v : Specific volume of feed water at the temperature at the time of data measurement m^3/kg

: Density, pressure, temperature correction factors of fuel gas flow measuring orifice

$\{(\rho_d/\rho)(P/P_d)(T_d/T)\}^{0.5}$
$= \{(0.772/0.7018)(2.133/2.447)(288/296.8)\}^{0.5} = 0.9646$

ρ_d : Density of fuel gas at the time of orifice design — kg/m³

ρ : Density of fuel gas at the time of data measurement (0.561 × 1.251) — kg/m³

P_d : Absolute pressure of fuel gas at the time of orifice design — kg/cm²

P : Absolute pressure of fuel gas at the time of data measurement — kg/cm²

T_d : Absolute temperature of fuel gas at the time of orifice design — K

T : Absolute temperature of fuel gas at the time of data measurement — K

[Heat loss]

① Loss by exhaust gas retention heat (including water vapor), L_{11}

$L_1 = G \times C_g \times (t_g - t_o)$ kJ/Nm³-Fuel

$G = G_0 + G_w + (m - 1) \times A_0 + G_{w1}$ Nm³/Nm³-Fuel

$$\begin{aligned} G_0 &= \{\Sigma\{(4.76 \times x + 0.94 \times y) \times C_xH_y\} + N_2 + CO_2\}/100 \\ &= \{(4.76 \times 1 + 0.94 \times 4) \times 98.54 \\ &\quad + (4.76 \times 2 + 0.94 \times 6) \times 0.40 \\ &\quad + (4.76 \times 3 + 0.94 \times 8) \times 0.14 \\ &\quad + (4.76 \times 4 + 0.94 \times 10) \times 0.05 \\ &\quad + (4.76 \times 5 + 0.94 \times 12) \times 0.01 \\ &\quad + 0.83 + 0.03\}/100 \\ &= 8.513 \end{aligned}$$ Nm³/Nm³-Fuel

$$\begin{aligned} G_w &= (4CH_4 + 6C_2H_6 + 8C_3H_8 + 10C_4H_{10} + 12C_5H_{12})/200 \\ &= (4 \times 98.54 + 6 \times 0.40 + 8 \times 0.14 + 10 \times 0.05 + 12 \times 0.01)/200 \\ &= 1.992 \end{aligned}$$ Nm³/Nm³-Fuel

$$\begin{aligned} G_{w1} &= 1.61 \times z \times m \times A_o = 1.61 \times 0.0115 \times 1.466 \times 9.504 \\ &= 0.258 \end{aligned}$$ Nm³/Nm³-Fuel

$$G = 8.513 + 1.992 + (1.466 - 1) \times 9.504 + 0.256$$
$$= 15.19 \quad \text{Nm}^3/\text{Nm}^3\text{-Fuel}$$

$$L_1 = 15.19 \times 1.38 \times (179.8 - 23)$$
$$= 3287 \quad \text{kJ/Nm}^3\text{-Fuel}$$

where;

G : Actual exhaust gas flow per fuel of 1 Nm^3 (including water vapor) — Nm^3/Nm^3-Fuel

G_0 : Theoretical dry exhaust gas flow — Nm^3/Nm^3-Fuel

G_w : Water vapor flow generated by combustion — Nm^3/Nm^3-Fuel

G_{w1} : Water vapor flow caused by moisture in the combustion air — Nm^3/Nm^3-Fuel

C_g : Mean specific heat of exhaust gases (1.38 kJ/($Nm^3 \cdot$°C) in general) — kJ/($Nm^3 \cdot$°C)

t_g : Exhaust gas temperature at air preheater outlet — °C

t_o : Ambient temperature — °C

m, z, A_0: Same as before

② Loss caused by incomplete combustion, L_2

$$L_2 = 128 \times \{G_0 + (m - 1) A_0\} (CO)$$
$$= 128 \times \{8.513 + (1.466 - 1) \times 9.504\} \times 0.03$$
$$= 50 \quad \text{kJ/Nm}^3\text{-Fuel}$$

where;

G_0, m, A_0, (CO): Same as before

③ Radiated heat loss, L_3

$$L_3 = I_r \times (H_h + Q_a)/100 = 0.85 \times (39{,}735 + 137)/100 = 339 \quad \text{kJ/Nm}^3\text{-Fuel}$$

$$\begin{aligned} H_h &= H_l + 10.05 \times (4CH_4 + 6C_2H_6 + 8C_3H_8 + 10C_4H_{10} + 12C_5H_{12}) \\ &= 35{,}732 + 10.05 \times (4 \times 98.54 + 6 \times 0.40 + 8 \times 0.14 \\ &\quad + 10 \times 0.05 + 12 \times 0.01) \\ &= 39{,}735 \quad \text{kJ/Nm}^3\text{-Fuel} \end{aligned}$$

where;

I_r : Radiated heat loss rate

Assumed as 0.85 from the diagram of ASME that indicates general values of radiated heat loss rate against high heating values.

H_h: High heating value of fuel kJ/Nm³-Fuel

④ Other heat losses, L_4

$$\begin{aligned} L_4 &= H_l + Q_a - (Q_s + L_1 + L_2 + L_3) \\ &= 35{,}732 + 137 - (29{,}386 + 3{,}287 + 50 + 339) \\ &= 2807 \quad \text{kJ/Nm}^3\text{-Fuel} \end{aligned}$$

f. Considerations

① The water feed rate during the performance test period was 21.76 t/h at maximum and 19.17 t/h at minimum among the momentary recorded values, and the load variation is minor. As the feed water flow rate and fuel gas flow rate are not controlled as linked, correlation between them could not be observed in a short time as shown in Figure 5.1.24.

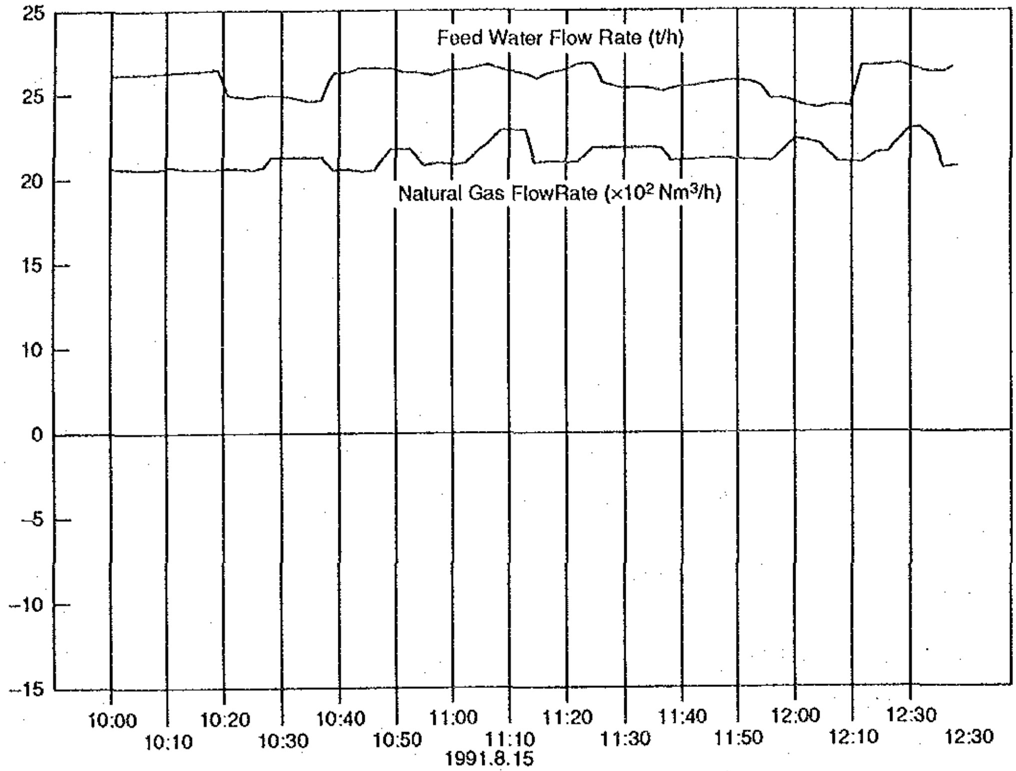


Figure 5.1.24 Variation of feed water flow rate and fuel gas flow rate

② "Other losses" is large in the heat balance chart. The following factors can be raised as its causes.

a) The duct on the suction side of the induced draft fan (IDF) is connected with the duct of another boiler, the running of which is suspended, and isolation from it is made with a damper only. Therefore, there is a possibility where cold air is leaking, and it is considered that IDF inlet exhaust gas temperature is recorded at a level that is lower than the actual temperature. When the air preheater outlet exhaust gas temperature is corrected from the air ratio between boiler outlet and IDF inlet under these circumstances, 179.8°C is obtained. But it is judged that this value is still lower than the actual value.

The exhaust gas temperature at the air preheater outlet calculated from the heat balance between the water side and the combustion air side is 274°C. Values calculated using this value as a reference are jointly indicated in ( ) of the heat balance chart.

b) Calculation of the air ratio was made based on the assumption that the exhaust gas analyzed value at the boiler furnace outlet is same as the exhaust gas analyzed value at the air preheater outlet. At the air preheater outlet, however, it is considered that the air ratio is larger than this value because of air leakage. It means that "Loss of exhaust gases retention heat" was calculated to a level that is lower by this extent.

C) Countermeasures for improvement and their effect

a. Improvement of air ratio

The O_2% rapidly increases as the load factor decreases, and the air ratio is excessive in general, as seen in the records obtained with hybrid recorders (Figure 5.1.25).

Load	O_2 %	Air ratio
20.7 t/h	7.46%	1.466
90%	9%	1.8
70%	13%	2.6

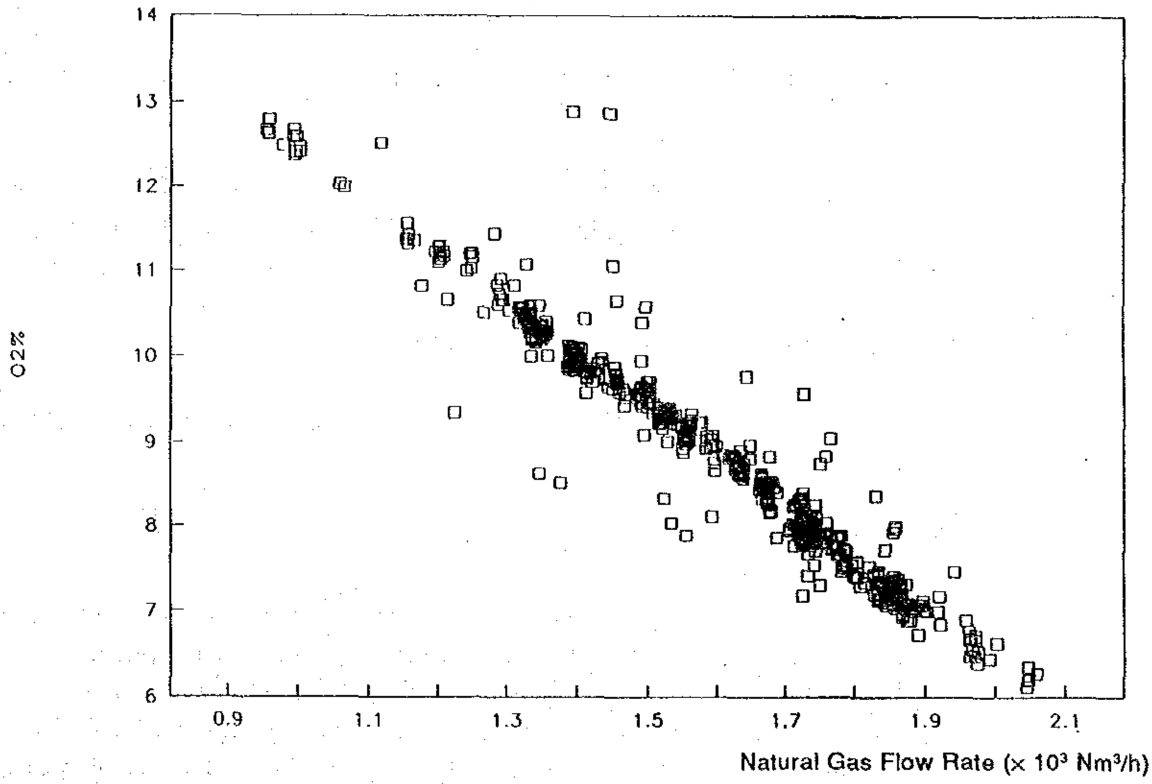


Figure 5.1.25 Exhaust gas O_2 characteristics

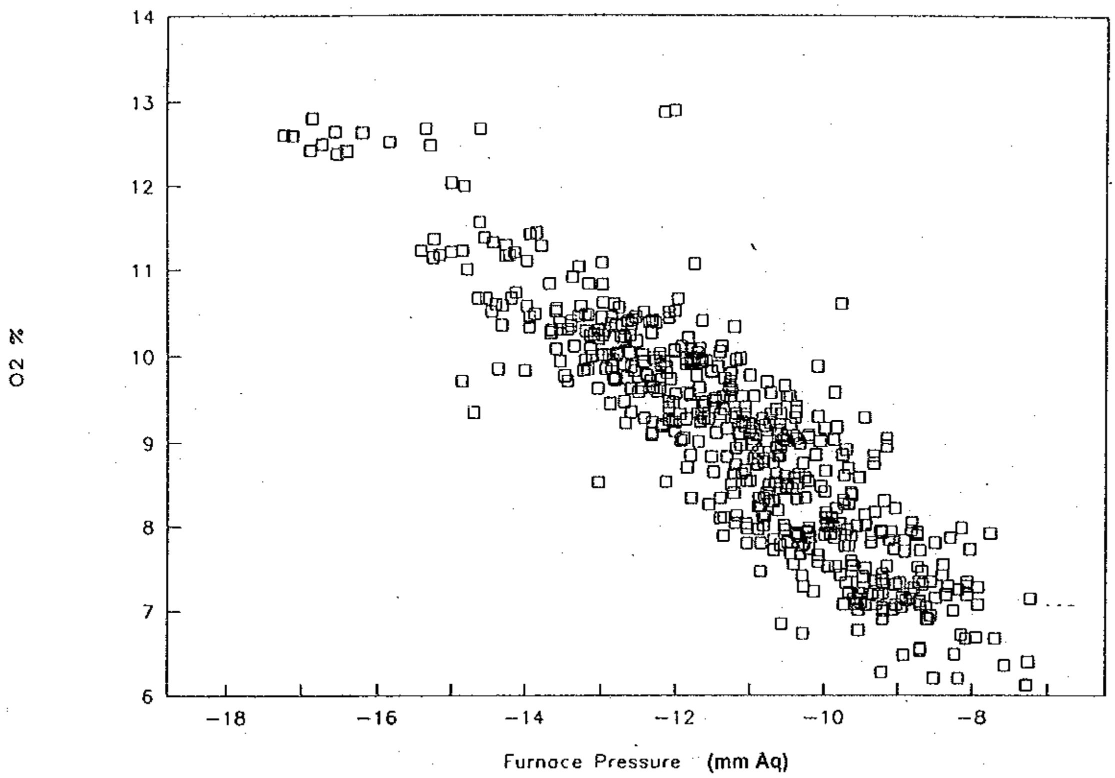


Figure 5.1.26 Correlation between internal pressure and exhaust gas O_2

These burners are natural gas firing burners, and it is sufficiently possible to set the air ratio at 1.15 at the time of maximum load. The countermeasures stated below are effective.

① Change to setup of air-fuel ratio controller

Make readjustment of the air-fuel ratio setup cam for each of the right and left burners.

② Introduction of internal pressure controller

As the damper on the IDF suction side is fixed at the present time, the internal pressure drops as the load decreases, suction of cold air from outside occurs, and it constitutes a cause for increase of the air ratio (Figure 5.1.26). The air-fuel ratio will not become stable unless the internal pressure is kept at a fixed level regardless of load variation. It is desirable that an internal pressure controller is introduced.

As the internal negative pressure can be set at a low level when an internal pressure controller is introduced, the effect is multiplied. (With a balanced draft type boiler in Japan, the internal pressure is set at –5 ~ –2 mmAq in general.)

Furthermore, it is recommended that complete isolation is made by inserting a shield plate to the communicating damper between the flue duct of this boiler and the flue duct of another boiler.

③ Adjustment of burner

It is hard to secure the burner performance unless the flames of right and left burners are equal. It is recommended that adjustment is made so that fuel is equally distributed to right and left burners simultaneously with change to setup of the air-fuel ratio controller.

Furthermore, make adjustment so that mixing of the combustion air with natural gas is completely made. (See item ① of b. Prevention of heat loss caused by incomplete combustion on p. 5-1-59.)

④ Introduction of boiler steam pressure controller

Rapid increase or decrease of the fuel flow makes the air ratio unstable, and excessive air ratio and incompletion combustion will result. It is therefore desirable that the boiler steam pressure controller is introduced.
(See item ② of b. Prevention of heat loss caused by incomplete combustion.)

The expenses required for installation of internal pressure controller and boiler steam pressure controller are about ¥4,300,000 (equivalent to about 2,200,000 Ft) in Japan including pressure converters, regulators and working expenses.

The effect obtained in the case where the air ratio is improved is as follows.

[Preconditions]

20.7 t/h load running time	1,600	h/y
90% load running time	3,200	h/y
Boiler fuel unit price	13.5	Ft/Nm^3

Symbols m, A, G, L_1, Hl are same as those in the section of heat balance.

[Effect]

① At the time of 20.7 t/h load running

The fuel reduction rate at the time when the air ratio is reduced from 1.466 to 1.15 is as shown in Table 5.1.19. The air preheater outlet exhaust gas temperature was assumed as 274°C in this case.

Table 5.1.19 Fuel reduction rate by improvement of air ratio (1)

	m	A	G	L1	HI
Before improvement After improvement	1.466 1.15	14.19 11.13	15.19 12.19	5,262 4,222	35,732 35,732
Reduction rate	1 – {(35,732 – 5,262)/(35,732 – 4,222)} = 0.033				3.3%

② At the time of 90% load running

The fuel reduction rate at the time when the air ratio is reduced from 1.8 to 1.15 is as shown in Table 5.1.20. The air preheater outlet exhaust gas temperature was assumed as 240°C in this case.

Table 5.1.20 Fuel reduction rate by improvement of air ratio (2)

	m	A	G	L1	HI
Before improvement After improvement	1.8 1.15	17.42 11.13	18.37 12.19	5,501 3,650	35,732 35,732
Reduction rate	1 – {(35,732 – 5,501)/(35,732 – 3,650)} = 0.058				5.8%

③ The fuel reduction effect in this case is as follows.

Fuel reduction rate

$0.033 \times 1{,}817\ Nm^3/h \times 1{,}600\ h/y$
$+ 0.058 \times 1{,}817\ Nm^3/h \times 18.0/20.7 \times (1 - 0.032)/(1 - 0.058) \times 3{,}200\ h/y$
$= 397{,}300$ Nm^3/y

Reduced fuel cost

$397{,}300\ Nm^3/y \times 13.5\ Ft/Nm^3 = 5{,}364{,}000$ Ft/y

Therefore, the equipment expenses for improvement can be recovered in about six months.

b. Prevention of heat loss caused by incomplete combustion

CO emission of 0.06 ~ 0.01%, 0.02% at average is observed according to the data recorded during the period when recording was made using a hybrid recorder (19:10 on August 14 through 12:33 on August 15). As natural gas is used as the fuel, it is possible to reduce this heat loss to zero by taking the following countermeasures.

① Adjustment of burners

In order that mixing of the combustion air with natural gas is quickly made, the jet pattern determined by the angle, velocity and pressure distribution of the combustion air and the natural gas at the time of jetting out of each burner should be suitable.

Furthermore, in the case where gas under burning makes contact with the boiler's heat transfer surface in the combustion chamber, incomplete combustion occurs because combustion is not completed. Therefore, the flame shape should be corrected.

It is therefore necessary to obtain the optimum point by repeating adjustment of load distribution to the center burner and the ring burner, adjustment of center burner inserting position, burner tip change test and so forth.

② Adjustment of combustion rate

When the load varies, the followability of the air flow to the variation of the fuel flow involves capacity lag characteristics even if the motor linkage system is adopted. Accordingly, rapid increase of the combustion rate should be avoided because temporary air flow shortage results. Rapid reduction of the combustion rate, on the other hand, causes excessive air flow. It is recommended that 20%/min is aimed at as a criterion of fuel rate change speed. Furthermore, it is desirable that automatic control is made through introduction of the boiler steam controller.

The effect obtained when incomplete combustion is prevented is as follows.

[Preconditions] (Other preconditions are same as before)

Boiler load	18.0 (90% load)	t/h
Annual running time	4,800	h

[Effect]

Reduction of fuel consumption
($50\ kJ/Nm^3 \times 1{,}817\ Nm^3/h \times 18.0/20.7 \times 4{,}800\ h/y\ /\ 35{,}732\ kJ/Nm^3$)
=10,600 Nm^3/y

Reduction of fuel expenses
$10{,}600 \times 13.5 = 143{,}000$ Ft/y

c. Recovery of condensate

This factory is equipped with a condensate recovery equipment. But it is judged that the recovery rate is no good because the temperature of the water feed tank is low. The deaerator heating steam and make-up water can be saved when recovery of the condensate is strengthened.

① Formula for calculation

The deaerator heating steam flow and make-up water flow can be expressed by equations (12) and (13).

$$W_d = W \times (h_1 - h_d)/(h_s - h_d) \times 1.03 \qquad (12)$$

$$W_s = (W - W_d) \times (t_c - t_d)/(t_c - t_m) \qquad (13)$$

where;

W_d	:	Deaerator heating steam flow		kg/h
W	:	Boiler feed water flow	18,000	kg/h
W_s	:	Make-up water flow		kg/h
h_s	:	Enthalpy of deaerator heating steam	2,716	kJ/kg
h_d	:	Enthalpy of feed water at deaerator inlet		kJ/kg
t_d	:	Temperature of feed water at deaerator inlet		°C

h_1	: Enthalpy of feed water at deaerator outlet	438	kJ/kg
t_m	: Temperature of make-up water	10	°C
t_c	: Temperature of condensate	90	°C
1.03	: Coefficient that considers steam flow and heat radiation accompanying air discharge		

The relation among the deaerator feed water temperature, required heating steam flow and deaerator make-up water flow is shown in Table 5.1.21.

Table 5.1.21 Deaerator feed water temperature and heating steam flow

Deaerator feed water temperature t_d	°C	40	50	60	70	80
Heating steam flow $1.03\ W_d$	kg/h	1968	1692	1405	1109	803
Make-up water flow W_s	kg/h	10056	8179	6238	4231	2153

② Effect

The effect obtained when the deaerator feed water temperature is increased from 40°C to 60°C is as follows.

[Preconditions]

The water softening expenses are determined as 57.2 Ft/m³ from water expenses 19.8 Ft/m³ and cost required for softening 37.4 Ft/m³ (results of 1990).

[Effect]

The amount of reduction of the fuel expenses caused by reduction of the steam consumption can be calculated with equation (2).

$$
\begin{aligned}
&(1968 - 1405)\ \text{kg/h} \times 1/11.39\ \text{kg/Nm}^3 \times 4{,}800\ \text{h/y} \times 13.5\ \text{Ft/Nm}^3 \\
&= 237{,}300\ \text{Nm}^3\text{/y} \times 13.5\ \text{Ft/Nm}^3 \\
&= 3{,}204{,}000 \qquad \text{Ft/y}
\end{aligned}
$$

Reduced make-up water expenses

$(10{,}056 - 6{,}238)/1{,}000\ m^3/h \times 4{,}800\ h/y \times 57.2\ Ft/Nm^3$
$= 18{,}300\ m^3/y \times 57.2\ Ft/Nm^3$
$= 1{,}047{,}000$ Ft/y

Total reduction in the amount

$3{,}204{,}000 + 1{,}047{,}000 = 4{,}251{,}000$ Ft/y

d. Management by data

Measured data of high accuracy and prompt judgment based on the judging indices calculated based on the measured data are required in order to perform safe running and economic running of boilers. Data measured once every hour are recorded on the boiler log at the present time. But check by the following judging indices is required in addition.

① Boiler efficiency/load characteristics

These are indices which express the performance of a boiler. It is desirable that these indices are calculated once every 8 hours or once daily. The situations of internal contamination to heat transfer surfaces, occurrence of bypass of combusted gas due to damage to bricks, etc. can be judged through comparison of these indices.

Furthermore, the evaporation multiple obtained by dividing the evaporation rate by the fuel volume is convenient as a simple index that expresses the performance of the boiler. Simple evaporation multiple to use the measured data as it is as the evaporation rate and equivalent evaporation multiple calculated by the ratio to the evaporation latent heat of water at 100°C (2,256.9 kJ/kg) are available.

The simple evaporation multiple at the time of data measurement is as follows.

$W/F = 20{,}700/1{,}817 = 11.39$ kg/Nm^3

② Exhaust gas temperature at various points/load characteristics

③ Pressure at various points/load characteristics

④ Feed water temperature at various points/load characteristics

⑤ O_2%, CO_2% in exhaust gases, air ratio/load characteristics

It is needless to say that high accuracy is required in the measured data, and consolidation of measuring instruments and maintenance and inspection of instruments based on the work standards are important.

e. Installation of automatic control devices

It is not exaggerating to say that furnishment of automatic control devices is essential for all of uplift of productivity, enhancement of quality and labor saving.

The situations are entirely identical for boilers, and installation and modernization of automatic control devices such as controllers for steam pressure, drum water level, air ratio and furnace internal pressure are wanted.

f. Withdrawal of recirculation of exhaust gases

The exhaust gases of about 15% of the entire exhaust gases are recirculated in the furnace at 100% load at the present time with protection of furnace refractories as the objective. In general, there are cases where recirculation of exhaust gases is made with the objective to control the superheating steam temperature and to reduce nitrogen oxides in the exhaust gases. With these boilers, however, it is judged that the necessity of implementation of recirculation of exhaust gases is minor. It is rather considered that it exerts adverse effect over the air ratio and generation of CO.

The following effect can be anticipated when the exhaust gas recirculation system is withdrawn.

① Rise of generator net thermal efficiency caused by rise of superheating steam temperature

(It is estimated that the generator net thermal efficiency increases by about 8% when the superheating steam temperature becomes 340°C at the turbine inlet, and its effect is large.)

② Reduction of heat radiation loss from the exhaust gas recirculation system

③ Reduction of power of the exhaust gas recirculation fan

Cooperation of the boiler manufacturer is required to concretize the countermeasures. But refractories of higher refractoriness should be used for the furnace walls.

g. Use of vented turbine exhaust

The thermal power generation plant of this factory is not linked with the commercial power supply network. Accordingly, it is not possible to absorb the imbalance between the steam load and the electric power load by purchased electric energy. Therefore, power generation is made in correspondence to the load of the power generation system, and the surplus of the turbine exhaust is dispersed to the atmosphere. Thus, energy loss and high power generation cost are resulted.

It is desirable that linkage with the commercial power supply network is made with necessary protective measures taken. But use of surplus steam for feed water preheating and hot water production can also be considered.

Since the economy of countermeasures is related to the steam surplus rate, it is necessary to investigate the pattern of demand of electric energy and steam for a certain length of time when stable continuous operation is achieved.

(5) Steam lines

a. Prevention of heat radiation from pipelines

Many steam pipelines are laid both inside and outside of buildings, because there are many equipment using steam at this factory. But places where heat insulation is damaged and sections where no heat insulation was made from the beginning were observed. Furthermore, valves and flange are not heat insulated at all.

Heat insulation of steam pipeline system is a basic matter for prevention of thermal losses, and large economic effect will be obtained by taking measures.

The situations of heat insulation were investigated only in the room where the washer and the cylinder dryer are installed and the room where the textile printing machine is installed during the investigation of this time because of restrictions in time. It is estimated that the total extension of steam lines in the whole factory is about three times of the investigated portion.

The situations of heat radiation caused by incomplete heat insulation to pipes and valves in the investigated portions and estimated reduction of heat radiation rate to be achieved through improvement of heat insulation are shown in Table 5.1.22 and Table 5.1.23. It is anticipated that the heat radiation rate can be reduced as follows in the whole factory through repair to the heat insulation.

$$(166.2 + 166.5) \times 3 = 998.1 \text{ MJ/h}$$

The fuel saving amount can be calculated with equation (2). Here, it is assumed that the evaporation latent heat of the steam of 1.3 bar(G) is 2,189 kJ/kg.

$$998{,}100 \text{ kJ/h} \times 4{,}800 \text{ h/y} \times 1/2{,}189 \text{ kg/kJ} \times 1/11.39 \text{ Nm}^3\text{/kg} \times 13.5 \text{ Ft/Nm}^3$$
$$= 192{,}200 \text{ Nm}^3 \times 13.5 \text{ Ft/Nm}^3$$
$$= 2{,}595{,}000 \text{ Ft/y}$$

The amount required for heat insulation work is estimated as about ¥2,600,000 (equivalent to about 1,300,000 Ft) when the expenses in Japan are used as a reference. Therefore, the working expenses can be recovered in about six months.

The expenses for the heat insulation work on the field checked during the investigation were considerably lower than the level in Japan. Furthermore, the economy will be even better if heat insulation materials are purchased and working is performed by the factory employees.

Table 5.1.22 Insulation of Bare Steam Pipe

Room	Steam Press bar(G)	Pipe		Heat Loss Present		Heat Loss after Insulation				Saved Energy MJ/h
		Dia inch	Length m	kJ/(mh)	MJ/h	Material	Thick mm	kJ/(mh)	MJ/h	
Washer & Cylinder Dryer	1.3	8	1	3,654	3.7	Mineral Wool	25	407	0.4	3.2
	1.3	6	2	2,872	5.7	Mineral Wool	25	322	0.6	5.1
	1.3	3	20	1,658	33.2	Mineral Wool	25	195	3.9	29.3
	1.3	2	20	1,176	23.5	Mineral Wool	25	146	2.9	20.6
	1.3	1	2	712	1.4	Mineral Wool	15	132	0.3	1.2
Printing Shop	5	10	2.5	6,618	16.5	Mineral Wool	50	411	1.0	15.5
	5	4	9.3	3,081	28.7	Mineral Wool	40	246	2.3	26.4
	1.3	6	2	2,872	5.7	Mineral Wool	25	322	0.6	5.1
	1.3	4	4.5	2,068	9.3	Mineral Wool	25	238	1.1	8.2
	1.3	3.5	10	1,863	18.6	Mineral wool	25	216	2.2	16.5
	1.3	3	24	1,658	39.8	Mineral wool	25	195	4.7	35.1
Total					186.2				20.0	166.2

Method for calculation : See Guideline.

Preconditions for calculation:
1. The saturation temperature of the steam at the subject pressure was adopted as the internal temperature.
2. The room temperature is 25°C
3. The heat insulation thickness was determined with the economical thickness in Japan used as a reference.
4. The surfacre radiation rate is 0.9 at bare portions and is 0.5 at heat insulated portions.

Table 5.1.23 Insulation of Steam Valve

Room	Steam Press bar(G)	Valve			Heat Loss Present		Heat Loss after Insulation				Saved Energy MJ/h
		Dia inch	Number	Equi. L m	kJ/(mh)	MJ/h	Material	Thick mm	kJ/(mh)	MJ/h	
Washer & Cylinder Dryer	1.3	8	1	1.68	3654	6.1	Mineral Wool	25	407	0.7	5.5
	1.3	6	6	1.50	2872	25.8	Mineral Wool	25	322	2.9	22.9
	1.3	2	27	1.11	1176	35.2	Mineral Wool	25	146	4.4	30.9
	1.3	1	2	1.22	712	1.7	Mineral Wool	15	132	0.3	1.4
Printing Shop	5	10	1	1.68	6,618	11.1	Mineral Wool	50	411	0.7	10.4
	5	4	12	1.27	3,081	47.0	Mineral Wool	40	238	3.6	43.3
	1.3	6	6	1.50	2,872	25.8	Mineral Wool	25	322	2.9	22.9
	1.3	4	6	1.27	2,068	15.8	Mineral Wool	25	238	1.8	13.9
	1.3	3	3	1.25	1,658	6.2	Mineral wool	25	195	0.7	5.5
	1.3	2	6	1.11	1,176	7.8	Mineral wool	25	146	1.0	6.9
	1.3	1	4	1.22	712	3.5	Mineral wool	15	132	0.6	2.8
Total						186.2				19.6	166.5

Method for calculation : See Guideline.

Preconditions for calculation:
1. The saturation temperature of the steam at the subject pressure was adopted as the internal temperature.
2. The room temperature is 25°C
3. The heat insulation thickness was determined with the economical thickness in Japan used as a reference.
4. The surfacre radiation rate is 0.9 at bare portions and is 0.5 at heat insulated portions.

b. Selection and maintenance of steam traps

Traps are mounted at the bottom of a riser pipe, before a reducer valve or another automatic control valve and also at a drain separator or alike, with the objective to prevent inflow of the condensate into equipment and to prevent water hammer. But they will trigger steam loss unless their management is sufficient.

Traps of thermodynamic type are used in this factory, but traps of this type are unsuitable for the points where recovery of the condensate is made because maloperation occurs when the back pressure is high. It is recommended that they are replaced with traps of mechanical type.

In the case where traps are installed outdoors, sufficient attention should be paid to the measures for preventing freezing.

Insert short pipes as shown in Figure 5.1.27 so as to facilitate separation of the condensate for mounting traps.

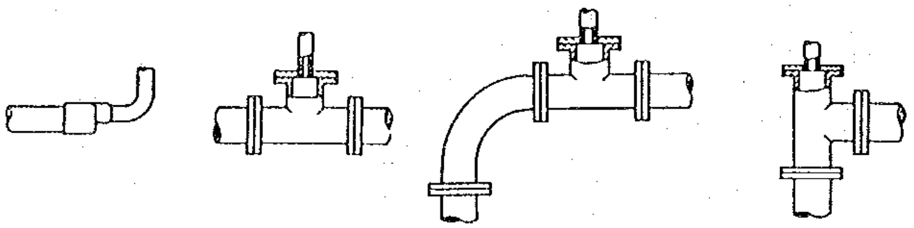


Figure 5.1.27 Method for discharge of condensate

When traps are used for a long time, the following troubles tend to occur due to wear to internal valves, valve seats, etc.

① Blow-off : Caused by faulty differential pressure or faulty valve or other moving parts

② Plugging : Plugging of the strainer (with rust, scale)

③ Steam leakage : Caused by flaws on valve, valve seat or float

Particularly at points where recovery of the condensate is made, discovery of blow-off or steam leakage is not made early enough, the back pressure rises in the recovery line, and adverse effect tends to occur over temperature control of other equipment.

Traps should be always kept in good conditions so as to maintain running of steam using equipment at high efficiency. For this objective, it is necessary to periodically inspect traps to check the operating conditions. Inspection of the operation can be made using the following means.

① Site glass

② Stethoscope

③ Test valve

④ Pressure gauge

⑤ Thermometer

Periodic patrol inspection is necessary with the traps for particularly important equipment.

(6) Power receiving, distribution and electrical equipment

A) General description of factory electrical equipment

There are two systems, i.e., purchased power and power generated in the factory. They are used independently.

The purchased power is received from a 10kV distribution line of the electric power company, and is distributed in the factory after stepdown to 380V with a transformer of 1,600kVA. Two transformers are provided, but No. 1 transformer is usually bearing the load.

On the other hand, the private power generation plant has two generators, i.e., generator G_1 of 3,000kVA and generator G_2 of 1,200kVA. But G_1 is usually run. It is driven by a back pressure turbine, and its exhaust is used for processes. It runs independently with the system of the electric power company, due to the fact that no synchronism detection unit is provided. When the steam demand is not matched with the generation rate, the excess steam is abandoned and is wasted.

The bus on the 380V side is divided into two systems, and generator G_1 is used while No. 1 transformer is running, or generator G_2 is used while No. 2 transformer is running. Switching means are provided for each load so that the electric power can be received from either system.

No single line diagram was found at the factory, but it is estimated roughly to be as shown in Figure 5.1.3.

The contracted power and power rates are shown in Table 5.1.24, and the situations of power consumption measured for a short length of time are shown in Table 5.1.25. The receiving electric energy on the day of measurement was considerably higher than the contracted electric energy.

Table 5.1.24 Contracted power and power rates

Time		Contract kW	Demand Charge Ft/kW	Energy Charge Ft/kWh
Day Time	6:00 - 16:30	750	400	2.60
Peak Time	16:30 - 21:00	250	700	5.50
Night Time	21:00 - 6:00	750	0	2.60

Reduction by Power Factor Improvement (% of Total Charge)

Power Factor	0.9/0.92	0.92/0.94	0.94/0.96	0.96/0.98	0.98/1.00
Reduction %	1	2	3	4	5

Table 5.1.25 Situations of power consumption

Time	Generator Side				Transformer Side			
	V	A	kW	P.F.	V	A	kW	P.F.
Aug 13 15:00	400	1,500	1,100	0.87	382	700	480	0.93
16:00	390	1,700	1,100	0.83	382	650	460	0.93
Aug 14 16:00	400	1,100	700	0.95	378	1,000	650	0.88
16:30	400	1,200	800	0.93	377	1,100	770	0.91
17:00	400	1,200	800	0.93	370	1,200	860	0.91
18:00	400	1,000	700	0.80	380	1,200	750	0.90
19:00	400	1,200	700	0.80	380	1,200	750	0.90
21:00	400	1,200	700	0.80	380	1,200	750	0.90
22:00	400	1,000	700	0.80	380	1,200	750	0.90
Average	400	1,230	810	0.86	380	1,050	690	0.91

Old motors are used at many places and traces of efforts for consolidation are observed. It is considered to be even better if a motor register that classifies and rearranges the motors in possession by type, by rated output and by revolution is prepared.

B) Countermeasures for improvement and their effect

a. Improvement of the operation of transformers

It was explained by the factory that two transformers are always impressed. But one transformer is sufficiently capable of covering the load as shown in Table 5.1.25.

As the no-load loss of the transformer of 1,600kVA is 5.38kW, the following loss can be reduced per year if both of primary side and secondary side of one transformer are opened up.

$$5.38 \text{ kW} \times 8{,}760 \text{ h} = 47{,}130 \text{ kWh/y}$$
$$47{,}130 \text{ kWh/y} \times 3.75 \text{ Ft/kWh} = 176{,}700 \text{ Ft/y}$$

b. Saving of lighting

The ceilings are low, and lighting equipment that mounts three 40W fluorescent lamps on one base is used. Selection of the light type is satisfactory.

Day light is brought in to a considerable extent at the materials warehouse and so forth, but there were many places where illumination is lit at unnecessary points.

The results of measurement of illuminance measured using an illumination meter are shown in Figure 5.1.28. While there are places where the luminous intensity is over 2,000 Lx, there also are places where the luminous intensity is as low as 250 Lx. The places where the luminous intensity is excessive are marked with ✻ in the drawing.

The reference of luminous intensity is as shown below in the Japanese Industrial Standards. (See the Guideline.)

Selection/inspection work in fabric industry:	750 ~ 1,500 Lx; 1,000 Lx at average
Ordinary work in general production process:	300 ~ 750 Lx; 500 Lx at average
Packing/packaging work:	150 ~ 300 Lx; 200 Lx at average
Passage, staircase, warehouse involving work:	75 ~ 150 Lx; 100 Lx at average

The factory buildings were divided into three sections, the number of lamps which may be put off was estimated for each section based on the standard indicated above, and is

The factory buildings were divided into three sections, the number of lamps which may be put off was estimated for each section based on the standard indicated above, and is indicated in the upper part of Figure 5.1.28.

The annual saving amount by put-off of lamps is as follows.

$\{40\ W \times 3 \times (72 + 56 + 34) + 400 \times 7\} \times 4{,}800 = 106{,}700\ kWh/y$
$106{,}700\ kWh/y \times 3.75\ Ft/kWh = 400{,}100\ Ft/y$

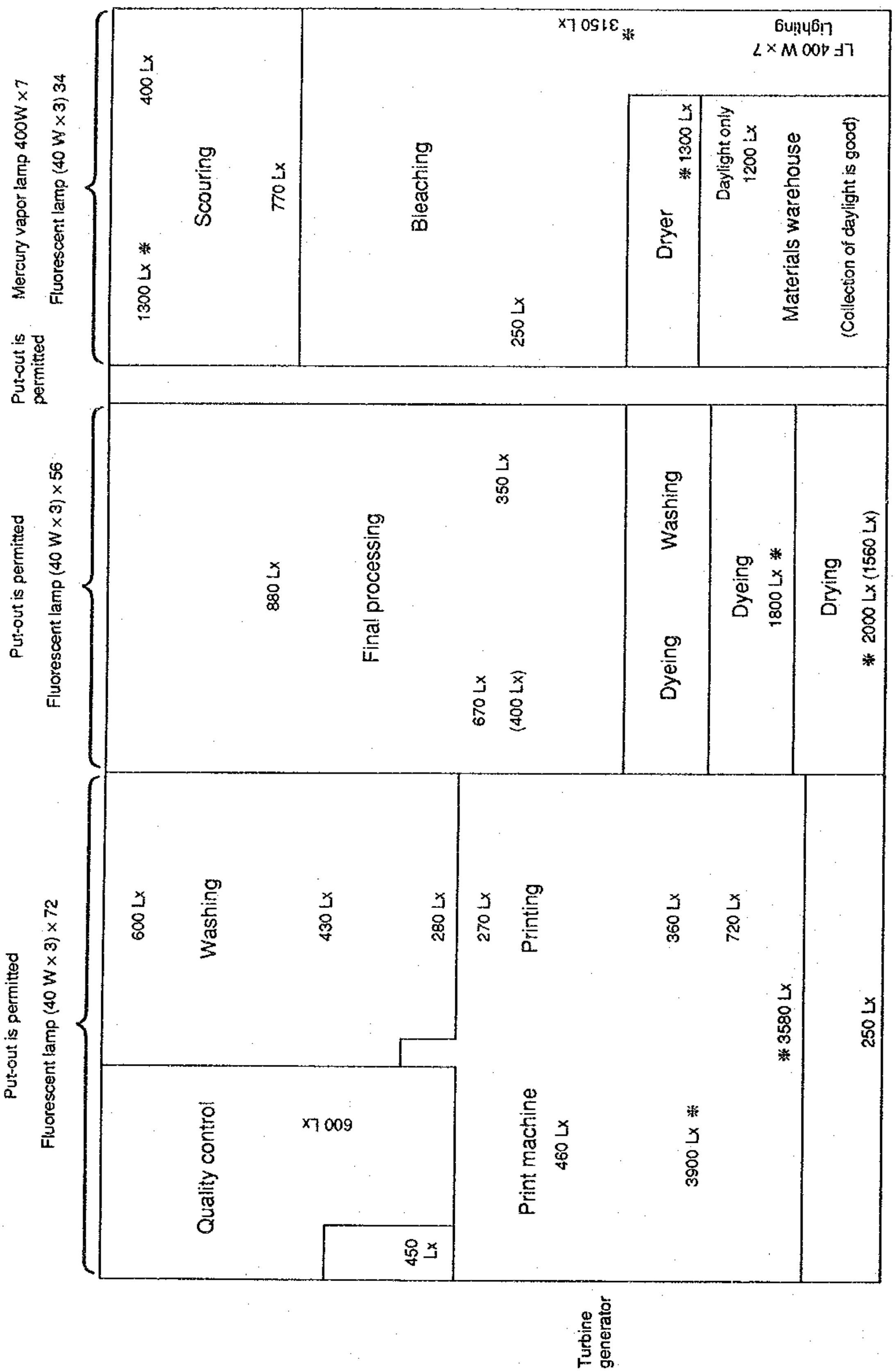


Figure 5.1.28 Illuminance in buildings

c. Improvement of operation of compressors

The situations of operation of the central compressor are shown in Table 5.1.26.

Table 5.1.26 Situations of power consumption of central compressor

Time Aug 15	Press bar	Input kW	kVA	kvar	P.F.	A
10:15	6.0	34.1	46.1	31.0	0.740	75.1
10:30	6.65	34.2	46.2	31.0	0.741	75.3
10:45	6.6	34.4	46.3	31.0	0.742	75.4
11:00	6.6	34.4	46.3	31.1	0.742	75.5
11:15	6.6	34.4	46.3	31.1	0.742	75.5
11:30	6.55	34.5	46.4	31.1	0.743	75.6
11:45	6.6	34.5	46.5	31.1	0.743	75.7
12:00	6.45	34.6	46.6	31.1	0.744	75.9
13:30	6.6	35.2	47.1	31.3	0.747	76.7
13:45	6.6	35.2	47.2	31.4	0.747	76.8
Avg	6.53	34.55	46.50	31.12	0.743	75.75

a) Drop of delivery pressure

The delivery pressure is over 6 bar. But the pressure required on the load side such as the washer is up to 4 bar in most cases, and accordingly, it is reduced to 4 bar or less for use on the working machine side. Local compressors are provided at places related to printing, etc. requiring pressure of 5 ~ 6 bar.

Since the pressure drop along the 2" mains is minor, it is estimated that the delivery pressure of the central compressor may be reduced by about 1 bar. It is recommended to repeat such an attempt that the pressure is reduced by about 0.1 bar, it is confirmed that there is no problem on the load side and the pressure is further reduced.

The electric energy can be saved by about 4% when the delivery pressure is reduced by 1 bar. (See the Guideline.) Therefore, if the mean input to the central compressor is 34.5kW, the effect obtained by the reduction of the delivery pressure is as follows.

$34.5 \text{ kW} \times 4{,}800 \text{ h/y} \times 0.04 = 6{,}600 \text{ kWh/y}$
$6{,}600 \text{ kWh/y} \times 3.75 \text{ Ft/kWh} = 24{,}800 \text{ Ft/y}$

b) Reduction of air intake temperature

The temperature of the central compressor room was 30°C at the time of investigation while the outdoor air temperature was 20°C. When the air intake temperature rises, the density drops and the actual airflow that can be sucked with the same power decreases. It is possible to reduce the air intake temperature if a hole is produced in the wall of the intake side to permit direct suction of the outdoor air.

The power can be saved by about 3.3% when the air intake temperature is reduced by 10°C. (See Guideline.)

The effect obtained by reduction of the air intake temperature is as follows.

$$34.5 \text{ kW} \times 4{,}800 \text{ h/y} \times 0.033 = 5{,}500 \text{ kWh/y}$$
$$5{,}500 \text{ kWh/y} \times 3.75 \text{ Ft/kWh} = 20{,}600 \text{ Ft/y}$$

c) Improvement of power factor

The power factor is as low as a little less than 75% because the compressor is running with the load of around 60% against the rated output of 55kW. When the power factor is low, the current that flows to the cable increases against the same input, and the power loss caused by resistance increases.

The power factor is improved to 99.9% if a condenser of 3-phase, 30kvar is connected in parallel with the 55kW induction motor that drives the compressor.

Reduction of the cable loss obtained by improvement of the power factor is as follows.

The current after improvement becomes 56.4A if the current before improvement is 75.8A, the power factor before improvement is 0.743 and the power factor after improvement is 0.999.

$$75.8 \times 0.743/0.999 = 56.4 \text{ A}$$

Annual reduction of the resistance loss is calculated as follows when the conductor resistance of 3C/60 mm² CV cable is assumed as 0.408Ω/km and when the cable length is assumed as 250 m.

$$3 \times (75.8^2 - 56.4^2) \times 0.408 \times 250/1{,}000 \times 4{,}800/1{,}000 = 3{,}800 \text{ kWh/y}$$
$$3{,}800 \text{ kWh/y} \times 3.75 \text{ Ft/kWh} = 14{,}200 \text{ Ft/y}$$

The expenses for purchase and installation of a condenser of 30kvar are considered to be around ¥100,000 or 50,000 Ft, and the amount of investment can be recovered in four years.

d. Improvement of receiving power factor

The power factor at the receiving point is 0.91 at average as shown in Table 5.1.25. The loss of the electric power company in the distribution system decreases when the power factor at users is improved, so incentive for improvement of power factor is provided as shown in Table 5.1.24.

The capacity of the condenser to be connected for improving the power factor from 0.91 to 0.98 against the contracted power of 750kW is calculated as follows.

$$750 \times \left(\frac{\sqrt{1-0.91^2}}{0.91} - \frac{\sqrt{1-0.98^2}}{0.98}\right) = 189 \text{ kvar}$$

Connect four condensers of 50kvar to the secondary side of the transformer.

The reactive power after improvement decreases to 142 kvar, and the power factor is improved to 0.98.

$$750 \times \frac{\sqrt{1-0.91^2}}{0.91} - 200 = 142 \text{ kvar}$$

$$\frac{750}{\sqrt{750^2 + 142^2}} = 0.982$$

The expenses for purchase and installation of these condensers and switches are considered to be around ¥800,000 or 400,000 Ft. When the annual power consumption is assumed as 2,700,000kWh/y, the increased amount of the incentive to be granted as a result of improvement of the power factor is as follows.

2,700,000 kWh/y × 3.75 Ft/kWh × (0.05 – 0.01) = 405,000 Ft/y

Another effect brought by improvement of the power factor is reduction of the load loss at the transformer. The reduction of load loss and the saving amount are as follows when the power factor is improved from 0.91 to 0.98, if the copper loss at the time of full load of the transformer of 3-phase, 1,600kVA is assumed as 21kW and the mean load is assumed as 690kW (see Table 2.1.25).

$$21 \times \left\{ \left(\frac{\frac{690}{0.91}}{1{,}600} \right)^2 - \left(\frac{\frac{690}{0.98}}{1{,}600} \right)^2 \right\} \times 4{,}800 = 3{,}120 \text{ kWh/y}$$

$$3{,}120 \text{ kWh/y} \times 3.75 \text{ Ft/kWh} = 11{,}700 \text{ Ft/y}$$

Saving of 416,700 Ft/y can be achieved by the increase of the incentive and by reduction of the loss of the transformer. Thus the equipment expenses for the condensers can be recovered in a little less than a year.

(7) Total of effect of improvement

Item	Expected Saving										Invest-ment	Payback Year
	Fuel Gas			Power			Water			Total		
	Nm^3/y	1,000Ft/y	%	kWh/y	1,000Ft/y	%	m^3/y	1,000Ft/y	%	1,000Ft/y	1,000Ft	y
Cylinder Dryer												
Steam Press Stabilization	0									0	80	
Elimination of Steam Leak	29,700	401	0.2							401	0	0.0
Steam Trap Arrangement	50,600	683	0.3							683	420	0.6
Condensate Recovery	63,800	861	0.4				4,600	91	0.3	952	500	0.5
Washer												
Water Flow Control							69,100	1,369	3.9	1,369	220	0.2
Waste Heat Recovery	180,800	2,441	1.1							2,441	2,100	0.9
Insulation of Washer	14,900	201	0.1							201	250	1.2
Frame Dryer												
Damper Control	347,000	4,685	2.1							4,685	0	0.0
Waste Heat Recovery	19,400	262	0.1							262	500	1.9
Boiler												
Air Ratio Control	397,300	5,364	2.4							5,364	2,200	0.4
Burner Adjustment	10,600	143	0.1							143	0	0.0
Condensate Recovery	237,300	3,204	1.5				18,300	1,047	1.0	4,251	0	0.0
Steam Line												
Insulation of Pipe	192,200	2,595	1.2							2,595	1,300	0.5
Electric Power												
Operation of Transformer				47,130	177	0.5				177	0	0.0
Saving of Lighting				106,700	400	1.0				400	0	0.0
Compressor Press Reduct'n				6,600	25	0.1				25	0	0.0
Compressor Suction Temp				5,500	21	0.1				21	0	0.0
Compressor Power Factor				3,800	14	0.0				14	50	3.6
Receiving Power Factor				3,120	417	0.0				417	400	1.0
Total	1,543,600	20,840	9.5	172,850	1,054	1.7	92,000	2,507	5.2	24,401	8,020	0.3

5.2 Results of investigation at a tyre factory

5.2 Results of investigation at a tyre factory

5.2.1 Outline of the factory

(1) Company name and factory name: TAURUS Hungarian Rubber Work
Abroncs Igazgatóság Nyíregyházi Gumigyár

(2) Category of business: Rubber industry

(3) Principal product name and production capacity

Principal products : Agricultural Tyre, Inner Tube, Bellows

Production capacity : 20,000 t

(4) No. of employees : 1,280

(5) Location of factory : 4401 Nyíregyháza, Derkovits u. 107

(6) History of the factory

This is the only tyre factory in the nation. Foundation of the company was as early as in 1883, but it was early in the 1960's when the company intensified its business in the rubber industry.

A division-oriented system in which authorities are given steeply to each division was introduced to flexibly cope with transition to the market economy. The tyre division takes charge of production and sales of tyres for trucks, for agricultural use and so forth. It runs two factories, one of which, this factory, was constructed in 1964. It is now producing tyres for agricultural machinery, fork lift tyres, inner tubes and bellows. It is also performing retreading of tyres. The tyre department has an 81% share of the sales of this factory.

The operations of the tyre department were commenced in 1979. The product types were increased year after year, and tyres of 48 different sizes are produced at present. The products are also exported to the U.S.A. and Australia as well as European nations.

A modern calendering facility was recently completed with a fund from the World Bank. In addition, the introduction of a total computer control system is in progress aiming at completion in the fall of 1991.

The factory does not have boilers, and purchases steam from the plant of Regional Heat Supply and Power Generation Public Corporation located about 2 km away.
As the atmospheric temperature drops to -20°C or less during winter, about 25% of the steam is consumed for heating. The ratio of the energy cost to the production cost is 6% to 7%.

According to the agreement at the time of signing of the Scope of Work, the scope of investigation at this factory is limited to the tyre curing process. Strengthening of heat insulation, improvement of the system for extraction of the condensate from the curing presses, introduction of microprocessors, update of air compressor and so forth were effected as the energy conservation measures for the curing process up to the present time. The energy unit consumption was improved by about 25% in the past five years as a result.

(7) Investigation period August 21 - August 23, 1991

(8) Investigators

Mitsuo Iguchi	Leader
Teruo Nakagawa	Subleader, Measuring Engineer
Taro Ihara	Tyre Process Engineer
Koichi Inaba	Heat Control Engineer
Tatehiro Tanabe	Heat Control Engineer
Toshiyuki Ochi	Heat Control Engineer
Ken-ichi Kurita	Electrical Control Engineer

AEEF Member

Mr. János Becz	Team Leader
Mr. Ferenc Pardavi	Electrical Enginner
Mr. József Stieber	Instrument Engineer
Mr. Gyula Petró	Electrical Engineer

MVMT Member

Mr. Lajos Roppolyi	Mechanical Engineer
Mr. Miklós Kenézy	Electrical Engineer
Mr. Zoltán Dudás	Electrical Technician

(9) Interviewees

Mr. László Jaczkó	Factory Director
Dr. József Orosz	Deputy Factory Director Finance
Mr. György Tanka	Deputy Factory Director Production
Mr. Erika Molnár	Manager Technical
Mr. László Rimanóczy	Advisor
Mr. Bottyán István	Dipl. Engineer & Economist
Mr. Ferenc Filetót	Heat Engineer
Mr. Jenö Juhász	Assistant Manager
Mr. János Répási	Chief Manager

(10) Trend of production (Table 5.2.1)

Name of Product	Unit	1986	1987	1988	1989	1990
Agricultural Tyre	t	16,543	16,142	16,803	17,935	17,651
Radial	t	11,227	11,383	12,744	12,378	14,519
Bias	t	4,298	3,467	2,684	3,812	1,733
Implement	t	1,018	1,292	1,375	1,745	1,399
	1000pc	223	233	242	271	247

(11) Trend of energy consumption (Table 5.2.2)

	Unit	1986	1987	1988	1989	1990
Fuel Oil	t	22	29	43	32	38
Kerosene	t	22	19	15	12	14
Natural Gas	m^3	32,000	41,000	45,000	37,000	43,000
Steam	t	123,000	118,000	106,300	101,700	93,300
Power	GWh	24.1	23.3	19.4	15.8	17.5
City Water	t	130,000	119,000	125,000	123,000	139,000
Total	TJ	460	438	389	366	365

(12) Trend of energy unit consumption (Table 5.2.3)

	Unit	1986	1987	1988	1989	1990
Agricultural Tyre	MJ/t	27,806	27,134	23,150	20,407	20,678
Radial	MJ/t	28,770	28,198	23,777	21,248	21,214
Bias	MJ/t	26,758	25,958	22,353	19,150	19,041
Implement	MJ/t	21,612	20,896	18,908	17,191	17,153

(13) Operating hours (Table 5.2.4)

	1986	1987	1988	1989	1990
Annual Operating Hours	6,072	6,168	6,168	6,096	6,072

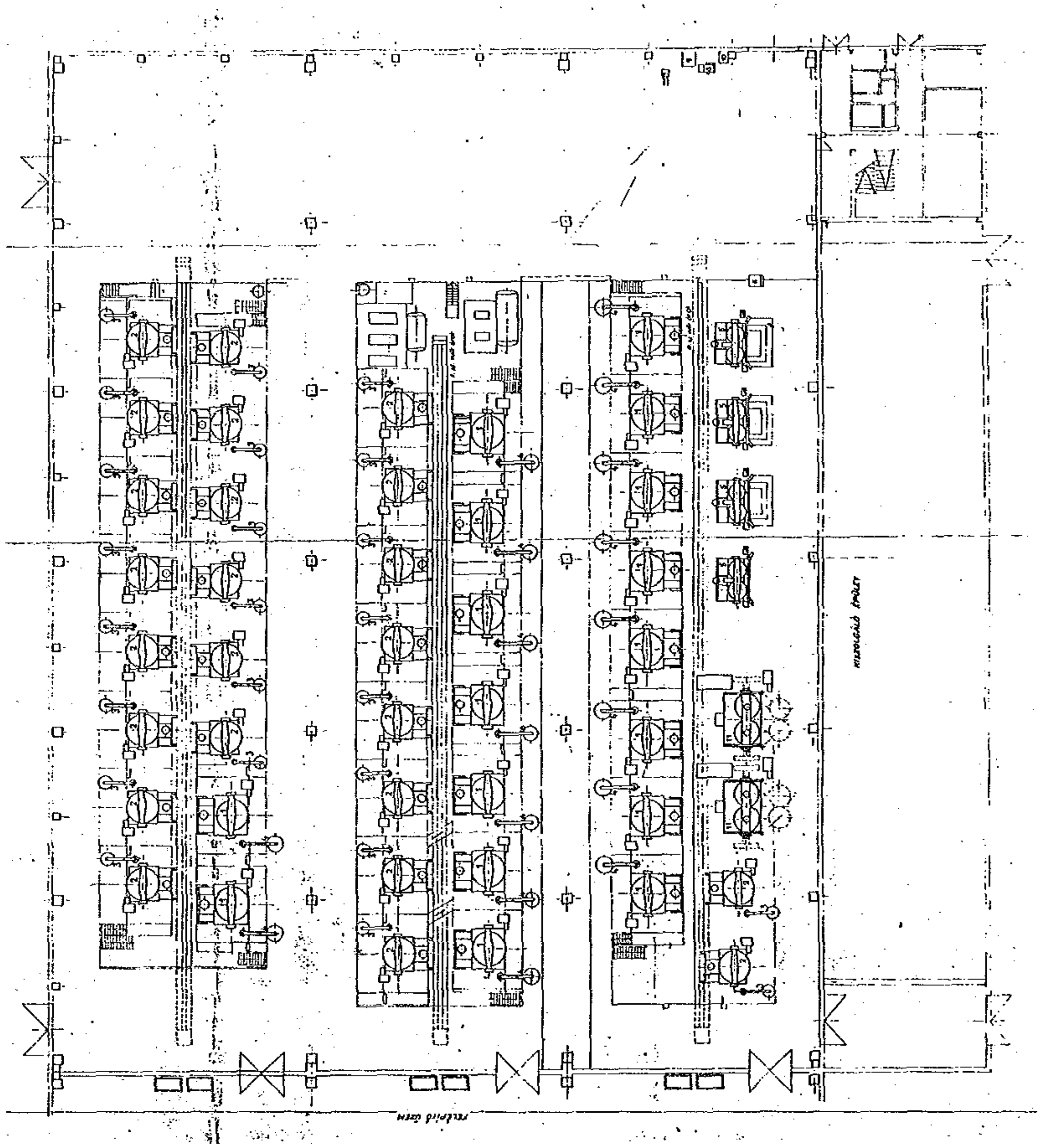
Reference operating hours for examination of countermeasures

24 hours/day × 330 days/year = 7,920 hours/year

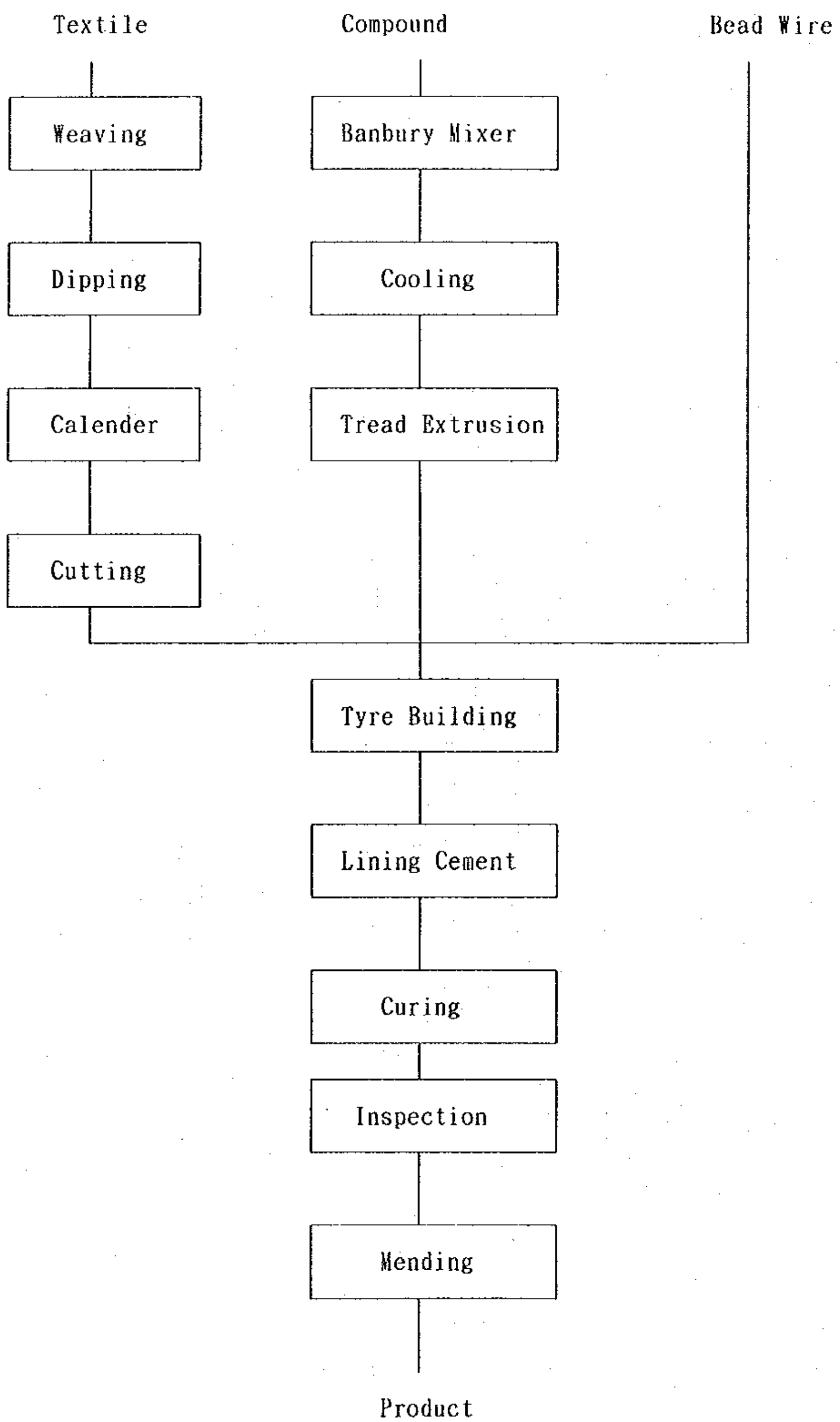
(14) Energy prices

Electric Power	3.69 Ft/kWh
Steam	750 Ft/t

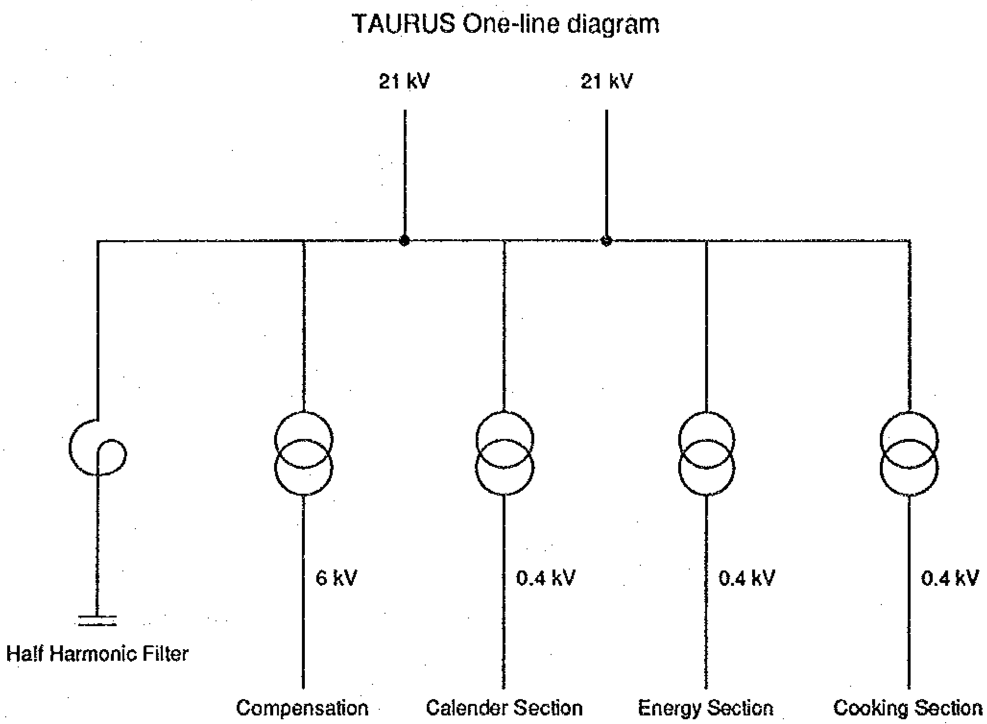
(15) Factory layout drawing (Figure 5.2.1)



(16) Manufacturing processes (Figure 5.2.2)



(17) Electric power one line diagram (Figure 5.2.3)



Fugure 5.2.3

(18) Outline of principal equipment (Table 5.2.5)

Name	Numbers	Specification	Manufacturer
Curing	6	55" Dual Mold Dome type	SKODA (CZECHOSLOVAK)
Press	29	75", Single Mold Dome type	BOLSEVIC KIEV(USSR)
		Diameter 1,828 mm	
		Flange Height 635 mm	
		Dome Height 1,200 mm	
		Dome Weight 4,270 kg	
		Mold Weight 8,675 kg	
	13	88", Single Mold Dome Type	BOLSEVIC KIEV (USSR)
		Diameter 2,240 mm	
		Flange Height 790 mm	
		Dome Height 1,500 mm	
		DomeWeight 7,810 kg	
		Mold Weight 12,399 kg	

5.2.2 Situations of energy management

(1) Setting the target of energy conservation

With agricultural tyres, the energy consumption per ton of products was reduced by about 26% during the 1986-1990 period. The steam condensate discharge system for the curing presses was improved and microprocessors were installed in the mean time. As a result these measures contributed to the improvement of the unit consumption. Measures for facilities for energy conservation have been positively made as represented by these facts.

However, no long-term energy conservation target value for the factory has been established. The only thing done is to incorporate the planned value as the energy consumption plan at the time of planning the annual production program.

Company-wide energy conservation activities should be evolved in order to allow as many employees as possible to participate in the energy conservation activities to make full use of their capabilities. For this objective, the top management, first of all, needs to appeal to all the employees that energy conservation has important significance in the management and that it is a matter of the issue that should be taken seriously in the company. It is also important that the top management indicates a concrete target and asks for cooperation to hit this target. Only when the policy of the business is clearly indicated and the target value of each workshop is set up based on the said policy, the employees will become conscious. Then they try to find problems in their concerned work or facilities, and will start examination of countermeasures for improvement for energy conservation actions.

(2) Systematic activities

It is desirable that a dedicated department in charge of practical affairs of energy management be provided, that may keep attention to the situations of energy consumption at the factory, give advice to the factory superintendent as required, and plan training and various events. A section in charge of energy management is organized in the power department of this factory, and ten staff members are working in this section.

To proceed with energy conservation activities participated by all, it is important to promote mutual understanding inter-sectionally and among the factory personnel so that everybody shares the common recognition. Key staff meeting attended by the factory superintendent is held every week and manager meeting is held periodically at this factory. Issues related to energy are pointed in dispute at these meetings, and information is exchanged. During the investigation of this time many concerned persons attended the meeting held to report the results of the investigation.

(3) Management by data

It is basically essential to seize the realities of consumption of energy, to compare them with the plan, and to take corrective measures against problems, if any. The result data of the factory is an important information source for improvement of energy unit consumption because problems in the energy consumption and the effect of energy conservation measures become clear only through analysis of data.

Energy consumption at principal processes is continuously measured at this factory. It is input to computers and analyzed daily. If any abnormality is recognized as a result, troubleshooting and corrective actions are immediately requested to the field. Result values and unit consumption of energy consumption are rearranged in the form of graphs, and are disclosed to the employees.

As the operation of each curing press is entirely recorded with computers, they can also be used for analysis of energy consumption besides quality control.

It is scheduled to introduce an energy management system using computers at this factory, and preparations are in progress aiming at introduction in the fall of 1991.

(4) Enlightenment of employees

Provision of sufficient information is indispensable to urge self-motivated activities of the employees. It is necessary to motivate the employees by providing information such as trend of energy prices, weight of energy cost in the production cost and cases of success at other factories. It is also essential to enlighten the employees to master the basic technology through training, distribution of manuals and so forth.

The engineers of this factory are trained through training courses held by the company and as dispatched to outside training courses held by public organs including State Authority for Energy Management and safety (AEEF) a number of times a year. Training of operators is not conducted in particular. But since cooperation of operators is essential for energy conservation activities, it is required that training of operators by the staff will be done in the future. A scheme to urge the employees to make proposals for improvement has been established, and rewards are given for adopted excellent proposals. Thus, this scheme is well utilized for motivation of energy conservation.

It can be said that the situations of energy management at this factory are good in general as stated above.

(5) Management of equipment

As described later, steam leakage from the curing press & steam trap, and heating troubles in steam piping were observed. Underground steam piping in this factory is so complicated with very short piping intervals that it makes inspection and repair difficult. Under the current situation, it seems difficult to improve the foregoing troubles; for their future modifications, however, maintainability should be first taken into consideration.

It will be also necessary to explicitly provide the chief supervisor of the process line with the authority to maintain the equipment, and thereby to allow him to take a prompt action for such a minor trouble as steam leakage.

5.2.3 Problems in the use of energy and countermeasures

(1) Curing presses

A) Situations of steam consumption

a. Results of measurement

About a half of the thermal energy used at a tyre factory is consumed in the curing process. The majority of the thermal energy is lost by heat radiation from equipment and pipelines and by discharge to the atmosphere and waste water. It is said in general that only a few percent of the thermal energy is effectively used for producing products. What is shown in Figure 5.2.4 is a typical "heat balance diagram" that indicates the outline of the structure of thermal energy consumption at a tyre factory. The situations stated above can also be read from this diagram. The total man-hour required for draw-up of such a detailed heat balance diagram is over 1,000 man-hours. It is recommended that the structure of thermal energy consumption is clarified at this factory by such a method and that countermeasures are taken to eliminate wasting of energy.

Since there was a time limit for the investigation of this time, measurement of the following items was taken with two curing presses, and attempts were made to calculate the net and gross energy required for production of products.

Steam flow rate to inside bladder and to outside dome
Condensate flow rate in outside dome
Curing press surface temperature

The measuring instrument mounting points are shown in Figure 5.2.5.

The tyre size, type, weight and curing conditions by the measured curing press are shown in Table 5.2.6.

The steam flow rate was measured using vortex flowmeters. The variation of flow rate was large, the maximum flow rate was nearly twice as much as the measuring span, and the length of time of low flow rate is also long as shown in Figure 5.2.6. Steam flowmeters are of low accuracy in the low flow rate area in general. And also with vortex flowmeters, about 4% of the measuring span is the lower limit of measurement. It is anxious, therefore, that the accuracy of the measured values obtained with vortex flowmeters this time is not necessarily good. But problems of this kind are not involved in the case where the condensate measuring methods are adopted.

With #37 curing press, it appears that there were errors in the setup of the flowmeter and abnormal values were output. It was therefore unavoidable and our great regret that the data on this curing press are excluded from the final data.

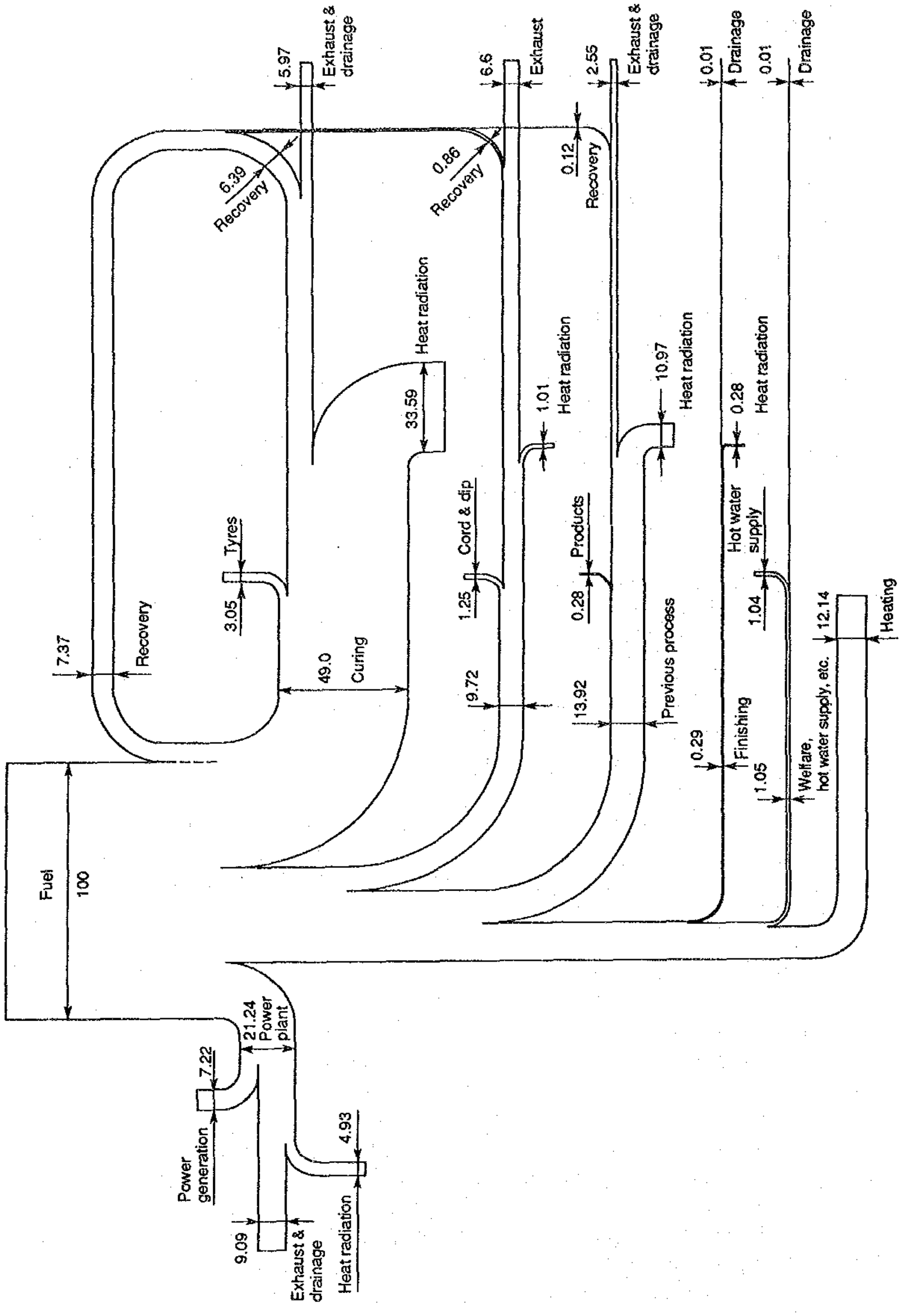


Figure 5.2.4 An example of heat balance chart

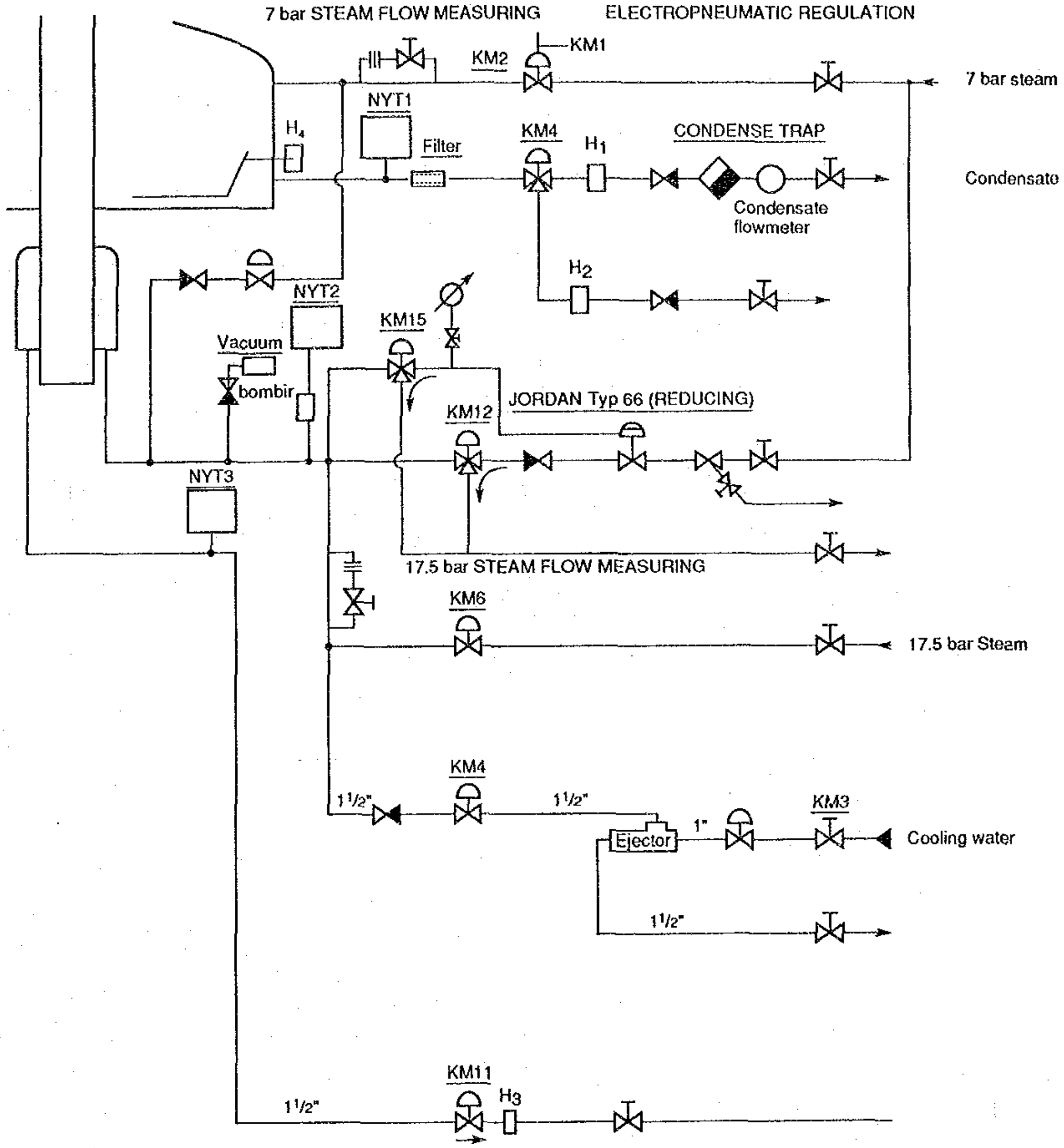


Figure 5.2.5 Measuring points

Table 5.2.6 Curing conditions

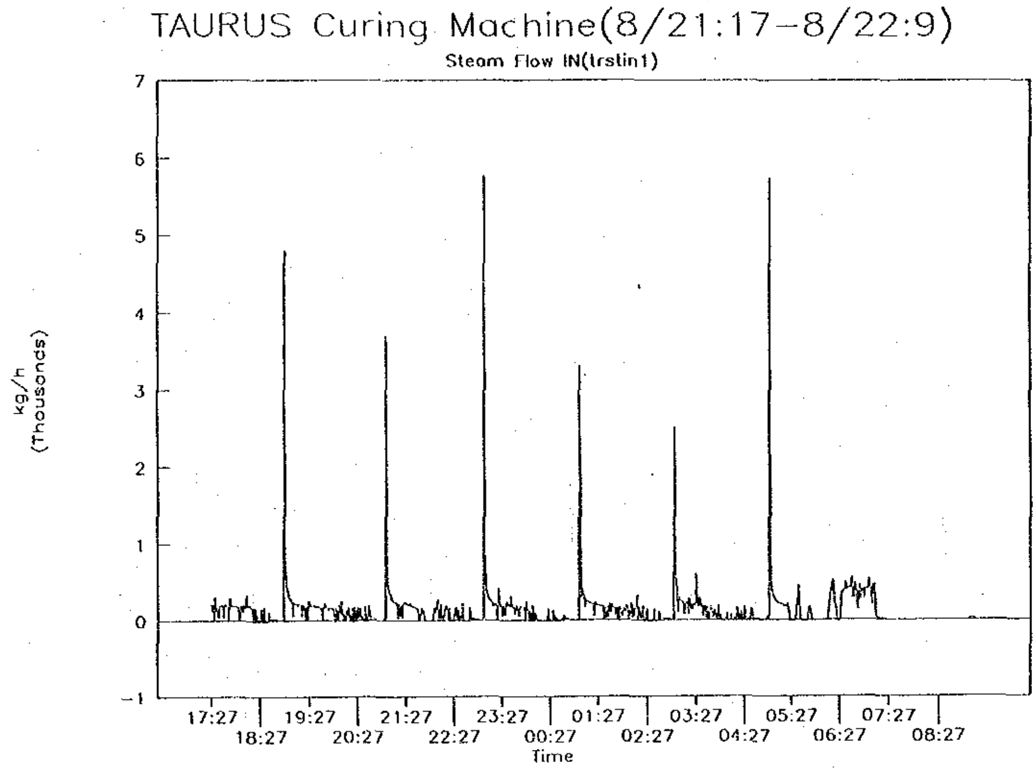
#47 Press

Tyre Size Tyre Weight	AGR 30.5R-32 319 kg	
Curing Cycle		
Time (min)	Outside Dome	Inside Bladder
0		Shaping (0.7-1.0-1.3 bar) Steam Supply
5	Steam Supply	
8	Ventilation	
102	Steam cut	
104		Exhaust
108		Vacuum
112	Press Open	

#37 Press

Tyre Size Tyre Weight	AGR 16.9R-30 175 kg	
Curing Cycle		
Time (min)	Outside Dome	Inside Bladder
0		Shaping (0.5-0.8-1.1 bar) Steam Supply
5	Steam Supply	
7	Ventilation	
71	Steam cut	
74		Exhaust
76		Vacuum
80	Press Open	

A graph of #47 curing press steam flow rate (inside bladder) is shown in Figure 5.2.6, and a graph of #47 curing press steam and condensate flow rates (outside dome) are shown in Figure 5.2.7. Furthermore, the steam consumption of #47 curing press is shown in Table 5.2.7.



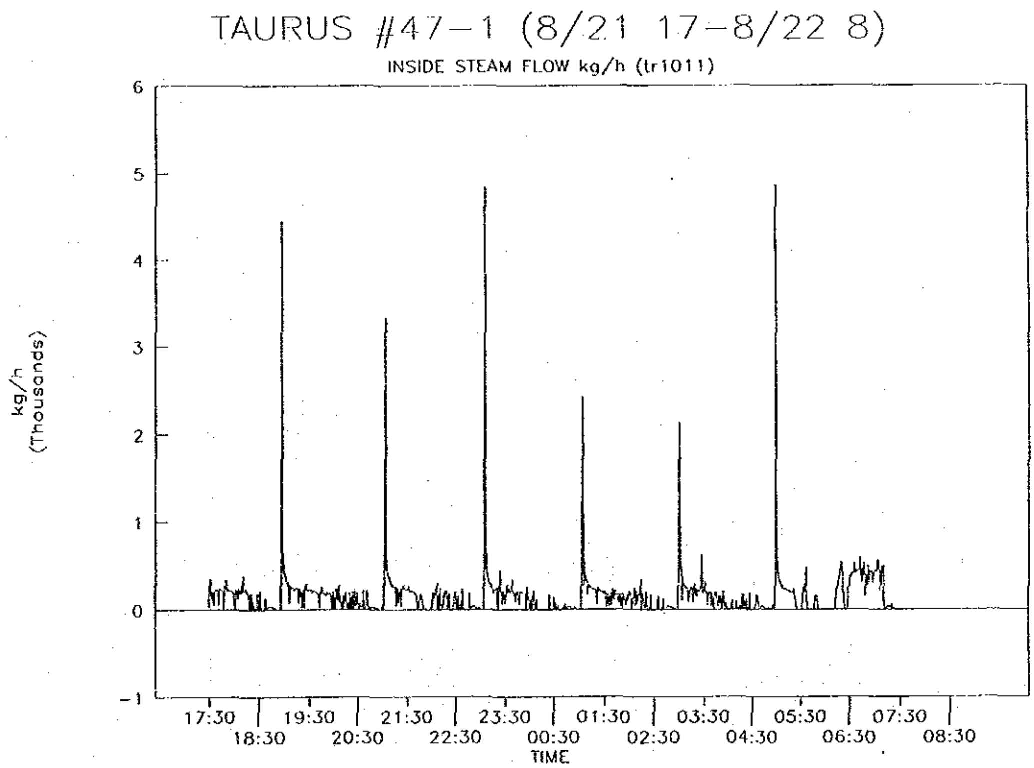
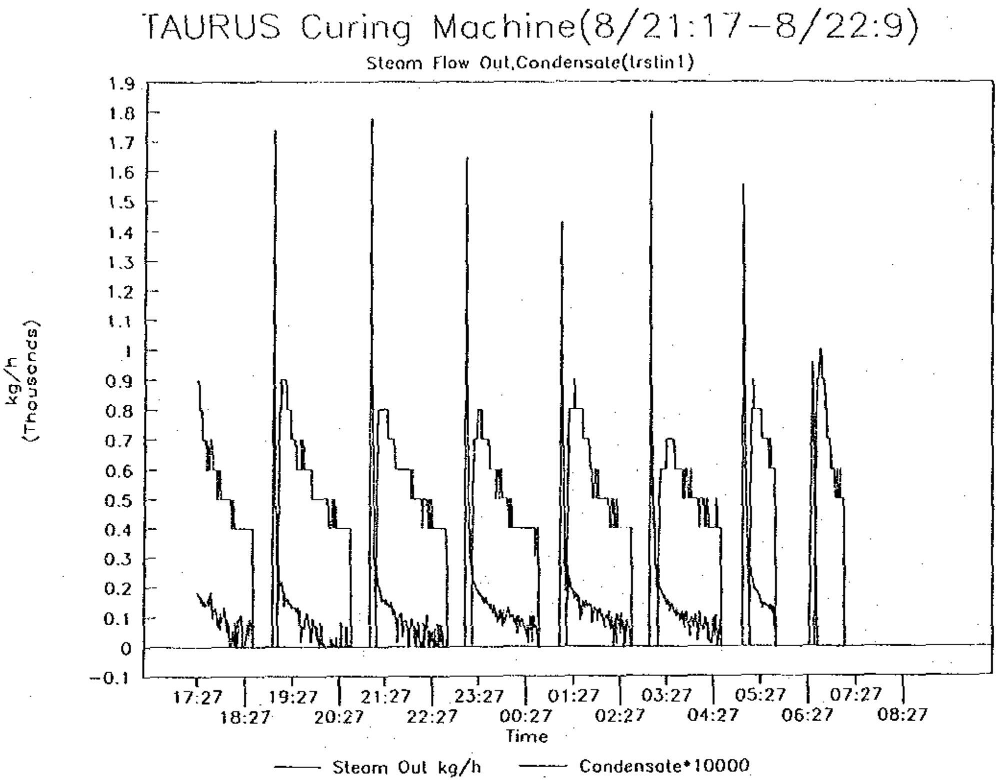


Figure 5.2.6 #47 curing press steam flow rate (inside bladder)



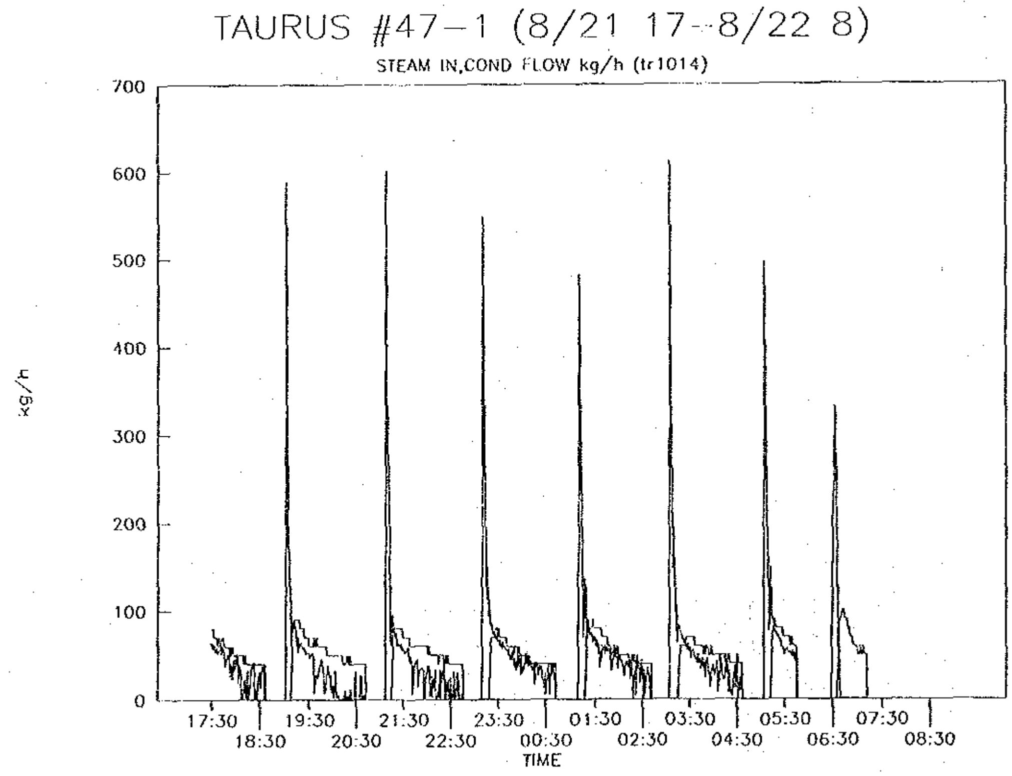


Figure 5.2.7 #47 curing press steam and condensate flow rates (outside dome)

Table 5.2.7 #47 curing press steam consumption

Date	Time	Cycle Time (min)	Interval (min)	Steam Flow (kg)		Condensate (kg)
				Inside	Outside	
8/21	18:56-20:45	108	-	325	68	88
	21:02-22:48	106	16	264	76	84
	23:04- 0:45	101	15	266	95	74
8/22	1:04- 2:44	100	18	258	89	79
	2:59- 4:39	100	14	216	64	72
	4:58- 7:12	134	18	520	97	80
	9:15-11:07	112	-	493	145	211
	11:31-13:22	111	23	398	72	100
	15:06-17:03	117	-	487	146	-
	19:18-20:58	100	-	362	88	86
	21:14-22:55	101	15	358	84	84
	23:10- 0:50	100	14	298	82	73
8/23	1:06- 2:47	101	15	407	95	80
	3:01- 4:41	100	13	373	89	72
	4:57- 6:38	101	15	391	97	-
Average 1		106	16	361	93	91
Average 2		104		350	92	92

Note (1): The length of time during which the steam pressure on the inside bladder side was 1 bar(G) or higher was used as the cycle time because no running record is available.

Note (2): Average 2 is the average value excluding the data of 8/22 4:58-7:12 when the cycle time was abnormally long. This value was used in the subsequent calculation.

b. Thermal efficiency and unit heat consumption rate

The ratio of the quantity of heat consumed for heating tyres was calculated from Table 5.2.7 and the results are shown in Table 5.2.8. It is learned that the net heat energy required for production of tyres is 5% or less of supplied quantity of heat.

Table 5.2.8 Quantity of heat for heating tyres

Heat Input				
Steam		Bladder Side	Dome Steam	Total
Pressure	bar (G)	16.5	5	
Flow Rate	kg	350	92	
Enthalpy	kJ/kg	2,794	2,756	
Heat Input (A)	kJ	977,600	253,600	1,231,100
Heat Gain of Tyre				
Weight	kg	319		
Specific Heat	kJ/(kg·°C)	1.38		
Initial Temp	°C	25		
Final Temp	°C	157		
Heat Gain (B)		319 kg × 1.38 kJ/(kg·°C) × (157 – 25)°C = 58,100 kJ		
(B)/(A)		58,100 / 1,231,100 = 4.7%		
Unit Heat Consumption Rate		(A) / 319 = 3,859 kJ/kg		

Note: The specific heat of tyres varies by the position because tyres are composite materials composed of rubber, fibers and steel wires. But the value of 1.38 kJ/(kg·°C) is used here for simplification.

c. Curing cycle time

Table 5.2.7 indicates the data measured and recorded on 15 cure cycles between 18:56 on August 21 and 6:38 on August 23. After returning to Japan, it was learned from the output of the data that the curing cycle time is dispersed to a considerable extent. Since the curing process is controlled with microcomputers and tyres of the same product type are cured, it cannot be considered that differences occur in the curing cycle time in such a manner. But since it is the cycle time judged from the steam pressure, the data were used as they are except for one extraordinarily long time.

d. Steam pressure

Changes in the steam pressure during curing cycles are shown in Table 5.2.9.

Table 5.2.9 Steam pressure

Date	Time	Steam Pressure kg/cm2				
		Inside Bladder			Outside Dome	
		Avg	Max	Min	Avg	Max
8/21	18:56-20:45	16.6	17.0	16.4	5.1	5.2
	21:02-22:48	16.6	17.1	16.4	5.1	5.3
	23:04- 0:45	16.5	17.0	16.2	5.2	5.3
8/22	1:04- 2:44	16.7	17.0	16.6	5.2	5.2
	2.59- 4:39	16.6	17.0	16.6	5.2	5.4
	4:58- 7:12	14.3	17.1	8.6	3.2	5.3
	9:15-11:07	16.5	17.0	16.5	5.1	5.2
	11:31-13:22	16.7	17.0	16.3	5.1	5.2
	15:06-17:03	16.6	17.0	16.6	5.2	5.3
	19:18-20:58	16.6	17.0	16.5	5.2	5.3
	21:14-22:55	16.5	17.1	16.5	5.2	5.3
	23:10- 0:50	16.6	17.0	16.5	5.1	5.3
8/23	1:06- 2:47	16.5	17.0	16.6	5.2	5.3
	3:01- 4:41	16.6	17.1	16.6	5.2	5.3
	4:57- 6:38	16.5	17.1	16.1	5.2	5.2

Note: The data obtained for the first 5 minutes and last 5 minutes of a cycle were excluded for the minimum pressure. The mean pressure on the low pressure side is the value for the stable period, and the minimum pressure was omitted.

The mean steam pressure is stable throughout a cycle, although variation of around ±0.5 bar was observed. There are cycles of extraordinarily long time in Table 5.2.7, and it is considered they are affected by the drop in the steam pressure.

B) Prevention of heat radiation

The results of measurement of surface temperature of curing presses are shown in Table 5.2.10.

Table 5.2.10 Curing press surface temperature

Press No		Surface Temperature (°C)					
		Dome 1	Dome 2	Dome 3	Dome 4	Avg	Frame
47	Avg Max	45.4 52.1	56.5 62.1	53.2 58.6	58.4 64.5	53.4 59.3	95.8 103.8
37	Avg Max	64.1 70.4	57.1 79.0	67.1 70.0	58.2 89.4	61.6 77.2	88.4 96.6

As is observed in THERMAL VIDEO SYSTEM "AVIO" and Table 5.2.10, the surface temperature of the curing presses themselves is 10 - 20°C higher than 40°C, which is the target of control at tyre factories in Japan. Immediate effect can be obtained when heat insulation is strengthened.

The materials used in general in Japan for strengthening heat insulation are ceramic fiber and calcium silicate. But other materials will do as long as they are effective as heat insulation materials.

Trial calculation of reduction of heat radiation by heat insulation is shown in Table 5.2.11. When the curing press surface temperature is assumed around 59°C, it is considered that heat radiation loss of about 40.3 MJ/h is occurring per press with the current heat insulation. It will become possible to reduce heat radiation by 8.3 MJ/h (20.6%) per curing press when heat insulation of bare portions (mold adjuster) and strengthening of heat insulation of entire top face of the curing press are implemented.

When the evaporation latent heat of 5 bar(G) steam is assumed 2,085 kJ/kg, the quantity and amount of steam to be saved are as follows.

8,300 kJ/h × 7,920 h/y / 2,085 = 31.5 t/y
31.5 t/y × 750 Ft/t = 23,600 Ft/y

The expenses required for this strengthening of heat insulation are about 70,000 yen (equivalent to about 35,000 Ft) per press in Japan. The expenses can be recovered in about one and a half years.

Thermal video of curing press

Thermal video of steam pipe line

Table 5.2.11 Insulation reinforcement for curing machine

Location	Steam Press bar(G)	Surface Area sq.m	Present Heat Loss					Heat Loss after Insulation					Saved Energy MJ/h	Note
			Material	Thick mm	Surface Temp.	kJ/(m² h)	MJ/h	Material	Thick mm	Surface Temp	kJ/(m² h)	MJ/h		
Upper Surface	5	3.35	Mineral Wool	15	58	1,213	4.1	Mineral Wool	40	39	524	1.8	2.3	
Upper Side	5	4.19	Mineral Wool	15	59	1,202	5.0	Mineral Wool	40	40	521	2.2	2.9	
Lower Side	5	2.78	Mineral Wool	15	59	1,202	3.3	Mineral Wool	15	59	1,202	3.3	0.0	
Bottom	5	4.15	None	0	158	5,912	24.5	None	0	158	5,912	24.5	0.0	
Mold Adjuster	5	0.38	None	0	158	8,873	3.4	Mineral Wool	40	39	524	0.2	3.2	
Total							40.3					32.0	8.3	

Method for calculation : See Guideline.

Preconditions for calculation:
1. The saturation temperature of the steam at the subject pressure was adopted as the internal temperature.
2. The room temperature is 20°C.
3. The heat insulation thickness was determined with the economical thickness in Japan used as a reference.
4. The surfacre radiation rate is 0.9 at bare portions and is 0.5 at heat insulated portions.
5. The representative length is 2.2 m.

The curing presses in use are of three different types. But if it is assumed that all of them are of the same conditions, steam saving by about 1,500 t/y can be anticipated in total with 48 curing presses.

31.5 t/y × 48 = 1,500 t/y

The heat insulation of the steam lines around the curing presses is not necessarily sufficient. With #47 curing press, the portion of 6.2 m out of the high pressure line of 21.7 m and the portion of 14.0 m out of the low pressure line of 26.8 m were not heat insulated (see AVIO photo).

The effect to be obtained when these bare pipes are heat insulated is shown in Table 5.2.12. The heat radiation rate will drop to about 1/8 when heat insulation of 25 mm is applied.

When the evaporation latent heat of 5 bar(G) steam is assumed 2,085 kJ/kg, the quantity and amount of steam to be saved are as follows.

27,600 kJ/h × 7,920 h/y / 2,085 = 104.8 t/y
104.8 t/y × 750 Ft/t = 78,600 Ft/y

The expenses required for this strengthening of heat insulation are about 50,000 yen (equivalent to about 25,000 Ft) in Japan. The expenses can be recovered in an extremely short length of time.

If it is assumed that all of the curing presses are of the same conditions, steam saving by about 5,000 t/y can be anticipated in total with 48 curing presses.

104.8 t/y × 48 = 5,000 t/y

C) Maintenance of steam traps

The operation of steam traps mounted to curing presses was checked using trap testers. The result was not necessarily satisfactory. The traps are mounted at ends of outside dome steam outlets, and they are of thermostatic type (KIPSZER GESTRAMK25/1).

It was found as a result of check of 33 steam traps that only 17 steam traps are operating normally, and as many as 16 steam traps are malfunctioning. The results of investigation of steam traps are shown in Table 5.2.13.

Table 5.2.12 Insulation of bare steam pipe

Room	Steam Press bar(G)	Pipe		Heat Loss Present		Heat Loss after Insulation				Saved Energy MJ/h	Note
		Dia inch	Length m	kJ/(mh)	MJ/h	Material	thick mm	kJ/(mh)	MJ/h		
Vulcanizer	16.5	1	6.2	1675	10.4	Mineral Wool	25	212	1.3	9.1	
	5	1.5	14.0	1508	21.1	Mineral Wool	25	185	2.6	18.5	
Total					31.5				3.9	27.6	

Method for calculation : See Guideline.

Preconditions for calculation:
1. The saturation temperature of the steam at the subject pressure was adopted as the internal temperature.
2. The room temperature is 20°C.
3. The heat insulation thickness was determined with the economical thickness is Japan used as a reference.
4. The surfacre radiation rate is 0.9 at bare portions and is 0.5 at heat insulated portions.

Steam traps frequently repeat open/close actions. Their service life is uncertain, and they should be considered as consumables. In the case where no open portions are provided along the trap outlet lines and the lines are connected to a tank as observed at this factory, leakage from steam traps is hardly detected. Therefore, periodic inspection and repair should not be neglected. In order to effectively manage the steam traps, it is recommended that a trap register is prepared, and that recording of installation, repair and replacement is made and thus management of steam traps is carried by recording the trend of failures.

If the steam trap failure rate is of the level indicated above throughout the year, the loss indicated below can be prevented when these failed steam traps are suitably repaired or replaced.

62 kg/h × 7,920 h/y = 491 t/y
491 t/y × 750 Ft/t = 368,000 Ft/y

Table 5.2.13 Results of investigation of steam traps

Operation	Quantity	Description of failure	No	Level	Leaking rate kg/h
Good	17				
No good	11	Leakage	21	10	6.0
			24	12	7.0
			32	7	4.5
			33	9	5.5
			38	10	6.0
			41	9	5.5
			45	8	5.0
			46	6	4.0
			47	14	8.0
			52	7	4.5
			54	10	6.0
Subtotal					62.0
No good	5	Low Temp			
Grand total	33				

Note: "Low Temp" means that the temperature of the trap portion is low due to faulty discharge of the condensate or due to unsuitable temperature setup.

D) Repair of steam leakage

Steam leakage from curing presses was observed at 19 places as a result of rough inspection, as shown in Table 5.2.14.

Table 5.2.14 Steam leakage from curing presses

Leaking point	No. of points	Press No.
Frame	3	11, 15, 16
Relief valve	5	12, 26, 38, 43, 45
Seal	1	31
Safety valve	4	32, 34, 43, 57
Vicinity of mold adjust bolt	5	32, 34, 41, 56, 57
Safety valve mounting area	1	43
Total	19	

The rate of steam leakage from a hole of area A m^2 can be expressed by the following equation.

$$G = 71.64 \times 10^4 \times A \times \sqrt{P / v''} \qquad (1)$$

where;

G : Steam leaking rate kg/h

A : Area of leaking hole m^2

P : Absolute pressure of steam kg/cm^2abs

$$P = P_1 \times 1.0197 + 1.033$$

P_1 : Steam pressure in pipeline bar(G)

v" : Steam specific volume in pipeline m^3/kg

The leaking rate of steam of 5 bar(G) from a hole of diameter 1 mm is as follows.

$$\therefore G = 71.64 \times 10^4 \times 0.785 \times 0.001^2 \times \sqrt{\frac{5 \times 1.0197 + 1.033}{0.3155}}$$
$$= 2.5\ kg/h$$

The leaking rate varies at the points indicated above. But if it is assumed that a hole of 1 mm is located at each of these 19 points, the annual loss amount is as follows.

$$2.5 \text{ kg/h} \times 19 \times 7{,}920 \text{ h/y} = 376{,}200 \text{ kg/h}$$
$$376 \text{ t/y} \times 750 \text{ Ft/t} = 282{,}000 \text{ Ft/y}$$

It is required to make efforts to completely eliminate steam leakage when running is suspended on weekend or during mold exchange.

(2) Process and operation

The energy consumption in the curing process of this factory is considerably high compared to that at Japanese tyre manufacturers. That is, the energy consumption is two or three times as much as that in Japan. The causes for such a major difference should probably be judged based on strict analysis. But it is considered that the following factors are involved.

① Whether steam-inert gas curing method is adopted or not
② Differences in degree and scale of operation
③ Differences in steam supply method
④ Differences in equipment management scheme
⑤ Differences in atmospheric temperature and other factors

A) Inert gas curing process

Steam is most commonly used as the curing heat source in the tyre industry. But hot water and hot air are also used besides steam. Tyre curing method is often expressed by the heating pressure source used in the interior of the curing press bladder in general. Since just over one decade ago, steam-inert gas curing attracted attention and today this method is adopted by many tyre manufacturers. It is because this method provides such merits that the bladder life is improved and the investment cost is low compared to hot water equipment for high pressure curing besides reduction (to nearly a half) of used energy.

The price of the inert gas generation plant is about 100,000,000 yen for the scale of raw rubber consumption 100 t/d. L'AIR LIQUIDE of France is known as a representative supplier of the plant and nitrogen gases. When nitrogen gas curing method is adopted, however, it is necessary to take countermeasures such as use of stainless steel SUS32 against corrosion to pipelines and valves by fume of sulfuric acid or nitric acid generated to a minor extent. It is also necessary to pay attention to take measures for preventing fatal accidents caused by breathing of leaking gases.

B) Degree and scale of operation

Continuous operation scheme of three shifts of four teams, which is identical to other industries, has been sequentially adopted since 1970 at tyre factories in Japan as a transitional measure for shortening working hours. The number of factory operating days in a year is 346 days. (19 days are off per year in total including New Year's Day period, holiday season in May and holiday season in August.)

At this factory, however, the operation is suspended on Saturdays and Sundays, and the number of factory operating days in a year is 253 days. It was explained that cooling of pipelines is prevented by feeding low pressure steam during suspension of operation. At any rate, however, the difference in the energy efficiency between continuous production and intermittent production is estimated to be about 10%.

In addition, differences in the scale of production are involved. As for the scale of operation in the tyre industry, raw rubber consumption of 100 t/d is said to be the minimum pay line. If the scale is small, the burden of fixed expenses becomes large, and all indices including energy unit consumption become inferior. As the production of this factory is about 50 t/d, there is a handicap when the energy unit consumption is compared with other factories. However, it would be better to withhold making clear decision in the scope of investigation of this time because there is a possibility where compensation is made by production of articles other than tyres.

A Japanese material is shown in Table 5.2.15 for reference in order to learn the extent of the differences in the energy unit consumption caused by the scale of the business.

Table 5.2.15 Fuel unit consumption in rubber industry by scale

Fuel unit consumption with tyres and tubes

Scale (number of employees)	Classification / Year	Fuel consumption (kl/month)	New rubber consumption (t/month)	Unit consumption (kl/ton)	Ratio to previous year (%)	1984=100 (%)
1,000 and up (10)	1984	1,294.6	4,148	0.312	—	100.0
	1985	1,269.6	4,320	0.294	94.2	94.2
	1986	1,191.5	4,095	0.291	99.0	93.3
	1987	1,179.4	4,302	0.274	94.2	87.8
300~999 (15)	1984	538.2	1,933	0.278	—	100.0
	1985	516.2	1,965	0.263	94.6	94.6
	1986	487.6	1,892	0.258	98.1	92.8
	1987	489.2	2,012	0.243	94.2	87.4
100~299 (4)	1984	156.7	441	0.355	—	100.0
	1985	154.8	463	0.334	94.1	94.1
	1986	151.4	452	0.335	100.3	94.4
	1987	152.3	492	0.310	92.5	87.3
30~99 (3)	1984	25.0	65.2	0.383	—	100.0
	1985	25.1	62.4	0.402	105.0	105.0
	1986	25.9	62.6	0.414	103.0	108.1
	1987	26.7	71.1	0.376	90.8	98.2
Up to 29 (-)	1984	—	—	—	—	—
	1985	—	—	—	—	—
	1986	—	—	—	—	—
	1987	—	—	—	—	—
Total (32)	1984	678.8	2,263	0.300	—	100.0
	1985	660.4	2,335	0.283	94.3	94.3
	1986	622.3	2,229	0.279	98.6	93.0
	1987	619.4	2,355	0.263	94.3	87.7

Note: Figures in ( ) are number of places of business.

Fuel unit consumption with industrial goods

Scale (number of employees)	Classification / Year	Fuel consumption (kl/month)	New rubber consumption (t/month)	Unit consumption (kl/ton)	Ratio to previous year (%)	1984=100 (%)
1,000 and up (4)	1984	317.7	979	0.325	—	100.0
	1985	308.4	972	0.317	97.5	97.5
	1986	289.4	911	0.318	100.3	97.8
	1987	304.8	927	0.329	103.5	101.2
300~999 (21)	1984	156.2	279	0.560	—	100.0
	1985	157.9	291	0.543	97.0	97.0
	1986	155.6	281	0.554	102.0	98.9
	1987	158.8	287	0.553	99.8	98.8
100~299 (22)	1984	59.5	193	0.308	—	100.0
	1985	65.5	207	0.316	102.6	102.6
	1986	66.9	203	0.330	104.4	107.1
	1987	69.1	205	0.337	102.1	109.4
30~99 (6)	1984	25.3	28.4	0.891	—	100.0
	1985	25.7	29.8	0.862	96.7	96.7
	1986	23.3	30.3	0.769	89.2	86.3
	1987	19.9	20.2	0.985	128.1	110.6
Up to 29 (4)	1984	8.1	10.7	0.757	—	100.0
	1985	7.6	10.7	0.710	93.8	93.8
	1986	7.2	10.5	0.686	96.6	90.6
	1987	7.4	10.9	0.679	99.0	89.7
Total (57)	1984	106.0	250	0.424	—	100.0
	1985	108.3	259	0.418	98.6	98.6
	1986	106.4	249	0.427	102.2	100.7
	1987	109.2	253	0.432	101.2	101.9

C) Method of steam supply

Many of tyre factories in Japan have small size boilers installed at a place that is close to the curing process which consumes steam by a large rate or switched to electrical heating. It is based on the judgment that steam supply to distant facilities is not favorable from the standpoints of heat radiation loss and capital investment.

This factory, however, is dependent on the steam supplied from the boiler of Regional Heat Supply and Power Generation Public Corporation (distance about 2 km). The meter for measurement of the energy purchase rate is installed at the office building and the distance to the curing plant is not very far (200 m). But it is considered necessary to raise this point as a subject requiring examination in a long run, instead of easily approving the current situations.

D) Differences in atmospheric temperature

The steam consumption by season in 1990 is as shown in Table 5.2.16.

Table 5.2.16 Steam consumption by season

Date		4.6	7.31	10.18	12.19
Steam purchased	t/h	23.9	17.5	36.1	30.7
Consumed at curing process		12.2	8.9	11.8	7.7
Consumed at other processes + for heating		10.1	7.8	22.3	20.0
Sold to other companies		1.6	0.8	2.0	3.0
Condensate recovered		7.3	4.4	11.5	20.5
Assumed outdoor temperature	°C	8	21	10	0

Differences in the production level are not known. But increase/decrease of around 0.66 t/h per degree Centigrade of atmospheric temperature is observed except for October. While the minimum of the monthly average atmospheric temperature in Tokyo is 6.3°C, the monthly average is –2.8°C at the place where this factory is located. A difference of around 6 t/h may occur during the cold season. It is important to review the heating in general including duplicating doors for entrances, heat insulation of the building and electrical heating application for some places.

(3) Steam supply system

A) Prevention of heat radiation

Bare pipes without heat insulation were observed at many places along indoor pipelines between the steam header of the curing plant and curing presses. The effect of reduction of heat radiation obtained by heat insulation of these bare pipes and non-heat insulated valves located along their systems is shown in Table 5.2.15 and Table 5.2.16. The steam loss will decrease as follows by heat insulation when the steam vaporization latent heat of steam of 22, 16.5 and 7.2 bar(G) is assumed as 1,858, 1,916 and 2,043 kJ/kg respectively.

$$(77.2/1{,}858 + (40.7 + 42.3)/1{,}916 + (20.4 + 63.0)/2{,}043) \times 1000 = 126 \text{ kg/h}$$
$$126 \text{ kg/h} \times 7{,}920 = 998 \text{ t/y}$$
$$998 \text{ t/y} \times 750 \text{ Ft/t} = 748{,}500 \text{ Ft/y}$$

The expenses required for this heat insulation are estimated to be about 370,000 yen (equivalent to about 185,000 Ft) in Japan. Therefore, recovery of expenses can be made within an extremely short period of time.

Table 5.2.17 Insulation of Bare Steam Pipe

Room	Steam Press bar(G)	Pipe		Heat Loss Present		Heat Loss after Insulation				Saved Energy MJ/h	Note
		Dia inch	Length m	kJ/(mh)	MJ/h	Material	Thick mm	kJ/(mh)	MJ/h		
Curing Press	16.5	6	4	6,963	27.9	Mineral Wool	50	419	1.7	26.2	
Steam Line	16.5	2.5	4.5	3,473	15.6	Mineral Wool	50	239	1.1	14.6	
	7.2	6	4.3	5,055	21.7	Mineral Wool	50	318	1.4	20.4	
Total	16.5				43.5				2.8	40.7	
	7.2				21.7				1.4	20.4	
					65.2				4.1	61.1	

Method for calculation : See Guideline.

Preconditions for calculation:

1. The saturation temperature of the steam at the subject pressure was adopted as the internal temperature.
2. The room temperature is 20°C.
3. The heat insulation thickness was determined with the economical thickness in Japan used as a reference.
4. The surface emissivity is 0.9 at bare portions and is 0.5 at heat insulated portions.

Table 5.2.18 Insulation of Steam Valve

Steam Line	Steam Press bar(G)	Valve			Heat Loss Present		Heat Loss after Insulation				Saved Energy MJ/h	Note
		Dia inch	Number	Equi. L m	kJ/(mh)	MJ/h	Material	Thick mm	kJ/(mh)	MJ/h		
17.5 bar Line	22	6	1	1.78	7,819	13.9	Mineral Wool	50	462	0.8	13.1	Globe Valve
Header	22	4	1	1.58	5,601	8.8	Mineral Wool	50	350	0.6	8.3	Globe Valve
	22	3	1	1.56	4,476	7.0	Mineral Wool	50	294	0.5	6.5	Globe Valve
	22	1	5	1.21	1,888	11.4	Mineral Wool	25	235	1.4	10.0	Globe Valve
17.5/7 bar	22	6	3	1.78	7,819	41.8	Mineral Wool	50	462	2.5	39.3	Globe Valve
Reducer	16.5	6	1	1.78	6,963	12.4	Mineral Wool	50	419	0.7	11.6	Control Valve
7 bar Line	7.2	6	4	1.5	5,055	30.3	Mineral Wool	50	318	1.9	28.4	Globe Valve
Header	7.2	6	2	1.76	5,055	17.8	Mineral Wool	50	318	1.1	16.7	Reducing Valve
17.5 bar Line	16.5	2.5	3	1.5	3,473	15.6	Mineral Wool	50	239	1.1	14.6	Globe Valve
Basement	16.5	1	9	1.21	1,689	18.4	Mineral Wool	25	213	2.3	16.1	Globe Valve
7 bar Line	7.2	2	3	1.11	2,063	6.9	Mineral Wool	25	239	0.8	6.1	Globe Valve
Basement	7.2	1	9	1.22	1,239	13.6	Mineral Wool	25	162	1.8	11.8	Globe Valve
Total	22					82.9				5.7	77.2	
	16.5					46.4				4.1	42.3	
	7.2					68.6				5.6	63.0	
						197.9				15.5	182.5	

Method for calculation : See Guideline.

Preconditions for calculation:
1. The saturation temperature of the steam at the subject pressure was adopted as the internal temperature.
2. The room temperature is 20°C.
3. The heat insulation thickness was determined with the economical thickness in Japan used as a reference.
4. The surface emissivity rate is 0.9 at bare portions and is 0.5 at heat insulated portions.

B) Recovery and reutilization of condensate

The condensate that is discharged out of curing presses is not used at present due to the reason that it is contaminated. However, the quantity of heat possessed by the condensate is large, and it is as much as 31% in the case of steam of 16.5 bar(G) and 24% in the case of steam of 5 bar(G). In addition, when the condensate is discharged to the open space at atmospheric pressure, 20% vaporizes and takes away the heat of 61% in the case of the condensate of 16.5 bar(G) as shown by the following equation. Also in the case of 5 bar(G), 11% vaporizes holding heat of 44%.

(Typical calculation)

Enthalpy of condensate of 16.5 bar(G) = 878 kJ/kg

Enthalpy of saturated steam of atmospheric pressure = 2,676 kJ/kg

Enthalpy of saturated water of atmospheric pressure = 419 kJ/kg

Vaporization V (kg) when condensate of 16.5 bar(G) of 1 kg is reduced to atmospheric pressure

$$878 = 2{,}676 \times V + 419 \times (1 - V)$$
$$V = 0.20 \text{ kg}$$

Ratio of quantity of heat = $2{,}676 \times 0.20 / 878 = 0.61$

It was observed at this factory that steam of a large quantity is discharged out of vents outside the building from time to time. On the other hand, however, it is considered that rubber kneading process, extrusion process, etc. have demand for heat that can be satisfied by low temperature heat sources. It is therefore recommended that the condensate is positively used.

As the method for use of the heat of the condensate, such a method that fresh water is heated using a heat exchanger can be considered. But what is often adopted is the method to use as flash steam. It is recommended that a pressure-tight flash tank is installed in front of the current underground tank, the condensate is led into this tank, its pressure is reduced to around 2 bar(G) and the generated low pressure steam is used for demand for low temperature heat.

The flow sheet is shown in Figure 5.2.8.

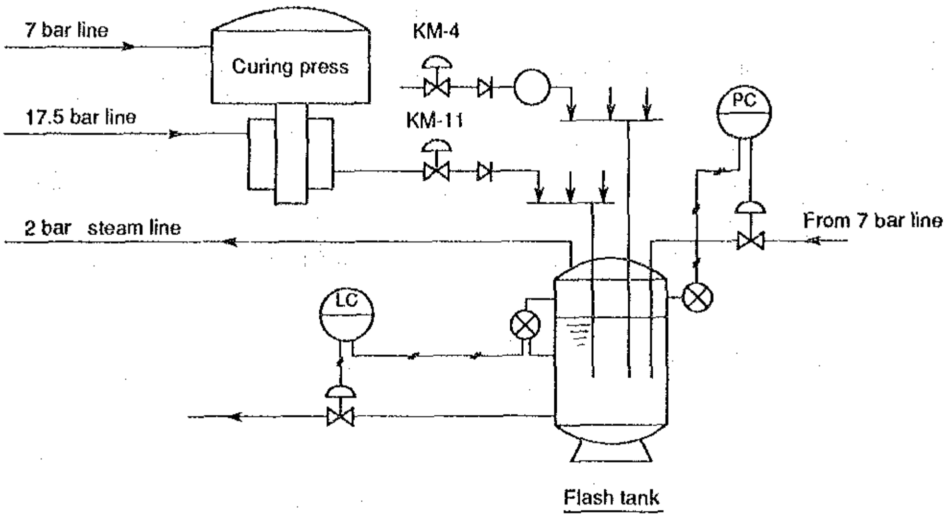


Figure 5.2.8 Flash tank

The volume and heat quantity at the time when flash steam is recovered at 2 bar(G) can be calculated as follows.

Table 5.2.19 Flash steam

	Enthalpty kJ/kg		Flash Steam	
	Saturated steam	Saturated water	kg/kg-Steam	Heat %
16.5 bar(G)	2,794	878	0.15	14.3
5 bar(G)	2,756	670	0.05	4.9
2 bar(G)	2,725	561	—	—

It is requested that examination of recovery of flash steam is made in correspondence to the demand for low temperature heat in the neighboring processes. When the flash tank is installed, the back pressure increases. It is therefore necessary to review the trap capacity and setup of control valves.

(4) Electrical equipment

A) Outline of electrical equipment

The electric power is received at 21kV and is supplied to loads with stepdown made to 400V at the substation for each process, as shown in the one line diagram in Figure 5.2.3.

The contracted power is 7,000kW, and the unit price is as shown in Table 5.2.20.

Table 5.2.20 Electric power unit price

Time		Energy Charge Ft/kWh
Day Time	6:00 - 16:30	3.20
Peak Time	16:30 - 21:00	4.70
Night Time	21:00 - 6:00	2.35

Control with process computers is introduced. As there is a fear of influence of higher harmonics, sufficient countermeasures have been taken against them.

Various technical materials and drawings are consolidated and the level of engineers is high.

The compressors and illumination related to the curing process were investigated.

B) Countermeasures for improvement and their effect

a. Improvement of the operation of compressors

Three compressors of 300kW and two compressors of 150kW are used at this factory, and compressed air to the curing process is supplied from a compressor of 150kW. The delivery pressure of this compressor is 9.2 bar(G) and the pressure on the curing press side is 8.4 bar(G), but it was stated that the required pressure is 6 - 7 bar(G).

The piping system is as shown in Figure 5.2.9.

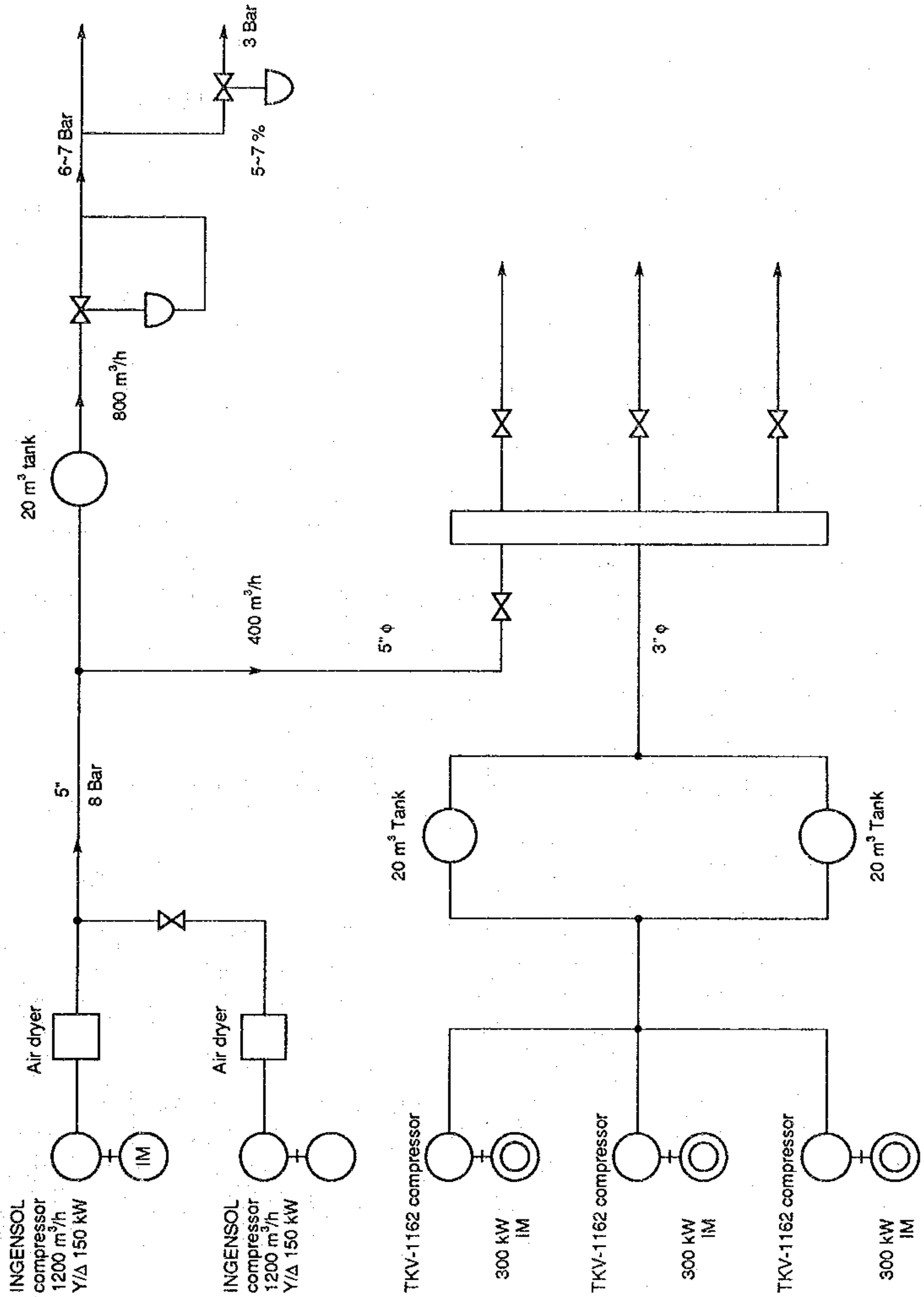


Figure 5.2.9 Compressed air piping system

The results of measurement of electric power of the 150kW compressor are shown in Table 5.2.21.

Table 5.2.21 Results of measurement of electric power

	Active Power kW	Apparent Power kVA	Reactive Power kVA	Power Factor
Average	127.9	142.8	63.6	0.896
Max	131.9	147.0	64.8	0.898

Measuring Time 8/11 10:32 - 8/23 9:13

The current air delivery pressure has a margin of about 1 bar to the required pressure. Therefore, it is recommended that the delivery pressure of the compressor is reduced in steps by a minor extent at a time while observing the situations of the air consuming equipment. The electric power can be saved by about 4% when the delivery pressure is reduced by 1 bar.

Annual electric power saving amount

127.9 kW × 0.04 × 7,920 h/y = 40,500 kWh/y
40,500 kWh/y × 3,69 Ft/kWh = 149,000 Ft/y

b. Improvement of lighting

Lighting equipment mounting three 40W fluorescent lamps are mounted at 408 places, i.e., 34 in the longitudinal direction × 12 rows, in the curing process building (72 m × 54 m). Ventilation holes are located at three places in the roof, but they are of a structure that is not of good use for introduction of daylight.

The results of measurement of illuminance taken on the working floor in the vicinity of curing presses in the morning of August 23 are shown in Figure 5.2.10. The mean value of 52 measuring points was 170 Lx, which is rather low for a general workshop.

Layout and illuminance of existing fluorescent lamps (average 170 Lx .. measured value)
Curing factory 72 m x 54 m

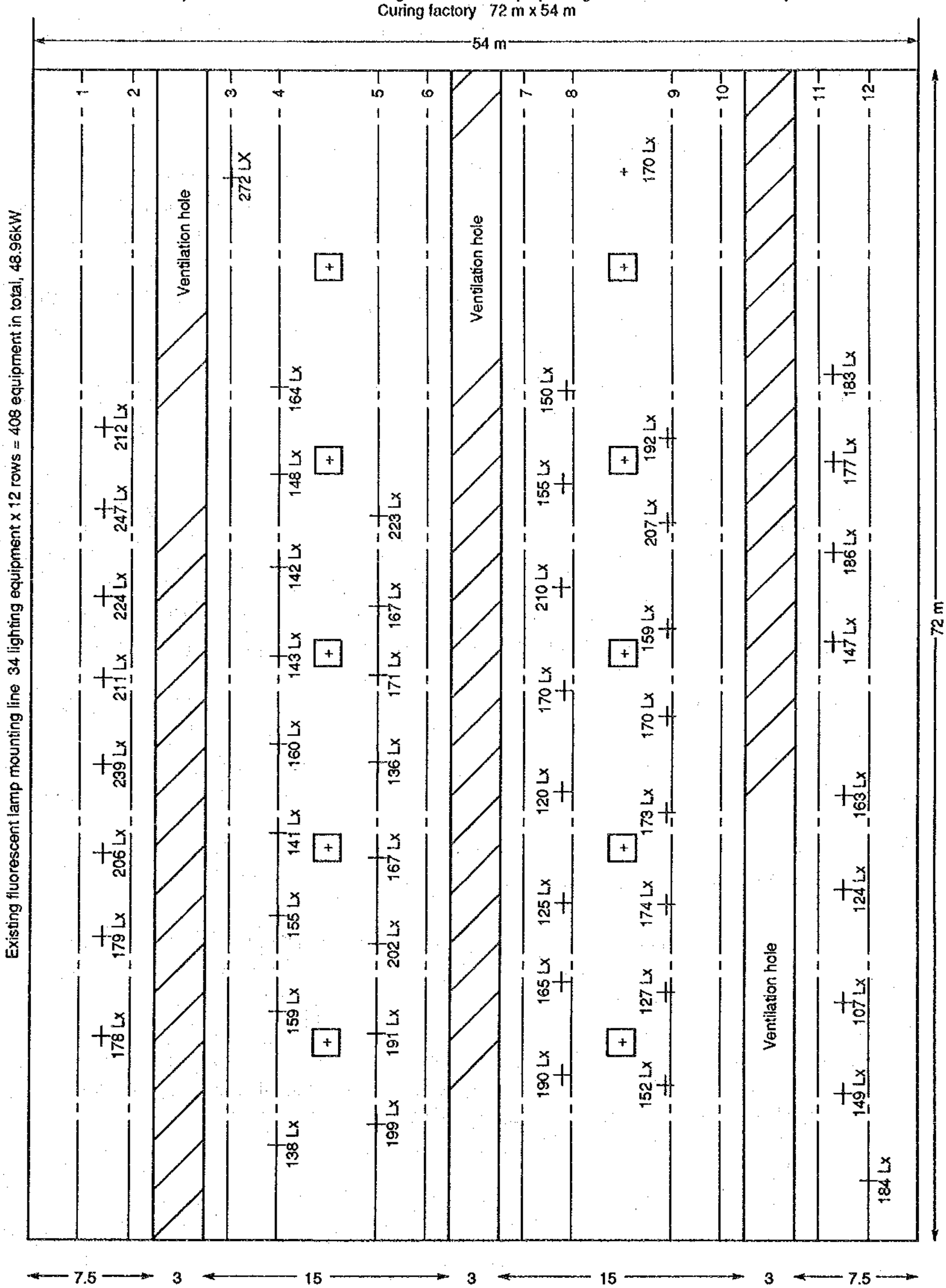


Figure 5.2.10 Distribution of illuminance in curing factory

It cannot be said that lighting equipment mounting positions are suitable for the locations of curing presses, and the lamp type is not suitable for high ceiling of height close to 7 m.

Fluorescent lamps are used in general in the case where the mounting height is low. High pressure sodium lamps and fluorescent mercury vapor lamps are used when the mounting height is high. As for the lamp efficiency, while it is around 75 lm/W with fluorescent lamps, it is 120 lm/W, that is 1.6 times that of fluorescent lamps, with high pressure sodium lamps.

The illumination design in the case where the fluorescent lamps are replaced with high pressure sodium lamps is indicated below.

The number of required lighting equipment is given by the following equation.

$$N = \frac{E \times A}{F \times U \times M} \qquad (2-1)$$

where;

N : Number of equipment

E : Mean illuminance (Lx) on working face : 170 Lx
A horizontal plane of height 0.85 m above the floor is taken as the working face

A : Floor space of the room (m^2) : $72 \times 54 = 3{,}888$

F : Total luminous flux of lamps per lighting equipment (lm) : 26,500 lm

U : Utilization factor

M : Maintenance factor : 0.66

Maintenance factor M is the figure that forecasts the ratio of drop from the initial illuminance as the working time elapses. It varies by to what extent maintenance of equipment is done.

Utilization factor is the ratio of the luminous flux that enters the working face to the total luminous flux emitted from the lamps. It is obtained from Table 5.2.22 by room index, ceiling reflection rate, etc.