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PLANNING OF THE 350 MW AND 600 MW CLASS THERMAL POWER STATION FOR MIDDLE LOADS

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I. INTRODUCTION

There has been a demand to develop middle load thermal power stations with excellent load-following capabilities and start-up capabilities for load control of the power system in keeping with the construction of large capacity thermal power stations and the practicality of large capacity nuclear power stations. In the near future, the base load will be handled by large capacity thermal and nuclear power stations of the 1,000 MW class or over. Units of the 350 MW and 600 MW class will be needed to handle peak loads due to rapid load changes and operation with week end and night shut-downs will probably be required. In order to meet these demands, the units must possess excellent start-up and load-following capabilities and the thermal efficiency must be higher than in past units. Middle load thermal power stations will therefore probably need start-up capabilities of 60 to 90 minutes from ignition to full load and load-following capabilities of about 5%/min. This article gives an outline of the planning of 350 MW and 600 MW middle load thermal power stations which have these capabilities.

II. BASIC UNIT SPECIFICATIONS

1. Basic Condition Assumption

When planning, the following conditions were assumed :

Utilization factor :	0.45/year (equalized full load time : 3,942 hours)
Fixed cost factor :	0.12 (0.15 also considered as reference)
Fuel cost :	0.6 yen/Mcal
Fuel :	Heavy oil
Coooling water :	Seawater at 21.1°C (Maximum 30°C)
Atmospheric temperature :	20°C
Start-up capability :	60 to 90 minutes from ignition to full load
Load following capability :	5%/minute or over
Minimum load :	20% or less

2. Main Steam and Reheat Steam Conditions

In investigations of the main and reheat steam conditions, basic values of 251 ata for the main steam pressure and 538°C for both the main and reheat steam temperatures were employed. Economy

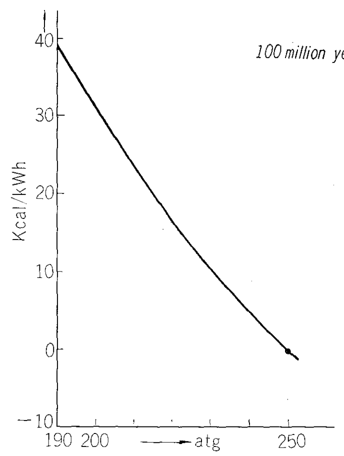


Fig. 1 Main steam pressure/heat rate deviation

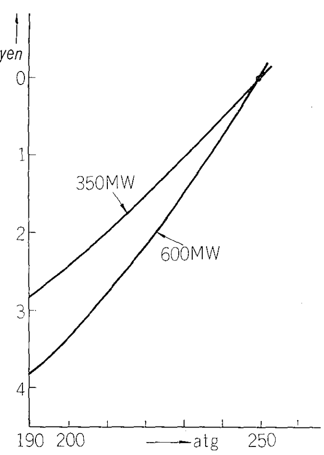


Fig. 2 Main steam pressure /initial cost deviation

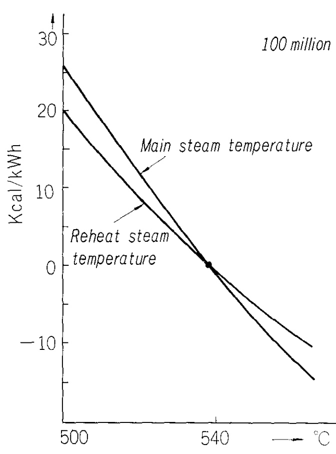


Fig. 3 Steam temperatuer/heat rate deviation

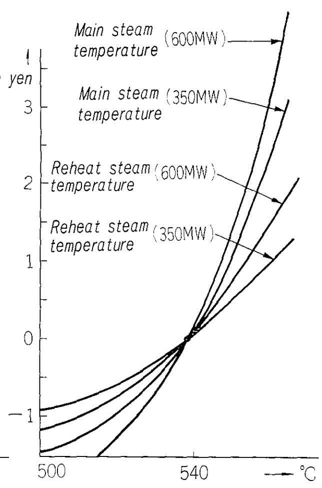


Fig. 4 Steam temperature/ initial cost deviation

comparisons were made for various steam conditions. *Figs. 1* to *4* show the relations between deviations in the steam conditions and the initial costs and heat rate for both 350 and 600 MW units according to our case studies. *Figs. 5* to *7* show deviations in the yearly costs in accordance with steam condition changes considering the utilization factor, fuel costs and fixed cost factor based on these calculated values. The best steam conditions as obtained from the results of these calculations are shown in *Table 1.* As can be seen from this table, the best steam condition values deviate somewhat from the assumed basic values due to differences in the unit capacities and

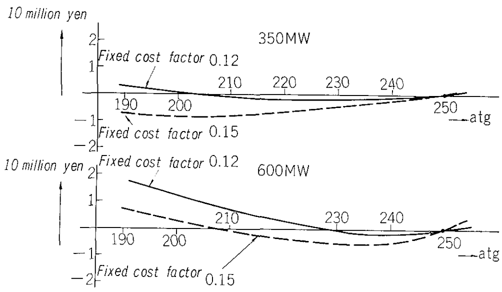


Fig. 5 Main steam pressure/year's cost deviation

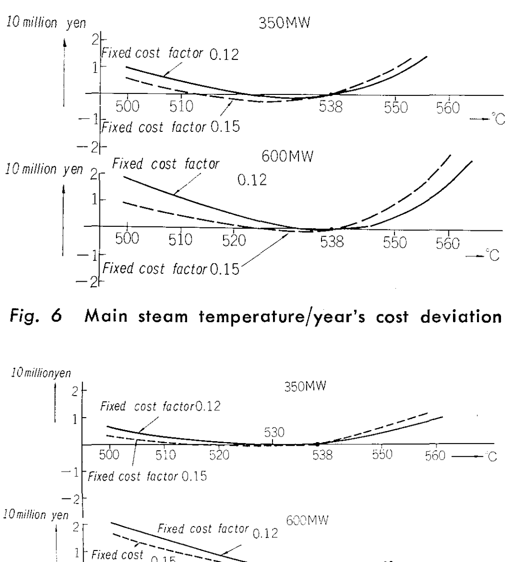


Fig. 6 Main steam temperature/year's cost deviation

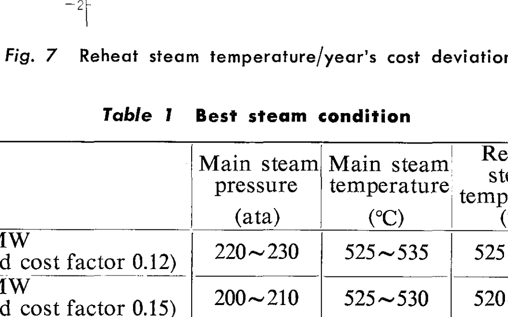


Fig. 7 Reheat steam temperature/year's cost deviation

***Table 1* Best steam condition**

	Main steam pressure (ata)	Main steam temperature (°C)	Reheat steam temperature (°C)
350 MW (Fixed cost factor 0.12)	220~230	525~535	525~540
350 MW (Fixed cost factor 0.15)	200~210	525~530	520~530
600 MW (Fixed cost factor 0.12)	230~250	Promise 540	540~550
600 MW (Fixed cost factor 0.15)	225~235	Promise 530	550~560

fixed cost factor. These deviations, however, are small and it was therefore decided that the assumed basic conditions of 251 ata and 538/538°C are appropriate for both the 350 and 600 MW units.

3. Operation System

There are two main types of operating systems used in thermal power stations: the constant pressure operation system which is a nozzle cutout governing system and the sliding pressure operation in which the main steam pressure is changed for each load and the turbine control valves are usually fully open. However, when rapid changes of very small loads such as the AFC are required, this latter system is used in combinations with a throttle governing system.

The system combining a sliding pressure operated Benson boiler and a throttle governed turbine has the following advantages.

1) Turbine efficiency is high since no control nozzle is required.

2) Because there is almost no change in the steam turbine temperature when the load is varied, limitations in respect to load changes are few.

3) With partial loads, the steam pressure is lowered, so that the stress is also lowered and the life is lengthened.

4) Since the volume flow is constant, the turbine efficiency is high in the partial loads.

5) The feed water pump pressure is low. This is especially true when it is the turbine driven pump.

6) Compared with constant pressure operation, the temperature at the high pressure turbine outlet is high during partial loads and therefore the reheat temperature can be kept high.

7) Since the turbine temperature does not change much even at low loads, start-up time during restarting after night shut-downs etc. is very short since the turbine temperature is kept high. The relations between load and steam temperature in the high pres-

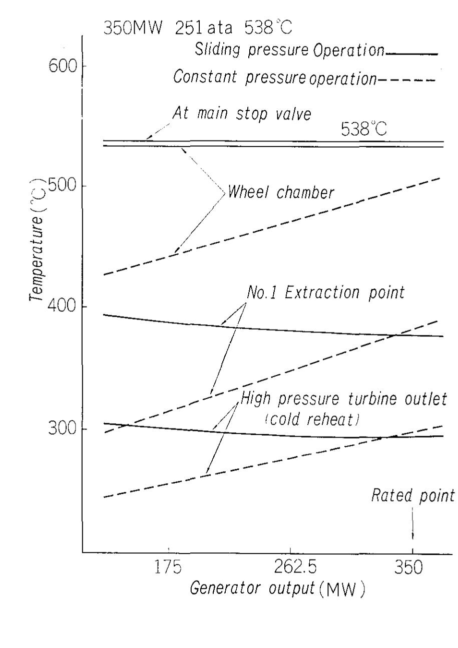


Fig. 8 **Load and steam temperature in high pressure part of turbine**

sure part of the turbine are compared in *Fig. 8* for constant pressure and sliding pressure operation. All of the above features are very effective in the operation of middle load thermal power stations.

III. BOILER

1. Boiler Start-up Systems

There are three types of Benson boiler start-up systems as can be seen in *Fig. 9*: the system with a flush tank, the system with a flush tank and a steam separator and the system with a steam separator and a circulating pump.

1) Start-up system with flush tank

This is the system which was most commonly used in the past. Previous to ignition the boiler is filled with water up to the superheater. After ignition, this water flows out into the flush tank. During the course of start-up, the feed water is kept at a constant value 30 to 35% that at full load so as to achieve average distribution of water in the evaporator pipes. The water in the flush tank is conducted to the feed water tank while the steam cools the reheater and is led to the condenser via the spill-over values. Since water is added up to the superheater at the time of start-up, the fuel can be added without considering superheating of the superheater during cold starts. This makes possible the rapid cold starts characteristics of Benson boilers. However, during hot starts, the residual pressure in the superheater is high and the header outlet is thick. Therefore it is first necessary to cool the boiler down to the steam saturation temperoture of the residual pressure and then srart. Thus, this method is not suitable when frequent, rapid hot starts are required. In addition it is impossible to go below a load which corresponds to 30 to 35% of the minimum flow in the evaporator for minimum boiler loads.

2) Start-up with flush tank and steam separator

In order to provide rapid hot starts without the defects mentioned for the previous method, a steam separator is installed after the evaporator. Previous to start-up, minimum flow of water is passed for 5 to 10 minutes through the evaporator. This water is separated by the steam separator and the blowdown tank contents are conducted to the flush tank. Therefore, no water flows in the superheater, the start valve is opened at ignition and the water level in the steam separator is controlled. With the combustion load is increased, the water level in the steam separator is lowered. The water level is not controlled and operation returns to the original Benson boiler operation. For cold starts, the flush tank is utilized as mentioned previously and the superheater is cooled. Compared with the previous method, this method has the advantages of allowing rapid hot starts and keeping start-up loss to a minimum, but it can not be said to be the most appropriate because of the heat loss due to the turbine by-pass steam when the water from the steam separator is conducted to the low pressure circuit.

3) Start-up with steam separator and circulating pump

This method is used when frequent starting and stopping is required or operation must be at minimum loads of 35% or under. During start-up or low load operation, the circulating pump is used and about 35% of the minimum flow of the evaporator is maintained. The water leaves the evaporator, passes through the steam separator and is returned to the boiler inlet by the circulating pump. This returned water contains some heat which can be transferred to the boiler feed water. Since the superheater is not filled with water before start-up, there is no water explosion after ignition and a flush tank is not necessary.

Of the above three methods, the last one using a steam separator and a circulating pump is the most appropriate for middle load thermal power stations. However, since it is not possible to fill the superheater with water during cold starts with this method, the excellent cold start characteristics of the original Benson boiler are lost. When the operating condi-

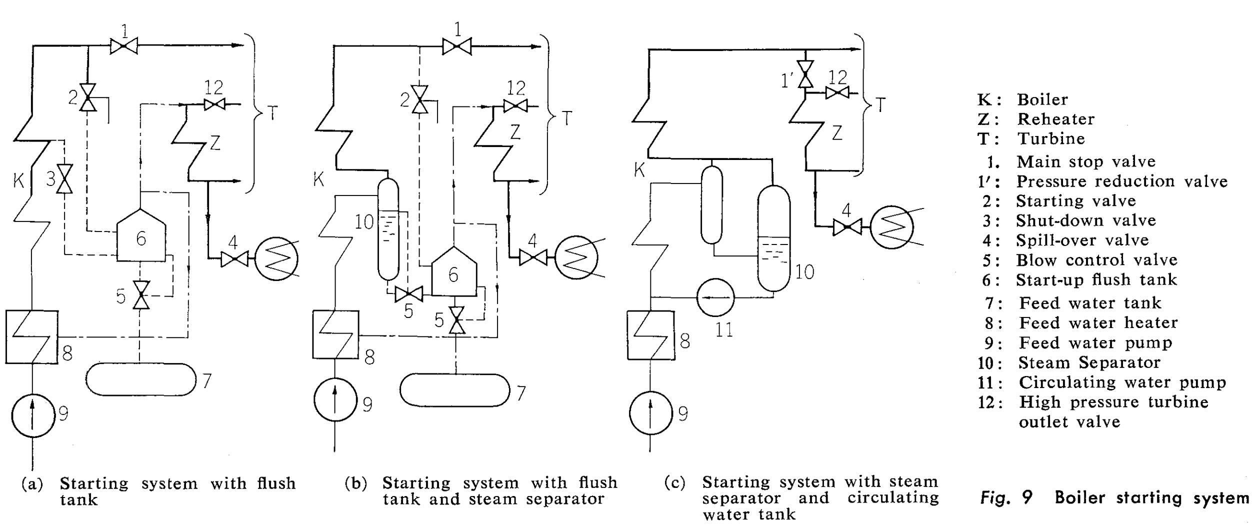


Fig. 9 Boiler starting system

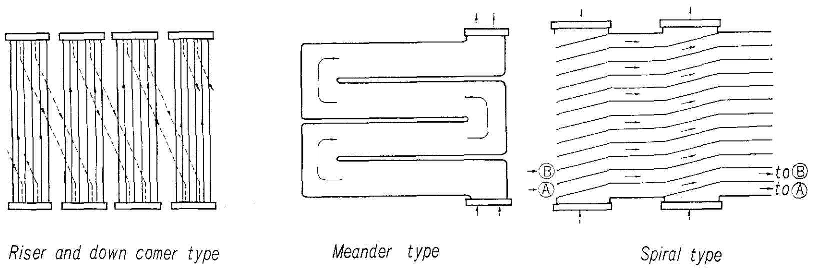


Fig. 10 Types of evaporation sections

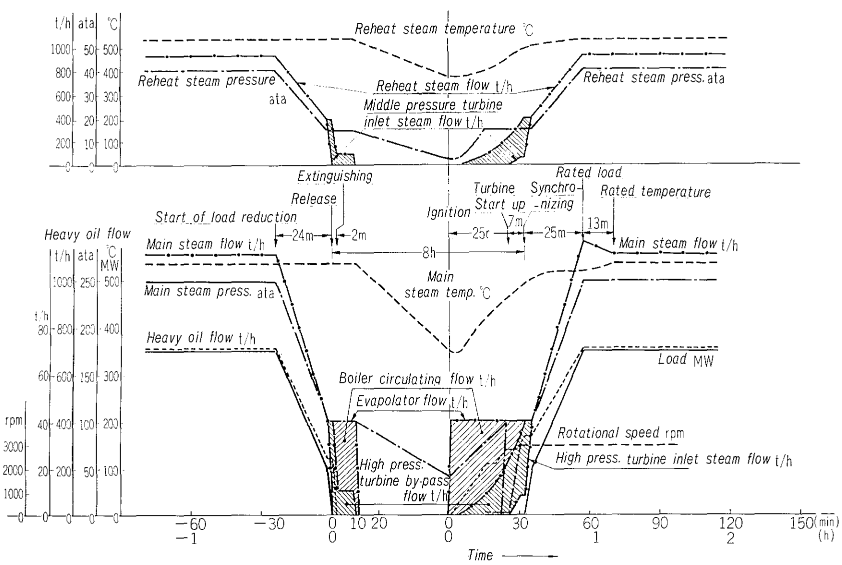


Fig. 11 350 MW starting-up diagram after 8 hrs standstill

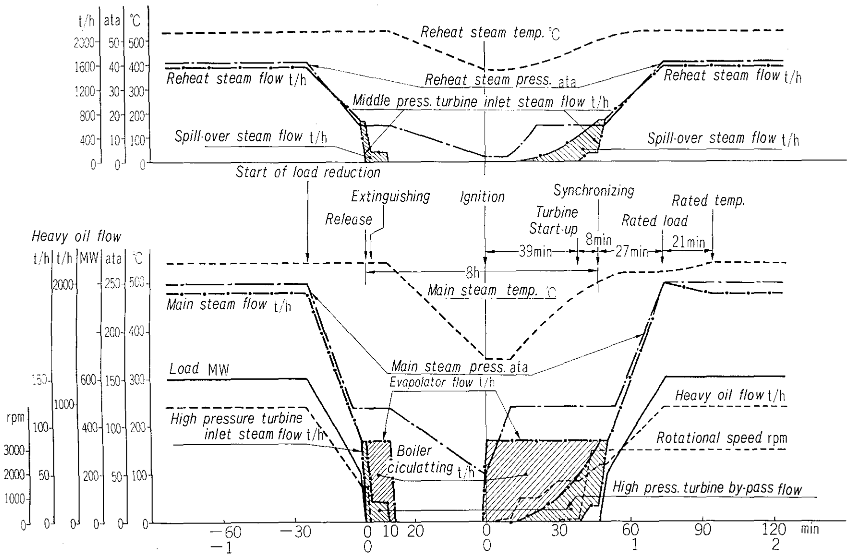


Fig. 12 600 MW starting-up diagram after 8 hrs standstill

tions require frequent cold starts, it is possible to use this method in combination with the start-up method using a flush tank.

2. Features of Boiler Construction

Benson boilers have changed considerably in the construction of their furnace generating tubes. Three types of evaporation sections are shown in *Fig. 10*. Initially, the riser and downcomer system was used in Benson boilers but later theoretical research progressed considerably and in about 1960, the meander system was conceived. From 1964, the spiral system came into use. With these changes, costs were cut and design became more rational in various ways including the realization of sliding pressure operation in boilers at supercritical pressures and also minimum feed water pump power due to the use of appropriate flow rates in the tubes.

In the spiral type evaporator, the feed water is raised upward in succession from the bottom of the boiler around the combustion chamber with heat applied a little at a time. Parallel tubes run around all four walls of the combustion chamber so that uniform heat is applied and the temperature differences between the tubes are very small. When risers and downcomers are used or when the combustion chamber is constructed with four walls of perpendicular tubes arranged so that there is one pass rising, the flow rate inside the tubes is related to the number of parallel tubes and there is a mutual relationship between the flow rate and the size of the combustion chamber (horizontal cross-sectional area). There are thus limits in the design of the chamber. However, in the spiral type, there is almost no relation between the size of the combustion chamber and the flow rate inside the tubes. Therefore an appropriate number of parallel tubes can be selected and the flow rate can be determined arbitrarily. In this way, the boiler can be planned with sufficient safety. In the spiral type, there is also no header from the combustion chamber inlet to the outlet and the liquid in the tubes generally flows in a gentle slope upwards so that there is no need to worry about any unequal distribution along the way. Therefore boilers designed for super critical pressures can also be operated at sub-critical pressures without any problems. This is one of the major features used in the planning of boilers for supercritical pressures and sliding pressure operation.

3. Boiler Start-up Capability

The limit imposed on the shortest start-up time is decided by the permissible temperature change rate of the pressurized parts. This permissible temperature change rate is limited by the thermal stress which occurs in the thick sections of the pressurized parts. The position when limits apply is considered to be around the hole of the mixing header at the final superheater outlet. In order to make up a starting program, it is necessary to determine the shortest start-up time within the permissible stress range. In practice, what is thought to be the most suitable star-up schedule must be obtained from this. These results are compared with those obtained from actual operation, mutual corrections are made and the most suitable sequence is established. *Figs. 11* and *12* show the start-np diagrams assumed for the 350 MW and 600 MW units respectively. The results of a stress analysis conducted around the hole of the mixing header at the final superheater outlet of a 600 MW boiler started on the basis of the start-up diagram in *Fig. 12* are as follows.

The analysing conditions were approximately as follows.

The analysing conditions were approximately as shown in *Fig. 13*. The temperature rise rate was a constant 4°C/minute, and the temperature was raised from 340 to 520°C. After being held at 520°C for 15 minutes, it was again increased to 545°C. Calculations of the surface temperature distribution were made at intervals of 1 to 2 minutes to obtain a two-dimensional transient temperature distribution. As shown in *Fig. 13,* the results revealed that the inner

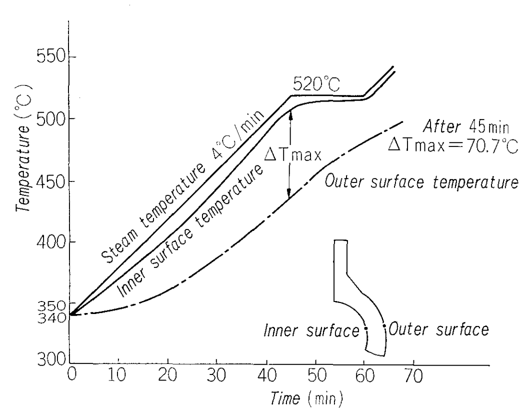


Fig. 13 Temperature-up at inner and outer surface of header

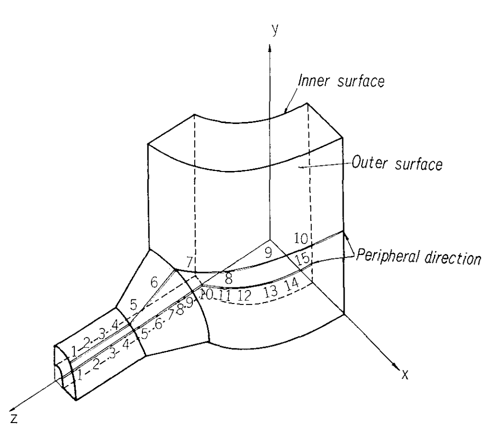


Fig. 14 Shape of calculated part

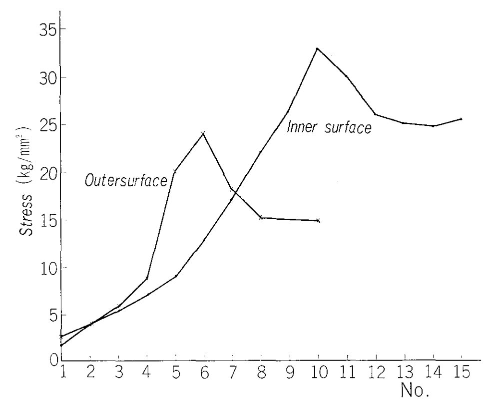


Fig. 15 Distribution of stress

surface temperature was a maximum at the time when the steam temperature was 520°C. This maximum value was $\Delta T_{max}=70.7$°C. However, the analysis of thermal tension was performed in time steps at which the tension reached a maximum by means of a three-dimensional arbitrary stress calculation program and a two-dimensional revolving symmetrical stress calculation program. *Fig. 14* shows the shape of the part analyzed and *Fig. 15* shows the stress distribution. As can be seen from these figures, the stress becomes a maximum around the periphery of the hole due to the influence exerted by the hole nipple. This stress was evaluated by ASME and SEC III and the life was determined from the repeated stress and the S/N curves. The permissible number of repetitions was found to be N=15,000.

In accordance with the same concepts, a thermal stress analysis was also conducted at the feed water port nozzle stub of the high pressure feed water heater which has the most severe thermal stress from the viewpoint of both operating conditions and the construction of tubing and auxiliary devices. The results showed that the permissible number of repetitions were greater than at the outlet header of the final superheater as described previously.

4. Load Following Capability

In previous units operated under constant pressure, the load change rate was generally limited by temperature changes in the high pressure part of the turbine. In this planning, there were no limitations on the turbine side due to the use of the sliding pressure operation as was described previously, and the unit load change rate was determined by the load change rate on the boiler side. In Germany, many large capacity sliding pressure type boilers are already in operation and in boilers using heavy oil, there is no problem in operation if the load change rate is 6 to 7%/minute or over. For small, fast load changes such as in frequency control, there is a delay in boiler combustion. As was described previously, the turbine control valve is kept at some constant degree of opening and for small, fast load changes, the so-called throttle sliding pressure operation

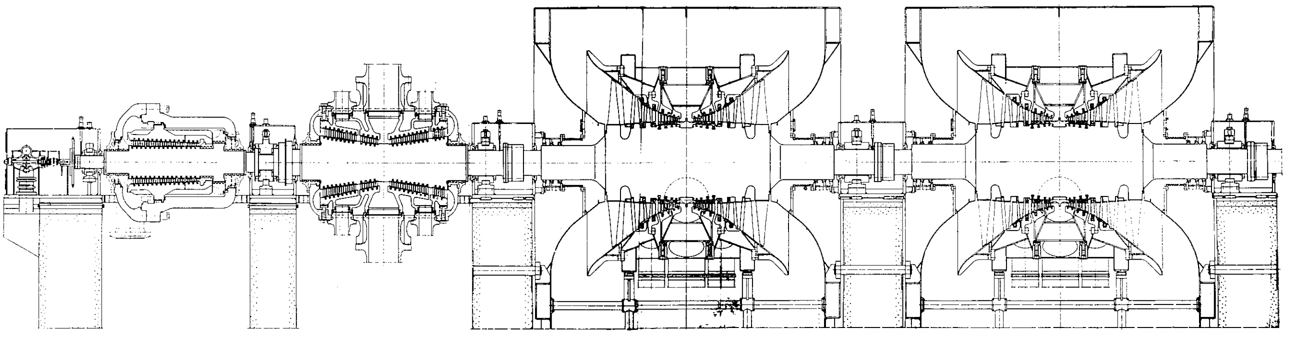


Fig. 16 Cross-sectional view of turbine

utilizing the heat accumulated in the boiler by switching the turbine control valve is performed.

IV. TURBINE

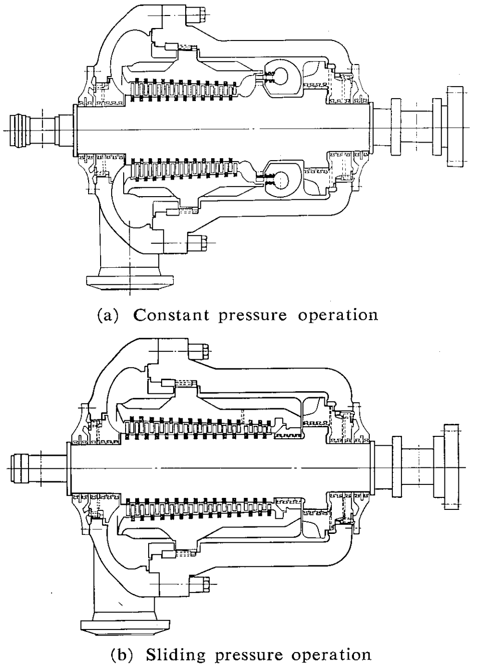
1. Construction of Super Critical Pressure Turbine with Sliding Pressure Operation

Fig. 16 shows a cross section of this turbine. Except the dimensional differences, the 350 MW and 600 MW units are of completely the same construction. The main unit is a tandem four cylinder construction consisting of a high pressure section, middle pressure section and two double-flow low pressure sections. The shaft bearing is of the so-called 3-bearing construction with one bearing between each cylinder. The high pressure section employs the compact pot-type construction. There is no nozzle chamber required for the throttle type sliding pressure operation and a full admission is used. The middle pressure section is of the counter flow type construction in which the steam flows in opposite directions in a double casing. The shaft thrust force is independently balanced. The two low pressure sections have outer casing of welded construction. This is of tripple casings altogether. The final low pressure blade is 680 mm with a 4 flow exhaust in the 350 MW unit and 875 mm also with a 4 flow exhaust in the 600 MW unit. The steam input to the low pressure sections flows into the innermost part of the inner casing and the many considerations have been paid to heat deformations. The main steam and the reheat steam enter into the turbine through four compound valves consisting of main stop valve and control valve which are arranged near each other in the turbine casing. The connection between these compound valves and the turbine casing is made by a special nut. The steam quantity in this connection part is small and it is safe in respect to thermal stress.

2. Characteristics of Turbine

Of the characteristins of the turbine, the most advantageous in the case of middle load thermal power stations are as follows.

1) The advantages of the sliding pressure operation are used to the utmost. With throttle control, there is no nozzle and a reaction stage with full admission is used which makes the construction simple. Since there are no temperature changes due to load changes within the turbine because of the use of sliding pressure operation, there are no limitations when changing the load or starting up caused by thermal stress, and rapid load changes and start-up are both possible. *Fig. 17* shows cross-sectional views of the high pressure sections for sliding pressure operation and for constant pressure operation with nozzle chamber. It is evident from these figures that the construction of the inlet part has been simplified considerably.



(a) Constant pressure operation

(b) Sliding pressure operation

Fig. 17 Cross-sectional view of HP part

2) The pot-type casing used in the high pressure section is for almost complete rotational symmetry, so that there is a high adaptability in respect to heat. The outer casing has no horizontal flange so that the flange is not exposed to high pressure high temperature steam in the steam inlet part. The high pressure flange bolts which often cause problems during start-up and load cut off are also not required. There is a cylindrical flange in the exhaust part of the high pressure casing but this only comes in con-

tact with the lowest temperature and pressure steam in the high pressure section.

3) There are 4 casings: one for high pressure, one for middle pressure and two for low pressure. The span of each of these casings is short and the rotor diameter is small. There are thus no useless thick parts and the casing walls are all thin.

4) There are 4 compound valves for both high and middle pressure and they are as close as possible to the casing so that the safety against shaft overspeed at the time of load shut down is increased.

5) There is a by-pass separating the low pressure and high pressure systems which aids in the rapid start-up of the boiler.

6) The blades are of the reaction stage type and there is a large axial gap between them. So that there are few limitations in respect to operation. The shroud is cut out of a blade so that there are no troubles caused by shroud rivets. The low pressure blade is also free and neither a shroud nor racing wire are used. There is therefore no danger of cracks.

7) In order that a wide rodge of automatic operation is performed from turning to full load, an electro-hydraulic governor, an electric rotation transmitter, a wall temperature device and a hydraulic turning device are all employed.

3. Turbine Start-up

The turbine is governed by throttle control and is combined with a Benson boiler which undergoes sliding pressure operation. *Figs. 18* and *19* show selection curves of turbine inlet steam temperature during start-up for the 350 MW and 600 MW units respectively. These figures show values which correspond to the temperatures of the metal at the high pressure casing inlet during shut down. The permissible inlet steam temperature and its rise time during turbine start-up and initial loading are selected on the basis of the results of a detailed analysis of the temperature difference between the inner surface and the center of the casing during start-up and loading. Since the appropriate limit for the steam inlet temperature is determined with a large margin,

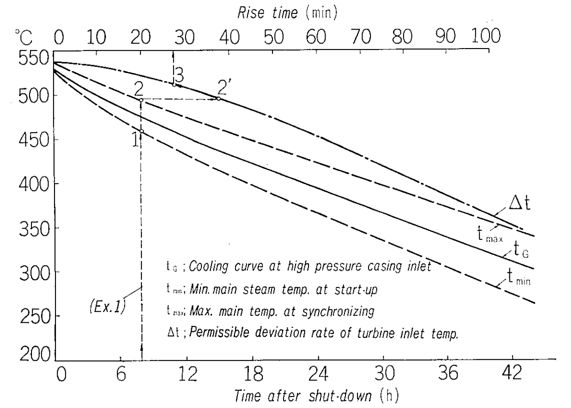


Fig. 18 **Main steam temperature and its rise time at start-up (350 MW)**

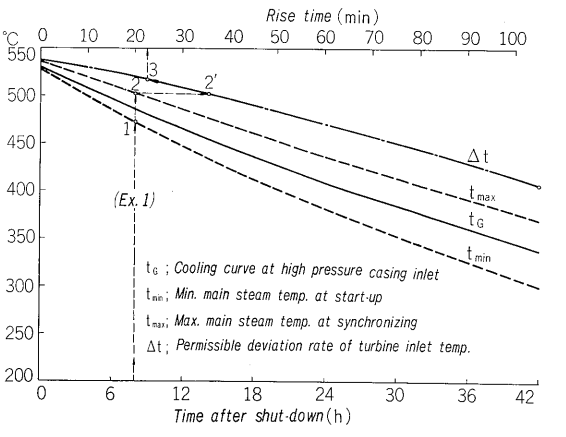


Fig. 19 **Main steam temperature and its rise time at start-up (600 MW)**

the thermal stress in the turbine casing metal is as small as possible even with frequent starting, and the life of the turbine is virtually unlimited. According to these curves, there is sufficient safety in the turbine even when started in the shortest time. The thermal stress is very low and the construction limit are never exceeded.

In the curves for both the 350 MW and 600 MW units, only one example is shown but in planning the turbine, the turbine was restarted after eight hours of stopping everyday to obtain the most frequent starting conditions.

For the 350 MW unit, the figure shows that the average casing temperature is 475°C. After the steam inlet temperature becomes 460°C, the turbine is started and loading begins at 495°C after speeding up and synchronizing. The full load condition is attained in 10 minutes and after this, the temperature rise time is 28 minutes until the rated steam temperature is reached.

After 8 hours of stopping for the 600 MW, the figure indicates that the unit is operated at a steam inlet temperature of 472°C with a high pressure casing inlet temperature of 486°C. After speeding up and synchronizing, loading begins at 502°C and the full load is attained after 13 minutes. Thereafter, the rise time is 23 minutes until the rated steam conditions are reached.

Figs. 20 and *21* show the turbine quick starting programs after 8 hours standstill. It is possible to carry out speeding up and synchronizing in a short time with no thermal stress problems but a certain time margin is provided for operation supervision.

4. Turbine Stress Analysis

In order to confirm the suitability of the previous curves, stress analysis were conducted in sections of the high and middle pressure turbine parts subjected to the most severe thermal stress conditions. The calculated results are shown in *Figs. 22* and *23*. As these figures show, the thermal stress of the high pressure casing wheel chamber in these calculations is the value for the outer casing. The thermal stress

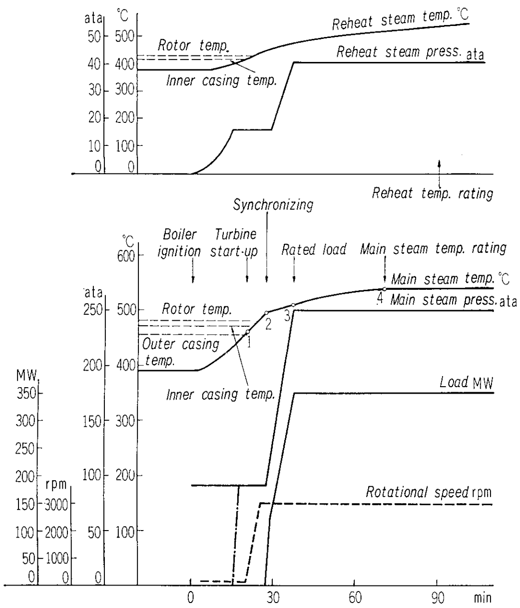


Fig. 20 **Turbine quick start-up diagram after 8 hrs standstill (350 MW)**

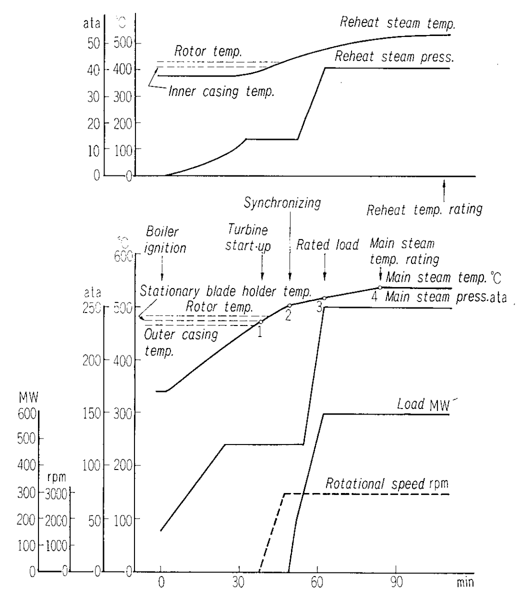


Fig. 21 **Turbine quick start-up diagram after 8 hrs standstill (600 MW)**

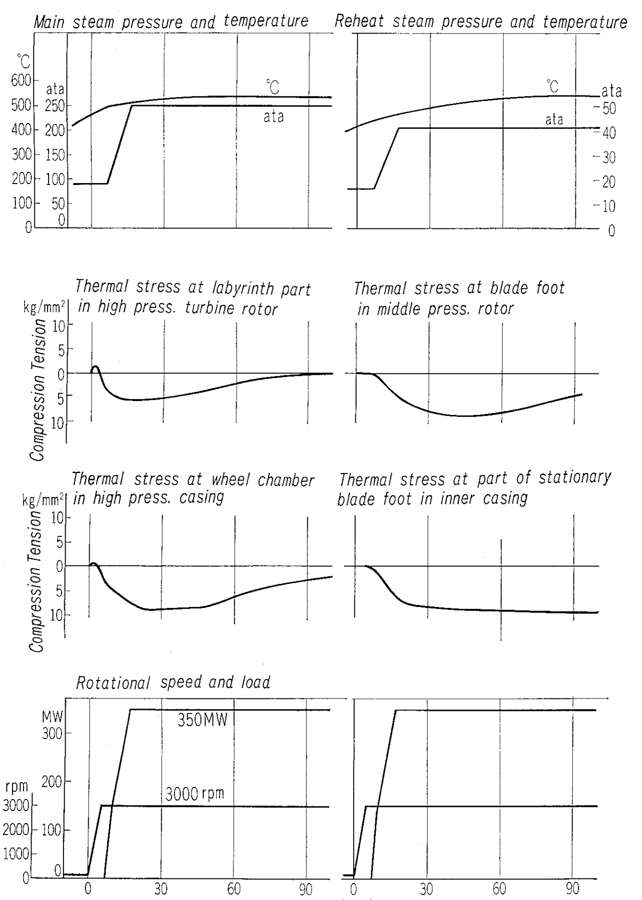


Fig. 22 **Thermal stress of turbine at start-up (350 MW)**

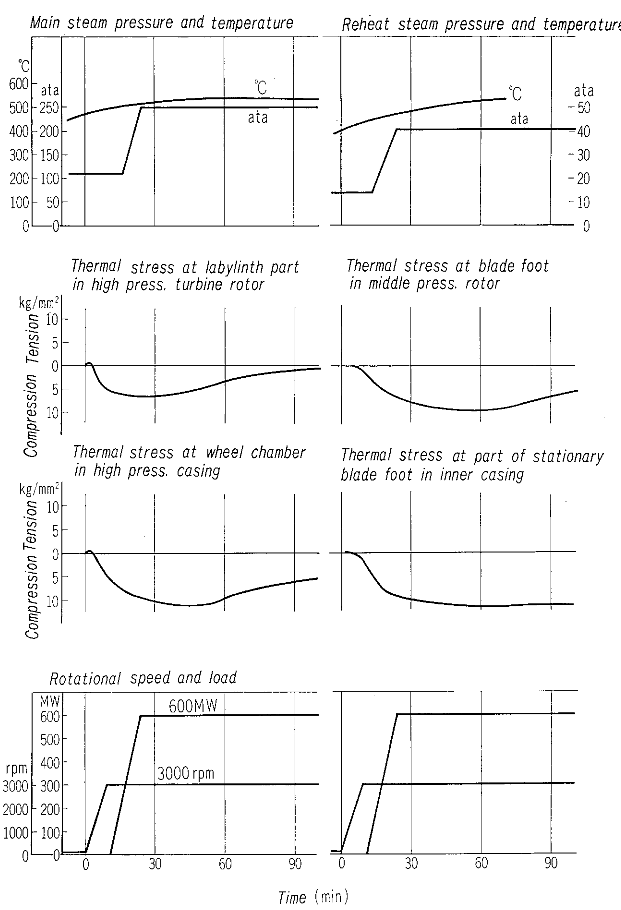


Fig. 23 **Thermal stress of turbine at start-up (600 MW)**

in other parts decreases in accordance with time. Thus even when there is very rapid starting, the thermal stress in all parts is kept at sufficiently low values. *Table 2* shows the life of each part in accordance with the calculated results.

Since it is possible to start up the turbine in a very short time due to the sliding pressure operation and these construction features, the start-up time for each unit is determined by the load following characteristics of the boiler. The start-up capabilities for the units are as shown in *Figs. 14* and *15*.

Table 2 **Max. thermal stress and life of turbine**

	350 MW		600 MW	
High pressuse turbine	Labyrinth part of rotor	Casing wall	Labyrinth part of rotor	Casing wall
Max. tensile stress (kg/mm²)	1.5	0.8	1.0	0.5
Max. compressive stress (kg/mm²)	5.7	8.8	6.5	11.6
Max. stress amplitude (kg/mm²)	3.6	4.8	3.75	6.06
Life (No. of premissible amplitude)	∞	∞	∞	∞
Middle pressure turbine	Blade foot part of rotor	Part of stationary blade foot in inner casing	Blade for part of rotor	Part of stationary blade foot in inner casing
Max. tensile stress (kg/mm²)	0.0	0.0	0.0	0.0
Max. compressive stress (kg/mm²)	8.8	9.5	9.5	11.7
Max. stress amplitude (kg/mm²)	4.4	4.75	4.75	5.85
Life (No. of permissible repeat)	∞	∞	∞	∞

Table 3 **Specifications of main parts**

	350 MW	600 MW
1. Boiler		
Type	Benson	Benson
Amount of steam generated	1,140 t/h	1,916 t/h
Main steam pressure (Superheater outlet)	260 kg/cm²·g	260 kg/cm²·g
Main steam temperature (Superheater outlet)	543°C	543°C
Reheat steam pressure (Reheater outlet)	42 kg/cm²·g	42 kg/cm²·g
Reheat steam temperature (Reheater outlet)	540°C	540°C
Feed water temperature	298°C	301°C
Combustion system	Heavy oil combustion	Heavy oil combustion
Air temperature	20°C	20°C
Ventilation system	Forced ventilation	Forced ventilation
2. Turbine		
Type	TC4F	TC4F
Generator output	350 MW	600 MW
Main steam pressure (Main stop valve inlet)	250 kg/cm²·g	250 kg/cm²·g
Main steam temperature (Main stop valve inlet)	538°C	538°C
Reheat steam pressure (Middle pressure stop valve inlet)	40 kg/cm²·g	40 kg/cm²·g
Reheat steam temperature (Middle pressure stop valve inlet)	538°C	538°C
Cooling water temperature	21.1°C	21.1°C
Condenser vacuum	722 mmHg	722 mmHg
No. of steam extraction	8	8
3. Generator		
Capacity	448 MVA	700 MVA
Power factor	0.85	0.9
Hydrogen pressure	4 kg/cm²·g	4 kg/cm²·g
Voltage	21 kV ±5%	21 kV ±5%
Rotational speed	3,000 rpm	3,000 rpm
Frequency	50 Hz	50 Hz
4. Capability		
Unit start-up time after 8 hrs standstill	ca. 56 min	ca. 74 min
Unit load-following capability	6 to 7%/min or over	6 to 7%/min or over
Minimum load	20% or less	20% or less

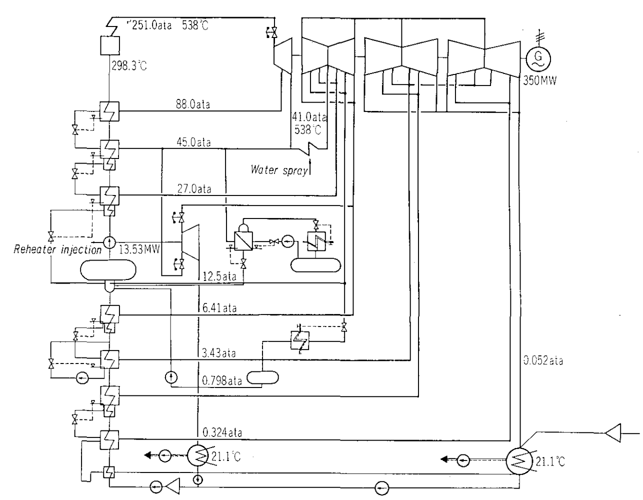


Fig. 24 Water-steam diagram (350 MW)

V. MAIN SPECIFICATIONS OF THE UNITS

From the results of the above investigation, the main specifications of the planned units were decided and are as listed in *Table 3*. *Fig. 24* shows an outline of the basic water/steam system for the 350 MW unit. The system for the 600 MW unit is basically the same. However, in the latter there are more feed water and condenser pumps and a desuperheater is provided at the outlet of the No. 3 high pressure feed water superheater in order to improve the heat rate.

VI. CONCLUSION

The equipment described in this article has excellent starting and stopping capabilities as well as load following characteristics. These characteristics are ideal for use in middle load thermal power stations. Due to lack of space, it was not possible to describe in detail all of the automation and control equipment and auxiliary devices. It is hoped that the article will be of use to the readers.