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# FUJI STEAM TURBINE FOR COMMERCIAL POWER STATION

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## I. FOREWORD

In meeting ever-increasing demands for electric power, steam turbines will continue to be one of the most important prime movers for production of electricity for considerable periods, whether the type of energy source is coal, oil or nuclear energy. In this article we will introduce Fuji Electric's large capacity steam turbine plants.

## II. STEAM CONDITIONS

The most important factor in the design of a large capacity steam turbine is the economy of the power plant. Many trials have been conducted to obtain optimum results in the following areas, such as combining a turbine with a drum boiler or Benson boiler :

Higher steam pressure and temperature
Higher reheat temperature
Improved efficiency of regenerative cycle

In the natural circulation boiler, the pressure limit is 140 kg/cm²g, and in the forced circulation boiler it is 170 kg/cm²g. Because of these conditions, an attempt to improve the thermal efficiency of the steam turbine combined with drum boiler was made by increasing the main steam temperature. The temperature was thus increased to 566°C, and all parts affected by high temperature, such as heaters, were made of austenite steel. However, this austenite steel is very expensive. Moreover not only is it inferior to ferrite steel in properties such as temperature shift and fluctuation, but it also presents problems such as vanadium attack in burning heavy oil and $SO_3$ corrosion, all of which occur as the temperature rises. Thus another system of increasing the steam pressure became highlighted, replacing the steam temperature increase method. That is, while suppressing the main steam temperature to below 538°C, the allowable limit for ferrite steel, its pressure is increased above 180 kg/cm²g. This pressure increase was accomplished without difficulty owing to the remarkable development of the Benson boiler by Siemens, providing thermal efficiency equal to that of a thermal plant of 566°C with the same capacity.

Even with the same thermal efficiency, this system is superior to the latter in low cost of manufacture and thermal flexibility.

## III. STEAM AND WATER PIPING SYSTEM

In order to insure high reliability and economy of the plant, careful examination has been made to determine the steam and water piping systems.

*Fig. 1* shows an example of the flow scheme.

As shown in this diagram, the turbine consists of three casings and its steam extraction performed in 8 stages. Water, after condensing, is led to the deacrator via two vacuum feed-water heaters and two low pressure feed-water heaters ; it is compressed by the pump in the extractor and is then led to three high pressure feed-water heaters. Water is heated to 275.4°C in these heaters, which is the final water temperature, and is supplied to the boiler. Under low load conditions of below ca. 30% load, steam supplied to the air extractor is automatically switched from the 4th to 2nd extraction and is maintained at 2 kg/cm².

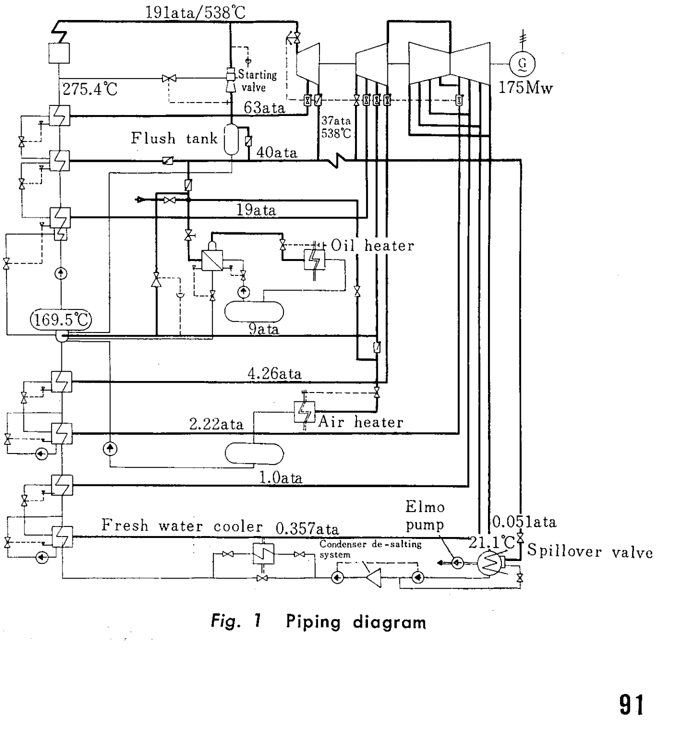



*Fig. 1* Piping diagram

However, under higher load conditions, the pressure of the air extractor depends on the turbine bleed pressure and varies accordingly. The low pressure and vacuum water heaters' drains are returned to the main feed water pipe by pumping to maintain maximum efficiency. As for its start up equipment, the boiler is equipped with a starting adjuster valve (steam connecting valve) which bypasses steam flow from the main pipe to the low-temperature reheating pipe, and a spillover valve which bypasses steam from the high temperature reheating pipe to the condenser. These valves are both capable of bypassing the entire volume of steam produced in the boiler, and consideration is given to minimize the necessity actuating the main steam and reheat safety valves in case turbine tripping occurs. In case trouble develops in the piping system, these bypass systems permit the plant to operate independently within its house load. It can be instantly returned to normal system operation upon removal of the trouble, thereby shortening the service interruption time.

## IV. PLANT LAYOUT

Layout of thermal power plant equipment varies according to several factors that differ from plant to plant, i. e. capacity, site condition, type of fuel, direction of power supply, direction of inlet/outlet of cooling water, etc. Careful planning is necessary to effect optimum layout under the existing conditions. Of the various factors to be taken into consideration, the following are the basic requirements to be met in layout of the plant;

(1) High operational reliability
(2) Safe and easy operation and control
(3) Simple and easy maintenance and inspection
(4) Effective combination of related components for smooth operation
(5) Efficient lighting and ventilation
(6) Length of pipes (especially steam pipes and cables) made as short as possible
(7) Housing occupying minium area and space
(8) Layout allowing easy installation of equipment and wiring

One of the most important considerations in meeting the above requirements is a clear-cut division of systems, such as water-feeding, steam, cooling, electricity, etc., with minimum crossover of these systems. *Figs. 2* and *3* show an example of the latest model of thermoelectric power plant, and *Figs. 4* and *5* show the basic layout planning of a plant.

## V. SUPERVISION OF WATER AND STEAM

To maintain maximum utility of the plant and to operate it at a planned thermal efficiency rate, efficient control of water-steam quality is a key factor. *Table 1* shows the standard values of water quality. Further, in order to maintain water quality required for the boiler and the turbine during the short period of the test run of the plant, and in case of water leakage from the condenser during normal operation, it may be necessary to install a de-salting system at the outlet of the condenser, which permits continued operation of the plant permissible under these conditions. Normally, this system is operated in one

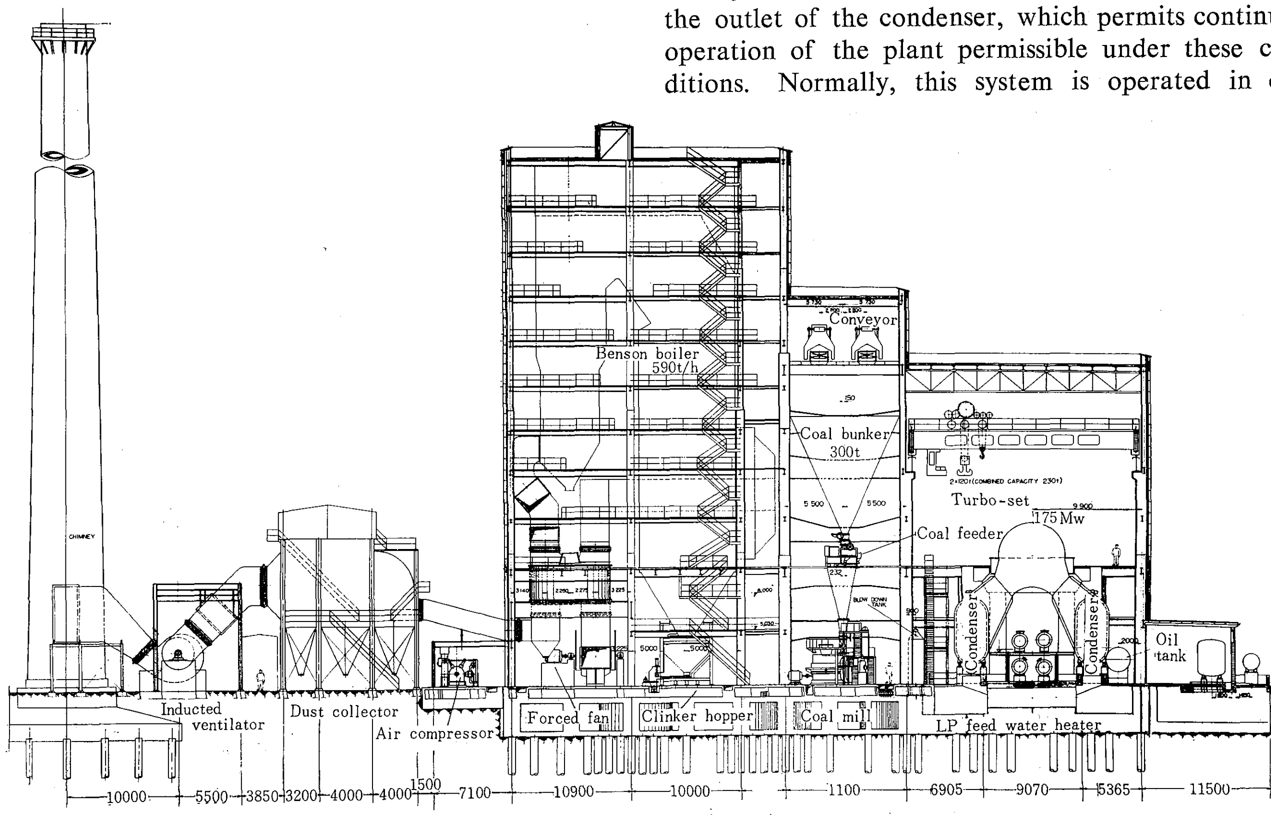
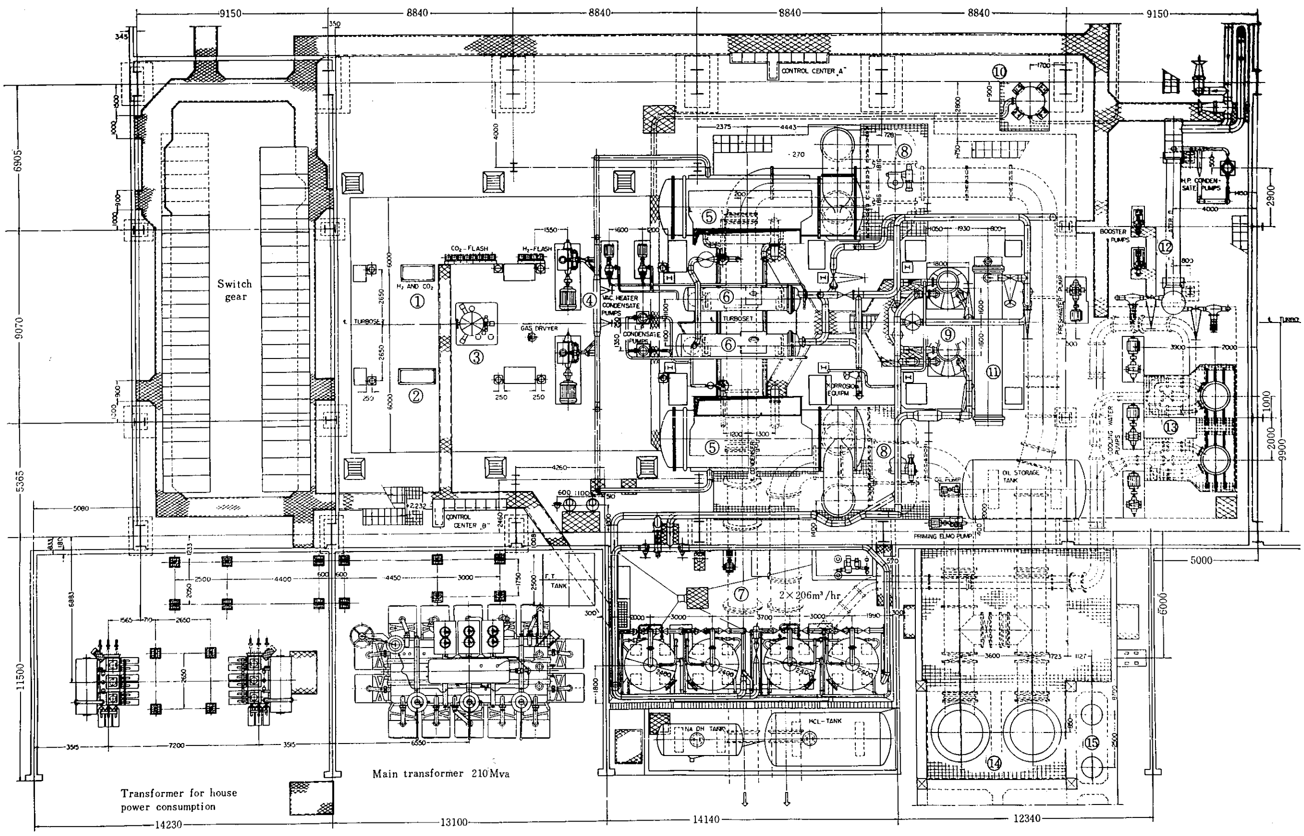



*Fig. 2* Section view of power station




① Hydrogen control sets
② Valve rack
③ Automatic gland seal steam controller
④ Elmo pump
⑤ Condenser
⑥ LP feed water heater
⑦ Condenser de-salting system $2\times205$ m³/hr
⑧ Reversing valve
⑨ Condenser pump
⑩ Blow down tank
⑪ Fresh water cooler
⑫ HP feed water heater
⑬ Cooling water cooler
⑭ Water circulation pump
⑮ Water pump for dust wash-down

*Fig. 3* Plan view of turbine room on 1st floor

***Table 1*** **Standard Values of Water Quality**

| Measuring Quantity \ Measuring Point | | Aux. Water Feeding | Water Feeding: At HP heater inlet | Water Feeding: At boiler inlet | Main Steam | Condensed Water |
|---|---|---|---|---|---|---|
| pH (20°C) | | ~7 | 8.3~9.5 | 8.3~9.5 | 8.3~9.5 | 8.3~9.5 |
| Conductivity | μ℧/cm | ≦0.1 | ≦0.2 | ≦0.2 | ≦0.2 | ≦0.2 |
| $O_2$ | mg/kg | — | <0.01 | <0.01 | <0.01 | <0.02 |
| $CO_2$ | mg/kg | Not measured | | | | |
| Fe | mg/kg | <0.010 | <0.020 | <0.020 | <0.020 | <0.020 |
| Cu | mg/kg | — | <0.005 | <0.005 | <0.005 | <0.005 |
| $SiO_2$ | mg/kg | <0.02 | <0.02 | <0.02 | <0.02 | <0.02 |
| Ammonia ($NH_3$) | mg/kg | — | 0.1~1 | 0.1~1 | 0.1~1 | 0.1~1 |
| Hydrazine ($N_2H_4$) | mg/kg | — | 0.05~0.20 | 00.5~0.2 | | |
| Overall Hardness | mg/kg | Not measured | | | | |
| $KMnO_4$ Consumption | mg/kg | <5 | <5 | — | — | — |
| Oil | mg/kg | Not measured | <0.3 | — | — | — |
| Natrium+Potassium | mg/kg | | | | <0.01 | |

channel, and upon initial plant operation or condenser leakage the full quantity of condensation is passed through two channels. Also, it is recommended that the condensing pump be specifically designed in such a way as to lead the condensed water from the 1st stage of the pump to the con-

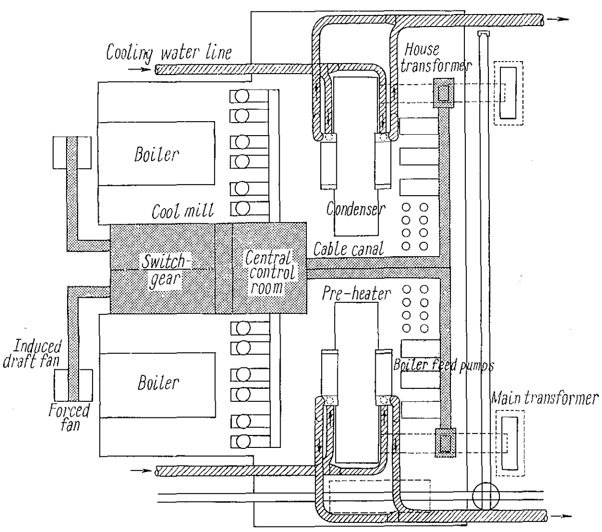



*Fig. 4* Typical plan for cable and cooling line

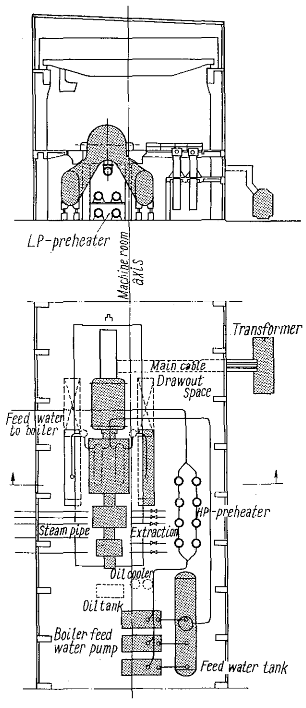



*Fig. 5* Typical plan for piping line

denser de-salting system, where it is processed and returned to the 2nd stage of the pump for further pumping up and then sent to the air extractor, so that the withstanding capability of the de-salting system may be maintained economically. The standard values of water processed in the de-salting system are as follows:

$SiO_2$ $<0.01$ ppm
Conductivity: $\mu\mho$/cm $<0.1$
Fe $<0.01$ ppm
Cu $<0.003$ ppm

## VI. STEAM TURBINES

The main characteristics of the reheat condensation turbine plant can be enunciated as follows;

(1) For reasons of economy, the steam temperature at the turbine inlet is maintained at 538°C, which is allowable for ferrite steel, and the steam pressure maintained at more than 180 kg/cm$^2$ by taking advantage of the Benson boiler constructed in this plant.
(2) The multiple casing structure consists of high, medium and low pressure turbines, with the high pressure turbine of pot-type structure.
(3) Single, cut-out moving blades with no riveting of shroud rings or no silver soldering of lacing wires.
(4) It is equipped with a high-speed, oil hydraulic turning device.
(5) An electrically remote-controlled drainage station is installed.
(6) All operations, such as starting and stopping, are remote-controlled from the central control room.
(7) A double condenser is employed (*Fig. 18*).
(8) It has fully automatic spillover system which serves for quick starting of the boiler and minimizing of starting loss.
(9) It has an Elmo vacuum pump with air ejector.
(10) It has an elastic-supported super-lightweight turbine foundation.

Summarizing the above, another major characteristic of the plant in addition to its high economy with low construction and operation cost is its ease of operation and capability of withstanding large load fluctuation compared to past products. *Figs. 7* through 10 are sectional diagrams of large capacity reheat condensation turbines of standard construction.

Each casing of the turbine is made as compact, lightweight and thin as possible, and the structure and material selected for each turbine is the optimum for the required conditions of steam, pressure and temperature.

### 1. High-pressure Turbine

The pot-type structure high pressure turbine was discussed in detail in a previous report. Therefore, only an outline will be given here.

The degree of heat elasticity of the entire machine mainly depends on the structure of its high pressure portions. The pot structure employed in this plant is most suitable for high-pressure high-temperature turbines with favorable performance records in turbines of steam pressure up to 300 kg/cm$^2$g for many years. The first structural problem encountered in the high-pressure high-temperature turbines is the design of its horizontal flange part. In pot-casing turbines, however, all contacting surfaces perpendicular to the axial direction are of the ring type of ideal rotational symmetry, using no flange, bolts or

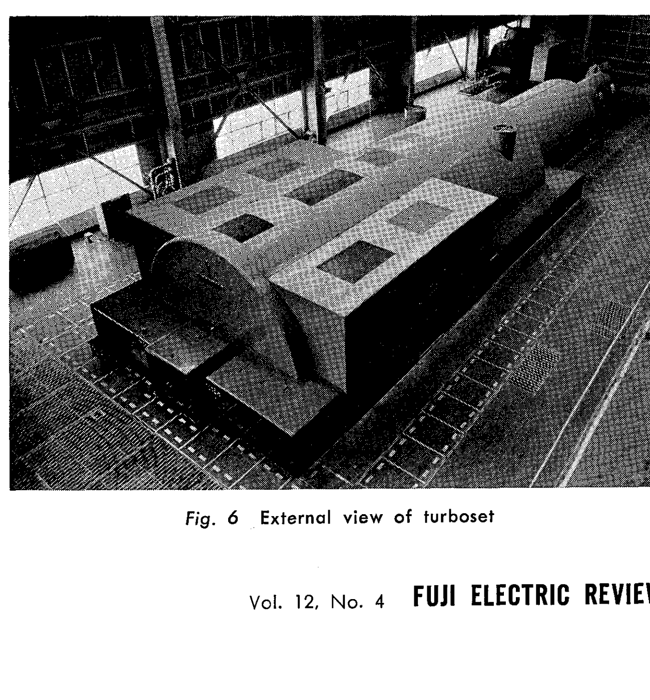


*Fig. 6* External view of turboset

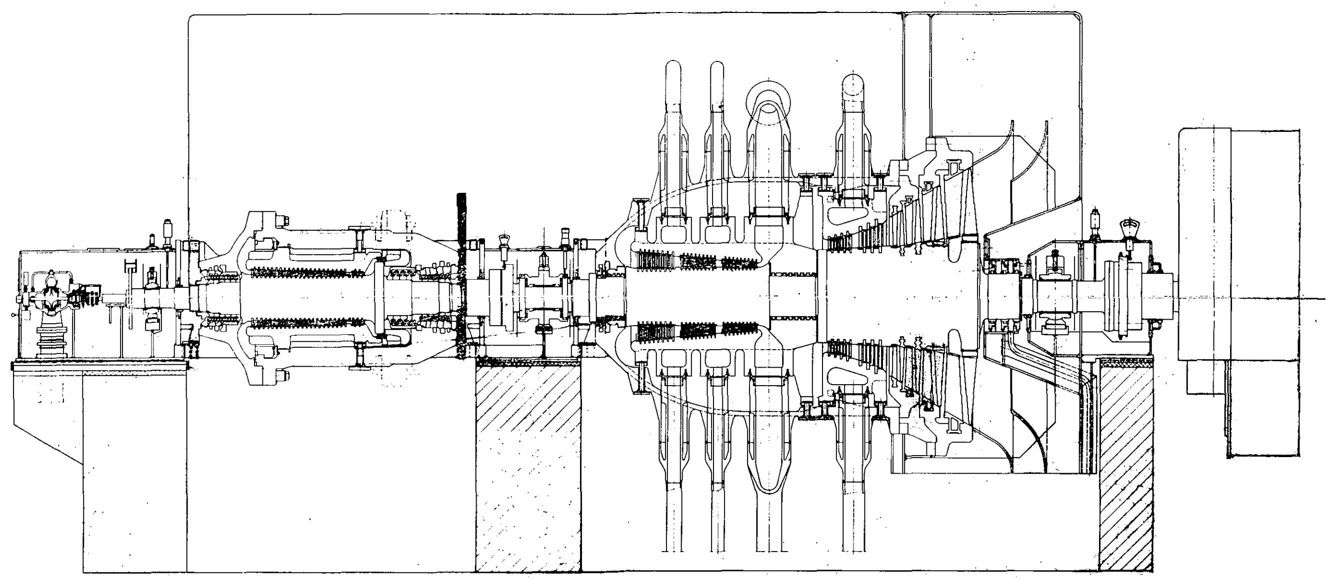


*Fig. 7* 75,000 kw reheat condensation turbine

nuts for connection portions of high-pressure high-temperature parts. The second problem encountered is the design of the steam inlet chamber and nozzle chamber. In our turbine, four nozzle chamber blocks are forged separately and later inserted into the pot-type casing. This insertion type nozzle chamber eliminated the problem encountered in conventional turbines, providing lightweight nozzle chambers made of thin material which permit free thermal expansion. (*Fig. 11*). The divided-type stationary blade holder, which is tightened by chamber pressure from outside, is held by a bayonet ring against thrusts in the axial direction, while being maintained by the pot casing so as to allow its flexibility in the axial and radial directions, also its mass is balanced with that of the rotor in order to equalize the rate of thermal expansion of these components. This is an application of the " mass balancing " principle. Further, besides its effective steam-tightness, the labyrinth gland in the radial direction has the advantage of securing a large elongation difference between the rotor and the casing.

### 2. Medium-pressure Turbine

This type of turbine is of the double casing structure, and its steam flow system is either the " opposed flow" (*Fig. 8*) or "double flow" (*Fig. 9*) system. (In the former, the reheat steam first flows into the center of the casing, passing in front of the forward half of the reaction blade rows; changing direction here, it passes through the steam chamber between the inner and outer casing chambers and flows into the rear half of the reaction blade rows.) This system insures uniform warming of casings upon starting or load fluctuation, as well as cooling of the outer casing.

The inner casings and stationary blade holders are both supported on the outer casing by means

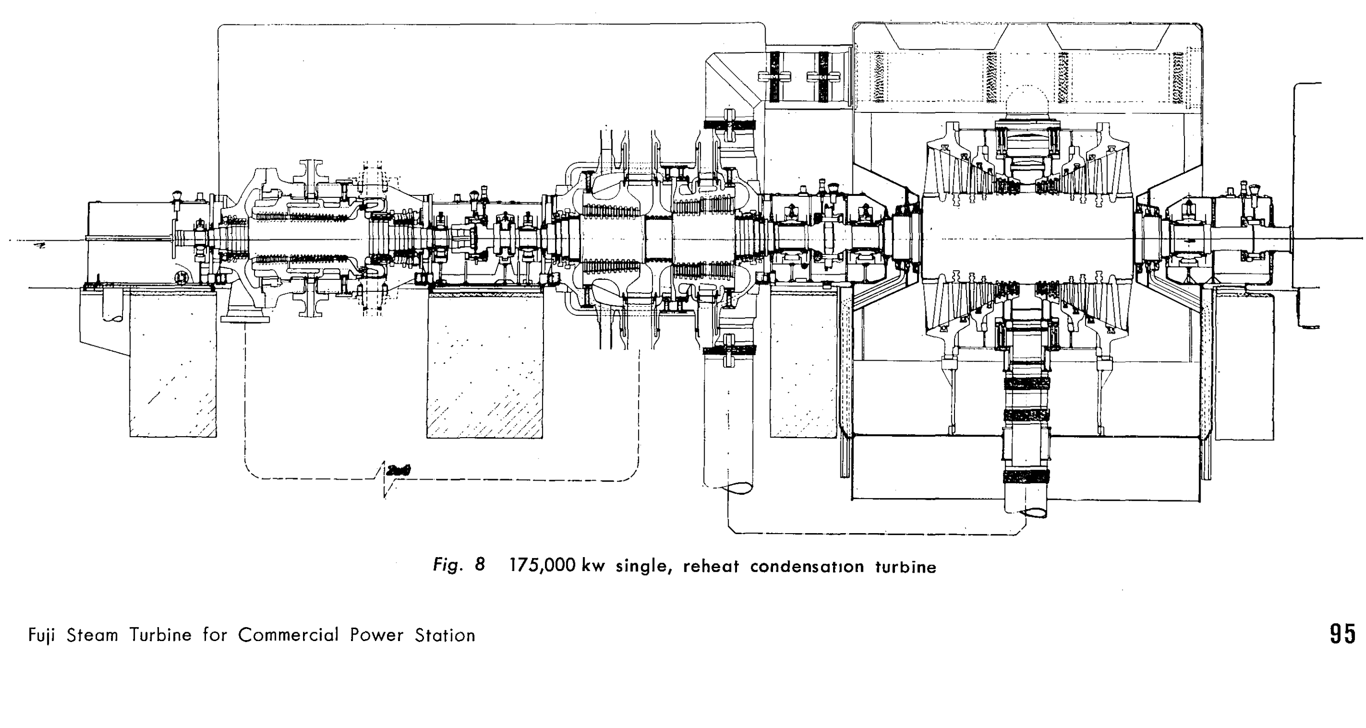


*Fig. 8* 175,000 kw single, reheat condensation turbine

of centripetal supports, with flexibility maintained for temperature variation and freedom from axial center deviation. Both high and medium pressure casings are center-line supported against the bearing stand, so that the axial centers of the rotor and the casing are always aligned without danger of misalignment during operation.

### 3. Low-pressure Turbine

Careful examination was made to determine the structural desigan of the low-pressure turbine, with adoption of a standard series of final stage blades, with the longest blade of size 825 mm, and a suitable exhaust flow rate to bring about optimum conditions for the plant. Casings are double casings of welded steel plate. A thorough simulation test was conducted primarily concerning the stress and distortion of each part due to the weight of the rotor and inner casings and vacuum load, and for vibration frequency characteristics during operation. The exhaust side of the turbine is directly welded to the double condenser into one unit. These casings are fixed to the turbine foundation by round keys underneath the forward bearing stand, and extend forward. The Mitchel type thrust bearing is located between the high and low pressure casings, and rotors expand toward their respective casings, i. e., HP rotors to the front and low-medium pressure rotors to the rear, thereby minimizing the rotor expansion range difference.

### 4. Blades

Impulse blades used for the governor high pressure No. 1 stage are twin blades, i. e. two blades welded in opposition to each other. These blades are imbedded in precision-cut T-slits of the blade wheel, maintaining a high degree of safety against impulses due to partial flow of high-pressure high-temperature steam acting on the blades. *Fig. 13* shows an example of reactionary blades used for the forward portion of high and medium pressure turbines. Irrespective of pressure level, no rivets are used to install shroud rings on the moving blades.

Shrouds are scraped out one by one for each blade and are imbedded in T-slits distributed around the circumference of the rotor, forming a ring when connected. The diagram illustrates an extreme of these shroud rings fitted with steamtight fins which decrease loss due to blade tip steam leakage through the labyrinth effect. One of the characteristics of the turbine is the employment of a blade lattice of small average diameter for the medium-high pressure turbine, providing radial clearance sufficient for operational safety requirements without causing excessive blade tip leakage loss. This design is also employed for the axial clearnace between blade lattices.

Neither shroud rings nor lacing wires are used for low pressure part blades. Moreover, protection against erosion due to the drain is provided by

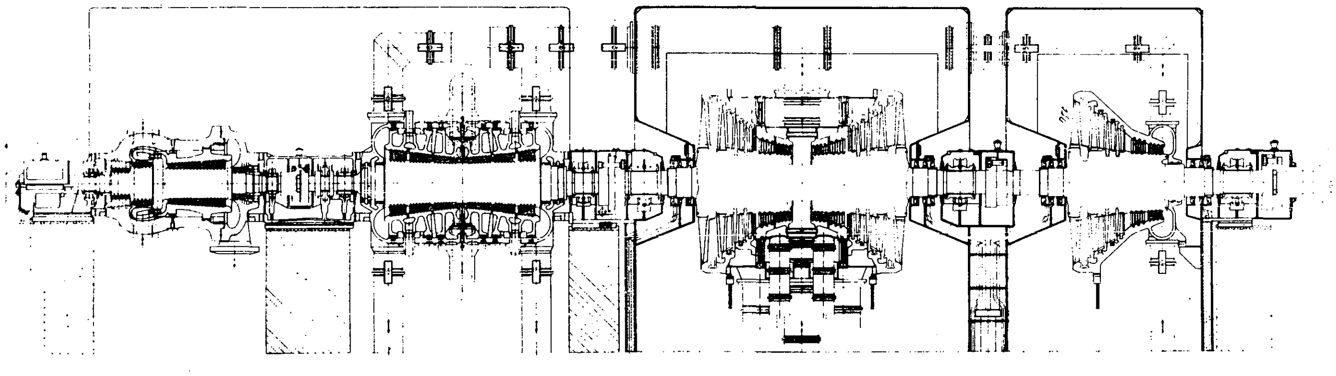


*Fig. 9* 300,000 kw single, reheat condensation turbine

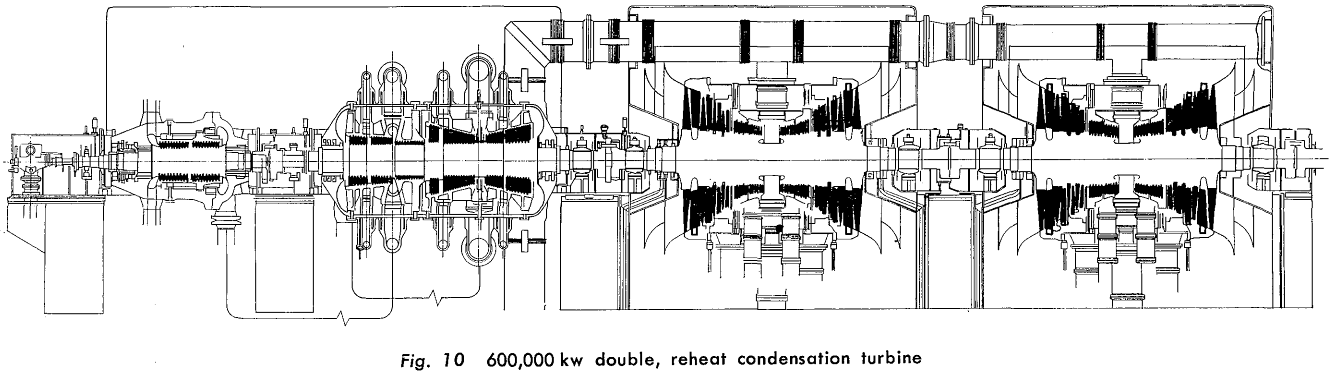


*Fig. 10* 600,000 kw double, reheat condensation turbine

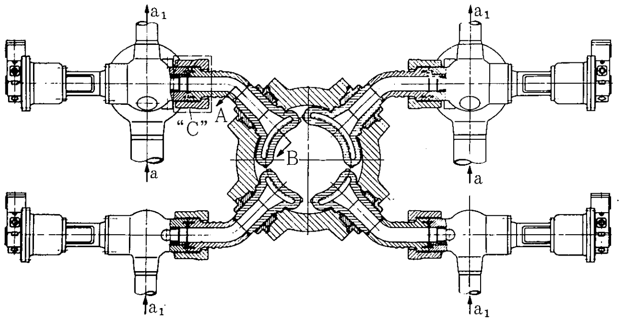



*Fig. 11* Separate nozzle boxes

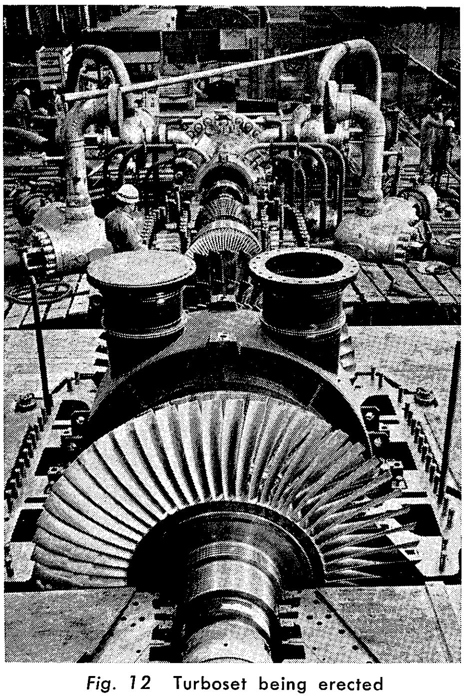


*Fig. 12* Turboset being erected

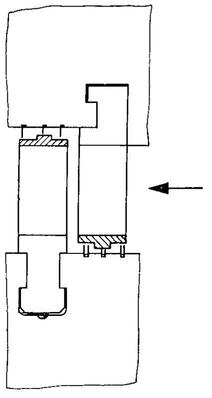


*Fig. 13* Example of reaction blade rows

special gas hardening applied to blade tips to increase their surface hardness, stellite faced not being used for this purpose. Each blade is independently imbedded in the rotor as in an aircraft propeller, its natural frequency being strictly adjusted individually to avoid the danger of resonance caused by steam induced frequency during operation. *Fig. 14* illustrates an actual measurement of the natural frequency of the last blade during operation using a test rotor. *Fig. 15* illustrates the distribution of the low pressure blades; particularly for the moving blades of the last stage, arc-type Christmas tree shaped legs are installed in the axial direction to withstand the enormous centrifugal force imposed on the blades during operation. The blades are designed in accordance with the hydrodynamic principle of mass flow density. Consideration is given to the selection of the blade lattice of the high-speed blade tips, concerning which high peripheral efficiency is difficult to attain. In extensive research and tests, including simulation tests with a shallow stream and trials at high speed, small model turbine blades have proven their superior quality.

### 5. Turning Device

The purpose of the turning device is to perform high-speed turning at approx. 100 rpm with high efficiency by means of the oil hydraulic turbine, thereby preventing the non-uniform distribution of temperature that occurs inside casings upon cooling, as well as preventing deformation of the casings in the vertical direction which hampers normal starting.

This device also serves to increase safety and efficiency by precluding the necessity for a separation system upon acceleration or temporary halting of the machine upon stopping or switching to turning device operation.

## VII. CONTROL AND SAFETY DEVICES

Installation of additional steam lines through which excess steam flow produced in the boiler by large

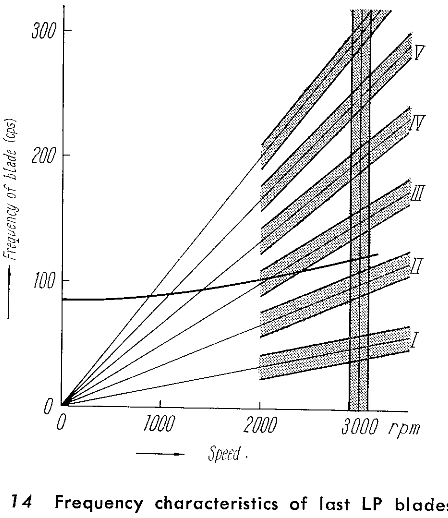



*Fig. 14* Frequency characteristics of last LP blades

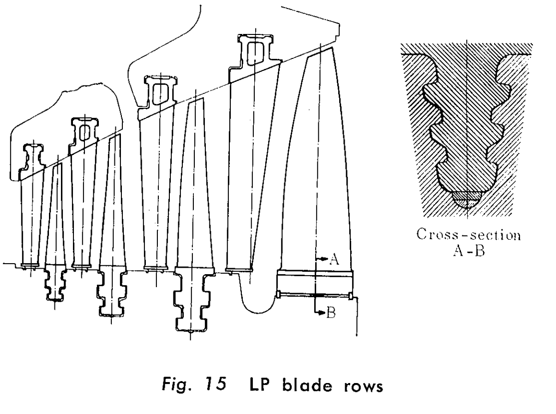
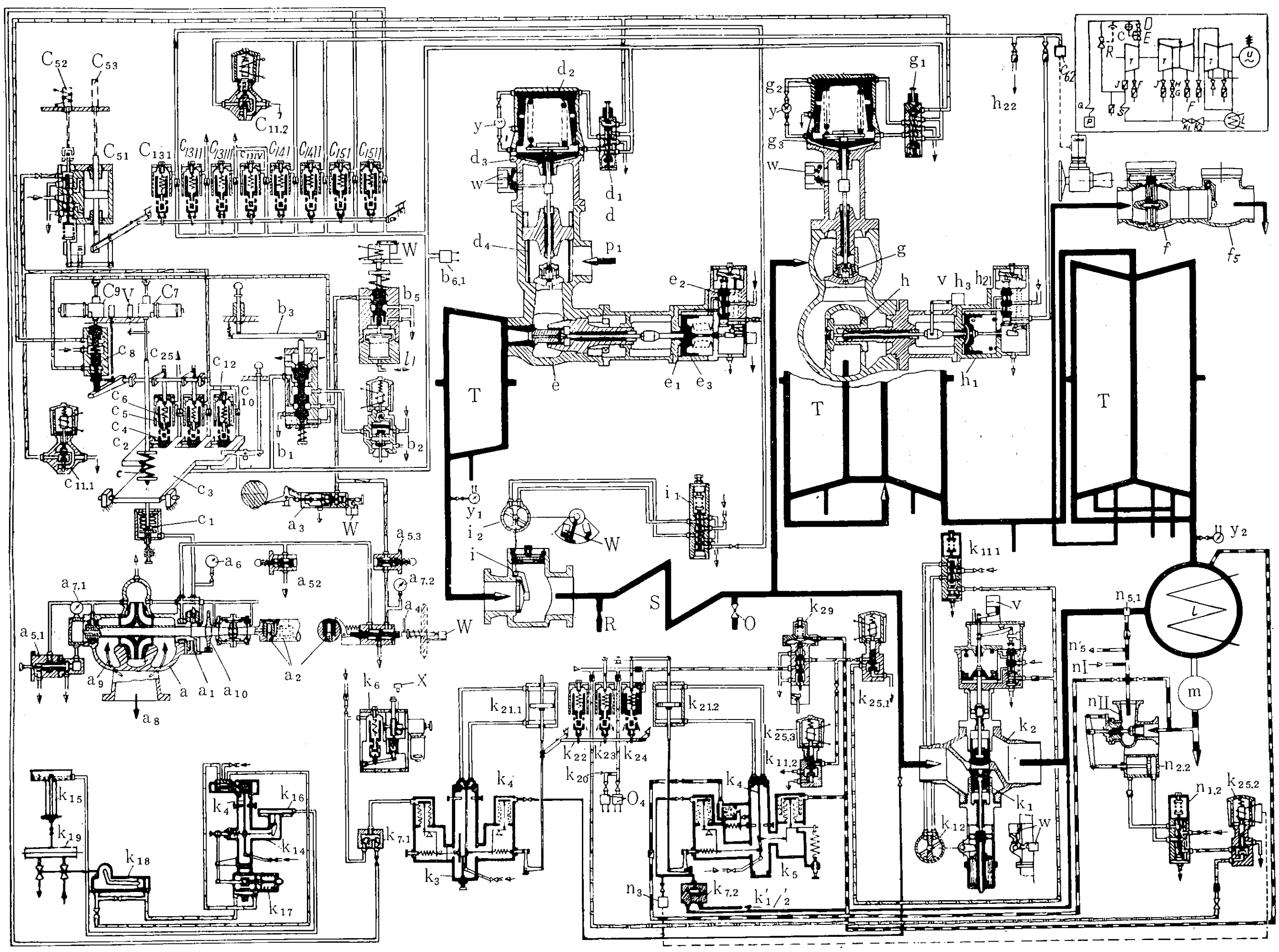



*Fig. 15* LP blade rows

load fluctuations, starting or stopping of turbine, can be recovered in the condenser bypass turbines of high, low and medium pressures, increases the safety not only of the turbine itself, but also that on the boilers and safety valves. It also helps to maintain the stability of the steam temperature at the transition stage. The low-pressure bypass system inside this bypass circuit, or the spillover system, is regulated by the control system of the turbine itself. *Fig. 16* shows a system of such control and safety devices.

The governor is of the oil hydraulic type ; initial oil pressure generated by the governor impeller is amplified to the auxiliary 2nd oil pressure at this point and then transformed to the 2nd oil pressure of multiple channels which impels each control valve in the system. High-pressure and medium-pressure control valves are regulated to let an almost equal amount of steam flow into the high and medium-pressure casings, respectively by the governor at up to 30% load. The medium pressure control valve closes the spillover valve when 30% load is exceeded and fully opens at 50% load to eliminate throttle loss under high load conditions. The high pressure control valve is installed close to the casings in order to prevent overspeed upon load dump which is caused by steam remaining in the wheel chamber. Another overspeed prevention device employed in this system is a load shedding relay, which electrically detects large load fluctuations above 65% (load drop) and temporarily shuts off the 2nd oil pressure for the high and medium pressure control valves and fully closes these valves until the governor is set to operate. By this system, it is possible to keep the rate of speed increase low even under full load dump conditions. The starting device provides such func-




- $a$: Main oil pump
- $a_1$: Governor pump
- $a_2$: Emergency governor
- $a_3$: Thrust bearing protector
- $a_5$: Emergency governor tester
- $b_1$: Emergency stop device
- $b_2$: Remote controlled emergency stop device
- $b_5$: Condenser protective device
- $c$: Speed control device
- $d$: HP emergency stop valve
- $c_7$: Speed setting device
- $c_{11}, c_{12}$: Magnet valve for load shedding relay
- $e$: HP control valve
- $f$: Circular disk type bleeder valve
- $f_5$: Bleeding check valve
- $g$: MP emergency stop valve
- $h$: MP control valve
- $j$: HP outlet check valve
- $k_{1,2}$: Spillover valve
- $k_3$: Spillover relative pressure control system
- $k_5$: Spillover pressure limiter
- $k_6$: Impulse limiter
- $k_{14}$: Automatic gland seal steam controller

*Fig. 16* System of control and safety devices tions as opening and closing of the high-medium pressure emergency stop and control valves upon starting, remote control of resetting the emergency stop system; it also serves as a load limiting device during operation.

A check valve for automatic oil pressure control is provided at the outlet of the high-pressure turbine in order to prevent counterflow of steam from the reheat circuit at the warm-up or initial stage of operation. At below 30% turboset load, the spillover control device regulates the spillover valve so as to allow excess steam from the boiler to overflow into the condenser, while maintaining the reheat pressure close to the steam pressure in proportion to its 30% load. When the turboset is loaded above this, the spillover valve is automatically closed. This necessitates raising the pressure setting by the load of turboset; accordingly, the spillover pressure is set in conformity with the 2nd oil pressure for spillover control (*Fig. 17*). Another function of the spillover control device is the control of cooling water valve to lower the temperature of the spillover steam. In this plant, this cooling water valve is regulated to open fully prior to the opening of the spillover valve in order to insure safety. In either case, some safety mechanism is indispensable for the condenser into which the steam of high temperature flows directly.

A spillover decompression control device is installed for this purpose, which either checks or trips the spillover valve in the control device in any of the following cases; (1) spillover capacity is exceeded, (2) the degree of vacuum in the condenser has dropped, (3) cooling water pressure build-up is insufficient. The spillover system is provided with sufficient capacity to dispose of the entire volume of steam which is generally produced in the boiler.

## VIII. CONDENSER UNIT

### 1. Condenser

Although its function of heat transmission remains the same as in conventional products, the condenser used for a large-capacity turbine is unique in its structural design. This condenser, called a double-type condenser, has cooling tubes divided into two lines which are laid outside the turbine foundation parallel to the axis of the turbine. (*Fig. 18*). As one of its characteristics, the condenser is welded with the low-pressure casing into one unit; its route is of the diffuser type, i. e., it deploys toward tube bundles. This design minimizes exhaust losses by transforming the velocity energy of exhaust into pressure, at the same time increasing the number of cooling tubes bundles initially in contact with the steam flow. Another unique setup is the position of the low-pressure feed-water heater; it is located between two condensers and directly beneath the low-pressure turbine cylinder. This setup provides bleeder exhaust pipe length. The weight of this unit is considerably reduced as a result of rational application of arc and reinforcement principles, with lighter load borne by the supporting springs.

The distribution of the cooling tubes is such that they are divided into bundles of long, thin pipes running almost parallel with steam flow; a plate is inserted between the bundles to divert water flow to prevent drenching of bundles.

Also, as the spillover steam from the reheat line cannot be led directly into the condenser intact because the spillover valve regulates only its flow quantity and does not alter the pressure or temperature of the steam, an expansion nozzle is provided on the body, which lowers the pressure and introduces condensed water from the middle stage of the condensing pump to cool the spillover steam, allowing it to flow into the condenser at a safe temperature and pressure (*Fig. 19*).

This system has demonstrated excellent performance despite its simple structure.

### 2. Air Extractor

An Elmo vacuum pump with air ejector is em-

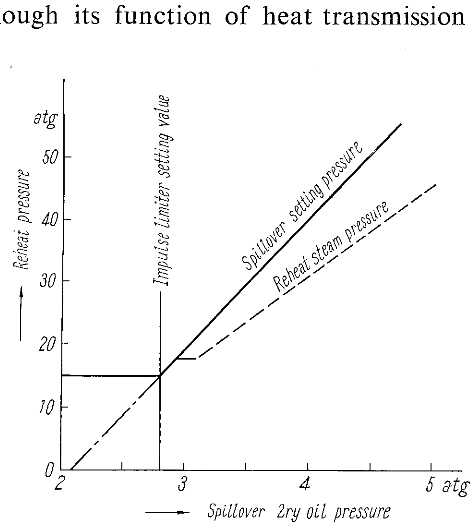



*Fig. 17* Setting of spillover pressure

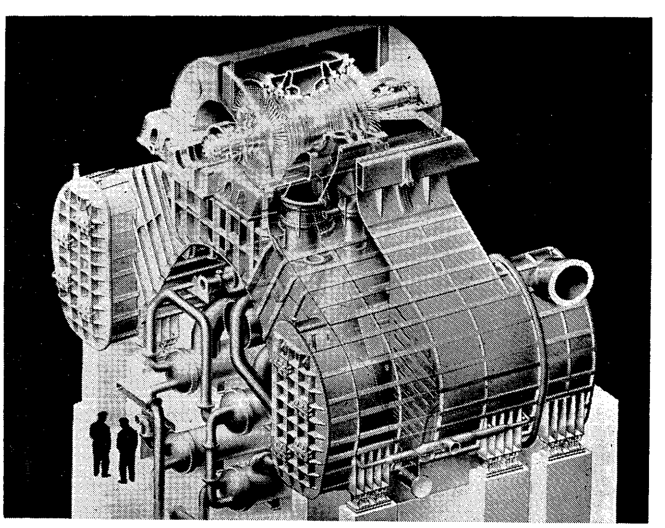


*Fig. 18* Condenser

ployed to extract air that leaks into the condenser. The Elmo water-seal pump, which has no sliding part other than the water sealing gland, insures safe and trouble-free operation through the use of an atomosphere-driven air ejector which maintains high efficiency at a high degree of vacuum as well as preventing cavitation. Moreover, the air ejector does not operate at a low degree of vacuum, the Elmo pump operating independently alone, precluding the necessity for a mechanism like a starting ejector. That is, the Elmo pump functions as a constant-volume pump covering atmospheric to high-vacuum conditions, so that the amount of air extracted at low vacuum becomes relatively large, and with shorter vacuum build-up time, approximately 40% vacuum can be obtained even prior to introduction of the axial sealing steam. When the vacuum starts to built up, the air ejector is set to perform preliminary pump operation. Its switching can be performed automatically through pressure switch or magnetic valve control.

Therefore, this air ejector can be remote-controlled from the central control room.

### 3. Drain Station

A considerably large number of drains are used in large-capacity turbines, and control of them presents another problem, especially upon starting. In our plant, these drains are classified into two series, high and low pressure. A drain station is provided to simultaneously operate the drain valves, which facilitates consolidated control from the central control room.

## IX. TURBINE FOUNDATION

In past products, mainly rigid foundations of high characteristic frequency have been utilized. However, even with this rigid turbine foundation, several vibration frequency peaks exist below normal operational speed due to the ground condition. In view of this, adoption of an elastic foundation for turbine mounting is more effective in many instances. *Fig. 20* shows an example of an elastic turbine foundation (175 Mw turbine). On its first floor, a mat is installed which is connected with numbers of steel pipe frames that remain durably fastened to the ground. The frequency of the foundation with respect to the machine rotation is properly selected to obtain allowable amplitude, and to prevent non-uniform sinking liable to occur after a long period of operation. Supporting poles are made thin, based on the concept that they are dampers to maintain the weight of the superstructure. Also, to prevent unintended vibration, the entresol for the isolate booth is supported by a pole independent of the turbine foundation. Thus adoption of elastic reinforced concrete for the turbine foundation has reduced the amount of concrete needed to 60% that of a rigid foundation for effective provided ample space underneath the foundation for effective use.

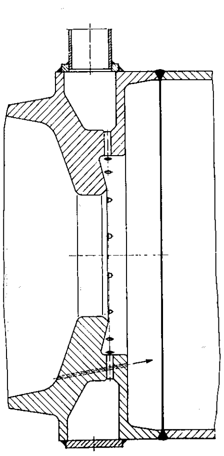


*Fig. 19* Spillover expansion nozzle

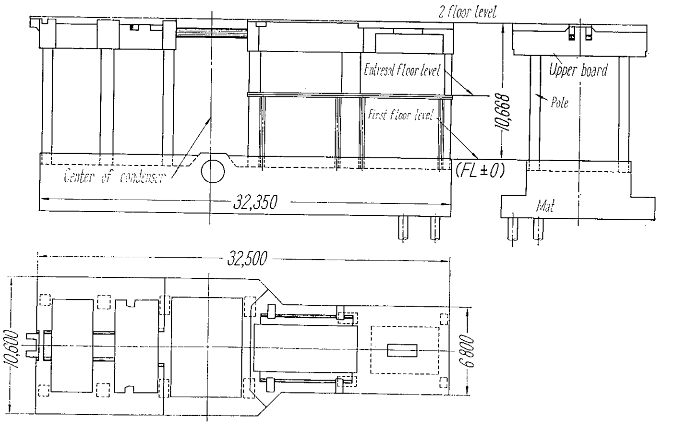



*Fig. 20* Turbine foundation

*Table 2* Examples of Bearing Vibration Measurements

Unit: 1/1000 mm

| Load(%) \ Bearing Number | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|---|
| 50 | — | 2.2 | 6.6 | 7.0 | 4.4 | 4.2 | 6.5 | 2.5 |
| 75 | 4.0 | 1.5 | 4.5 | 4.0 | 1.2 | 3.5 | 5.0 | 2.0 |
| 100 | 3.5 | 1.8 | 3.6 | 1.5 | 1.6 | 2.8 | 3.5 | 1.6 |

## X. OPERATION CHARACTERISTICS

### 1. Turbine Starting

*Fig. 21* shows an example of cold starting of a 175 Mw type turbine. The time required for starting is relatively short; i. e., 35 minutes from steam injection to paralleling and 25 minutes from paralleling to 75 kw loading. Even with this starting, factors such as relative elongation and casing wall temperature difference are within allowable limits. These properties display well the superior starting capacity of these turbines. Warming of the high-pressure turbine upon cold starting is performed simultaneously with the warming of the main steam pipes, with the high-pressure emergency stop valve and the high-pressure control valve both open. Upon completion of warming of the high-pressure casings, the machine immediately commences warming start at a low speed of 1500 rpm. This unique system greatly reduces the starting time of the entire turbine plant. *Fig. 22* shows an example of hot starting after a 12-hour shutdown period; 15-minutes from steam injection to paralleling, and 35-minutes to 120,000 kw loading. The high-pressure chamber temperature difference at load increase is sufficiently small, indicating the possibility of further shortening of the time required for loading.

## 2. Quiet Running

Quiet operation is insured throughout all operating conditions, including starting and load increasing. *Table 2* shows an example of bearing vibration measurements of 175 Mw turboset at each load level.

## 3. Governor Characteristics

*Fig. 23* shows oscillograms of 2/4 and 4/4 load dump tests. As is evident from comparison of these oscillograms, the load shedding relay is actuated when the load exceeds 65% and the dead time of the governor is conspicuously shortened. Even in the full load dump condition, the maximum momentary acceleration rate is kept as low as 7.0%.

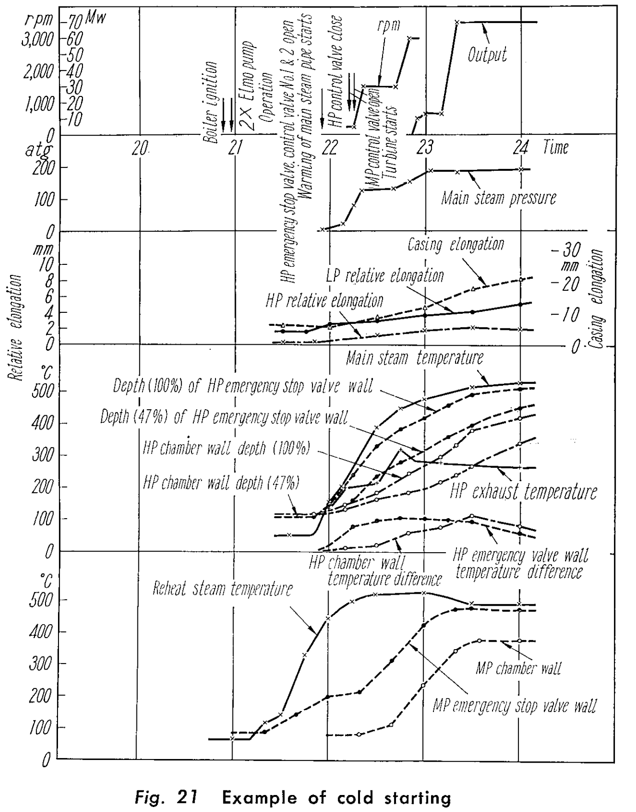



*Fig. 21* Example of cold starting

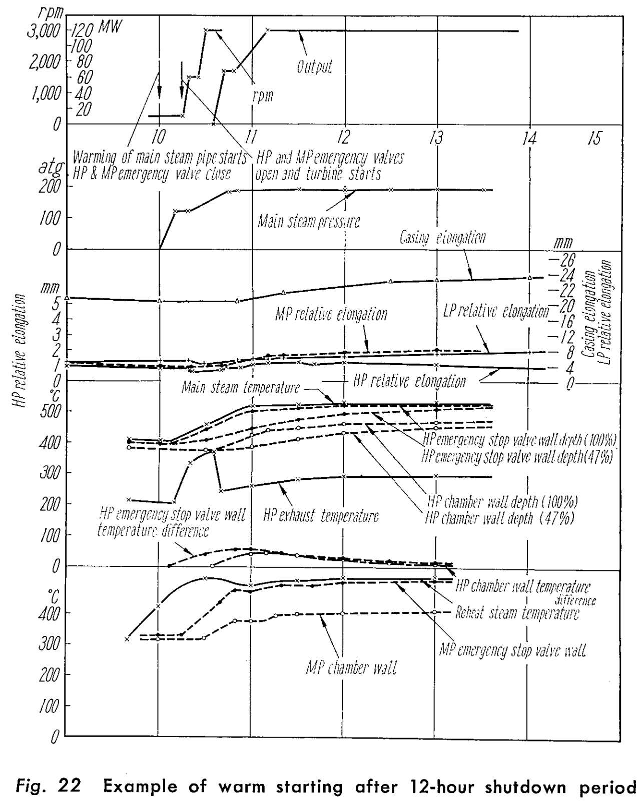
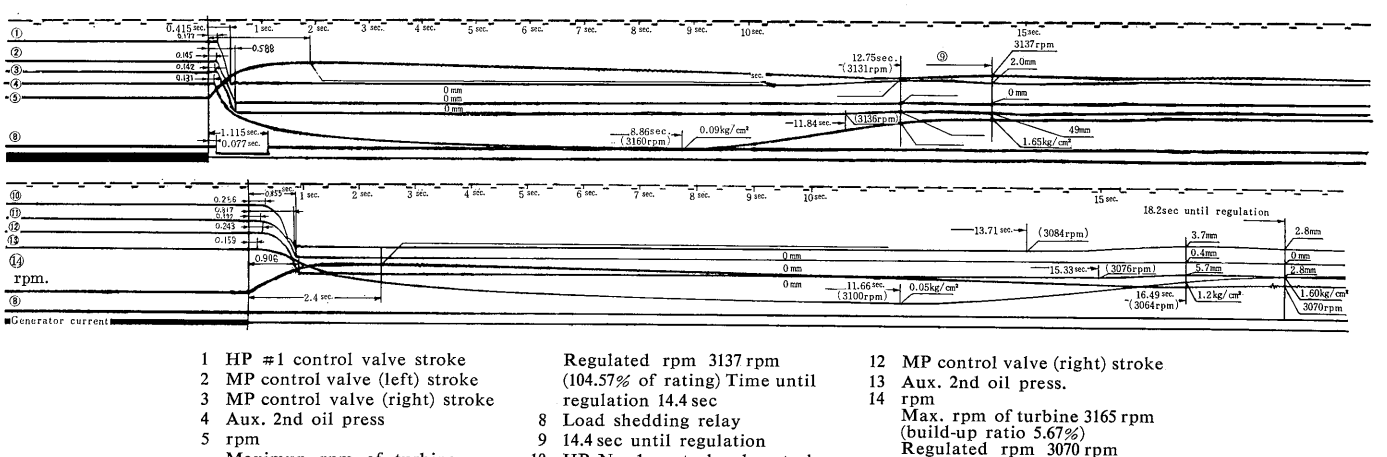



*Fig. 22* Example of warm starting after 12-hour shutdown period




1 HP #1 control valve stroke
2 MP control valve (left) stroke
3 MP control valve (right) stroke
4 Aux. 2nd oil press
5 rpm
Maximun rpm of turbine 3210 rpm (build-up ratio 7.00%)
Regulated rpm 3137 rpm (104.57% of rating) Time until regulation 14.4 sec
8 Load shedding relay
9 14.4 sec until regulation
10 HP No. 1 control valve stroke
11 MP control valve (left) stroke
12 MP control valve (right) stroke
13 Aux. 2nd oil press.
14 rpm
Max. rpm of turbine 3165 rpm (build-up ratio 5.67%)
Regulated rpm 3070 rpm (102.50% of rating) Time until regulation 18.2 sec

*Fig. 23* Oscillograms of 2/4- and 4/4-load dump tests