Unit 1 Steam Turbine of Noshiro Thermal Power Plant, Tohoku Electric Power Co., Inc.

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1. Introduction

Unit 1 Steam Turbine of Noshiro Thermal Power Plant, Tohoku Electric Power Co., Inc. has a 4 cylinder, tandem-compound construction composed of barrel-shaped high-pressure cylinder and intermediate-pressure cylinder as well as two low-pressure cylinders.

The unit, which has set the rating record for 50Hz systems in Japan, fundamentally has excellent middle-load power generating functions. The most important points in manufacturing the unit are high efficiency and enlargement of the combined rotor system, which consequently stabilize the total system. Since the initial steam injection at the end of 1992, the commissioning operations and tests have shown excellent results.

The technical features of the steam turbine facilities and the design reviews, which was performed to ensure high quality of these features, as well as results of the test operations, are described in the following. **Table 1**, **Fig. 1** and **Fig. 2** show the specifications, sectional and plane views of the turbine.

2. The Technical Features of the Steam Turbine

2.1 Construction

Each component of the steam turbine's 4 cylinder, tandem-compound system has a standard design construction called HMN series. To keep high reliability and low cost, the design concept of this standard series is , to

Table 1 Specifications of the steam turbine

Type		4 cylinder, 4 exhaust flow, tandem-compound reheat condensing reaction type
Rated output		600MW
Rated speed		3,000r/min
Steam condition	Main steam pressure (prior to main stop valve)	24.6MPa (251kg/cm² abs)
	Main steam temperature (prior to main stop valve)	538°C
	Reheat steam temperature (prior to reheat stop valve)	566°C
Exhaust pressure		0.00427MPa (728mmHg)
Number of extractions		8
Number of turbine stages	H.P.turbine	16 stages
	I.P.turbine	11 stages x double flows
	L.P.turbine	7 stages x 4 flows
Last stage blade length		1,050 mm (41.5 inch)
Total length		27.9m (from the front bearing pedestal end to the low-pressure turbine rear coupling end)

Fig. 1 Sectional view of Noshiro Thermal Power Plant's steam turbine

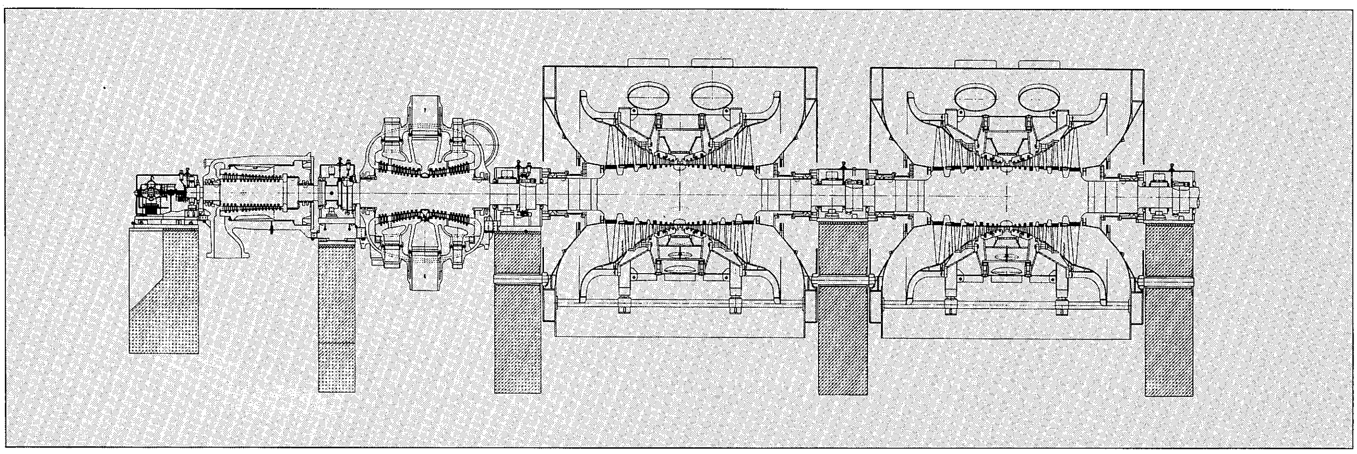
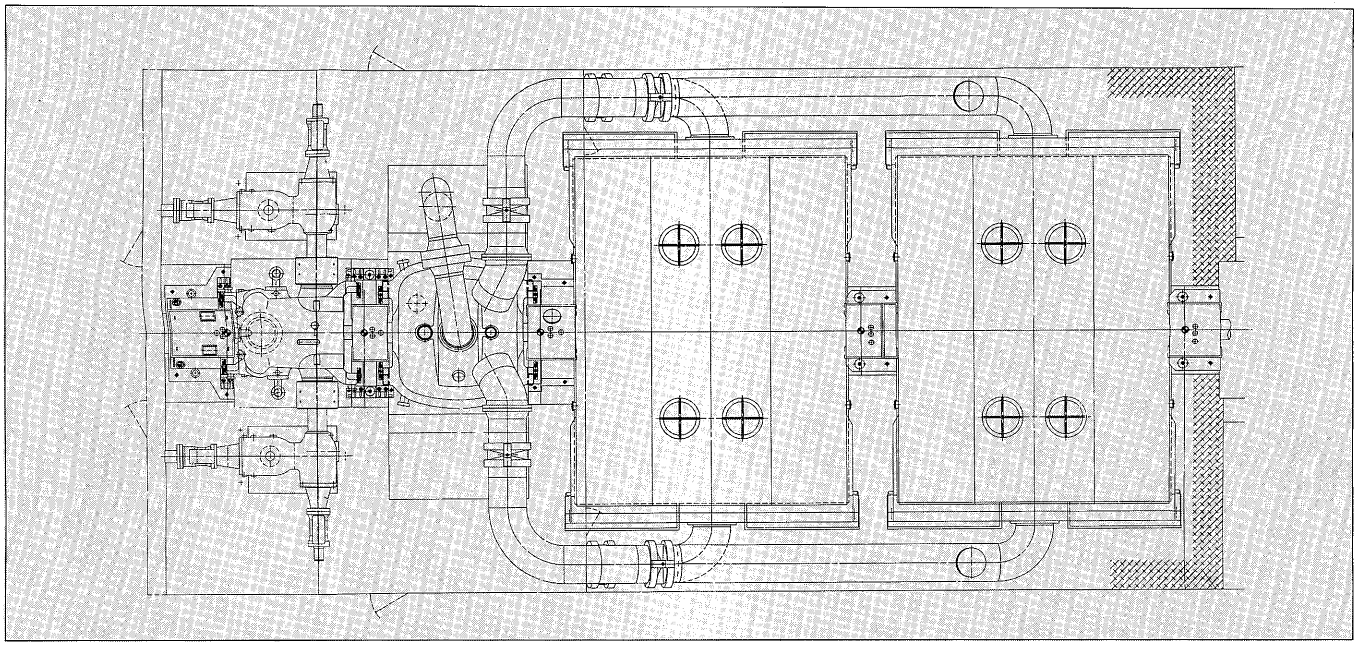


Fig. 2 Plane view of Noshiro Thermal Power Plant's steam turbine



only alter the materials, based on the same basic requirement: up to 30.6MPa (300 ata), 600°C/600°C of the main steam condition. Various combinations of the above components and steam conditions have been successfully proved through numerous operations.

(1) High-pressure turbine

High-pressure turbine has the barrel-shaped, double shell construction. This material is a ferrite-alloy steel with a low ferrite content. The main steam is coaxially (shaft-symmetrically) introduced from two horizontal inlets to operate the turbine with continuous full arc admission and with variable pressure (**Fig. 3**). High-pressure turbine has no control stage.

(2) Intermediate-pressure turbine

This turbine has a double flow and double shell construction. The rotor and inner casing are made of 12%-Cr steel and the reheat steam is coaxially (shaft-symmetrically) introduced. The front exhaust steam flows in reverse to cool the inner casing, joins the rear exhaust steam. The ratio of diameter to span of the casing is large. Thus the casing is rigid enough to avoid thermal deformation.

(3) Low-pressure turbine

This turbine is constructed of welded triple shells, made from steel plate. The steam from the intermediate-

Fig. 3 Sectional view of the high-pressure turbine

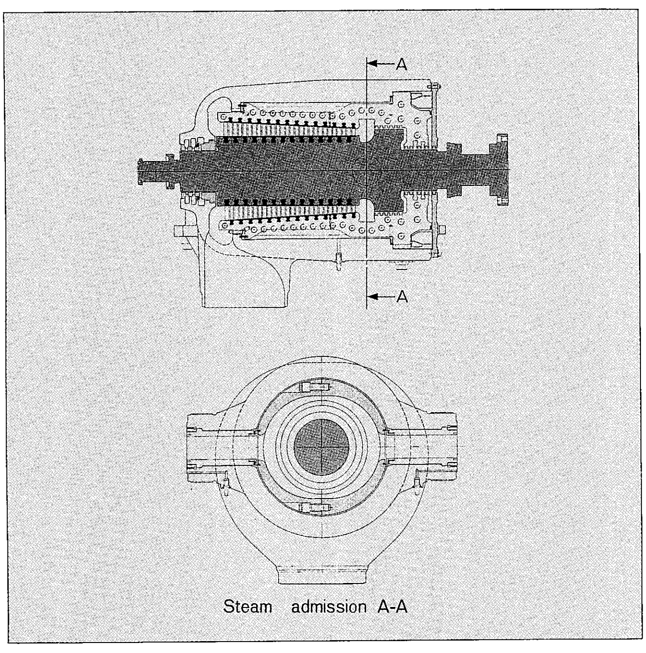
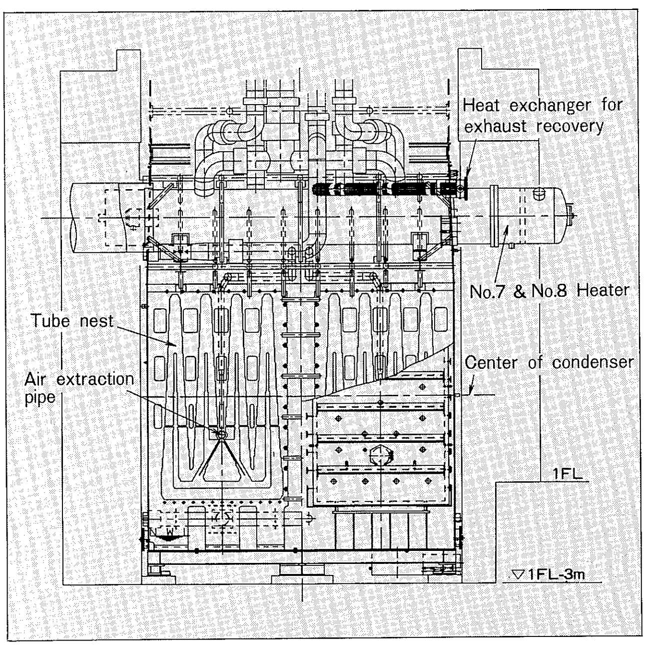


Fig. 4 Sectional view of the condenser



pressure turbine is introduced from the lower part of horizontal flange on both sides of the center part. The gland packing rings at the shaft ends are fixed to the bearing pedestal to flexibly connect to the outer casing through the bellows, so that the deformation of outer casing does not affect the center of the gland packing.

(4) Condenser

This unit is constructed of an axially arranged tube nest coupled with two low-pressure turbine outer casings through rubber bellows. Its function is to condense the exhaust of the main and boiler feed pump turbines and to receive the drainage of the hot water spillover from the boiler deaerator during start-up and stop operations. A new arrangement of the cooling tubes by which the steam flow is totally equalized around the tubes, also helped to minimize the size of the condenser as well as the excavating depth of the condenser floor (**Fig. 4**).

2.2 Reliability improvement

The major features of simplification and standardization to improve reliability of the main turbine are listed below.

Fig. 5 Self-standing, low-pressure moving blade

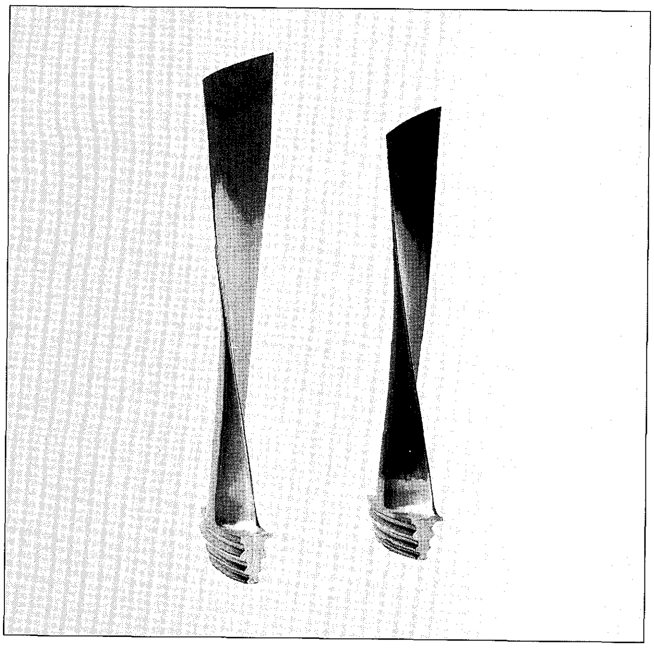
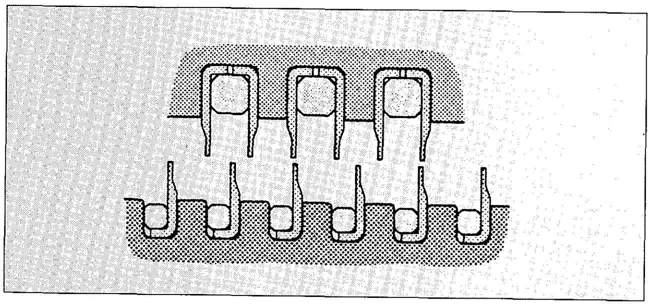


Fig. 6 Double fin sealing



(1) The stationary and moving blades of all stages (from the high-pressure first stage to the low-pressure fore-stage) are integral shroud reaction blades with an improved profile. This type of blade has experienced no breakage.

(2) The low-pressure, moving blades are self standing blades with their leading edge flame-hardened to protect against erosion by water. This type of blade needs no accessories like an anti-vibration lacing wire or a stellite shield, and is thus highly reliable. Since the harmonics of the operating frequency can be tuned for complete prevention of resonance, an wide allowable operating frequency band (–5% to +3% of rated speed) is safely ensured (**Fig. 5**).

(3) The low-pressure, last stage, stationary blades are hollow, and have the slits from which the drains on their surfaces. This construction prevents the following moving blades are sucked from erosion.

(4) All turbine rotors are completely solid without the center bore, reducing the centrifugal stress of the center and prolonging the lifetime against cracking, more than 10 times as compared to those with the center bore.

(5) The intermediate-pressure rotor is made of 12%-Cr steel, which needs no introduction of the external steam to cool the admission nozzle. They employs the journal which is overlay-welded with low alloy metal well-fitted to white metal. They are the solid rotors which require neither a plug-in sleeve nor a coupling boss.

(6) During start-up of turbine the high-and low-pressure rotors which have a rigid shaft design do not pass the critical-speeds during start-up of the turbine. Consequently, the high-pressure rotor is safe against the steam whorl.

(7) The bearings have a single point support structure, with one bearing between each casing. The rotors are flexible between the bearings and hardly affected by misalignment.

(8) Installed independently from the frame of the turbine casings, the bearing pedestals rarely cause misalignment.

(9) A double fin sealing construction, with no restrictions of differential expansion, is used for the gland packings of the high-and intermediate-pressure turbine exhaust parts and both shaft ends of the low-pressure turbine (**Fig. 6**).

(10) Connecting pipes between the intermediate- and low-pressure turbines are in a "side-around" arrangement that allows easy maintenance of the low-pressure turbine.

(11) The turning of the rotor during turbine stop is performed at about 100rpm by the oil-based hydraulic turbine, which is assembled into the coupling flange of the intermediate-pressure turbine rotor. The bearings are safely forced-lubricated, assuring easy handling without a mechanical construction.

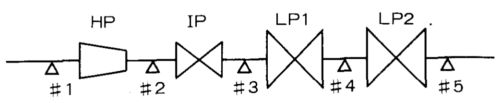
(12) The turbine control equipment is comprised of a digital electric governor. Its operating shaft is driven

Table 2 Major features of the design review in development

Item	Target	Contents
Strength of intermediate pressure first stage blade	To examine the safety for static and dynamic strength	• Numerical analysis, high temperature creep test and fatigue test of the improved blade material • Comparative evaluation on measurements of the existing units
Strength of low-pressure last stage blade	To examine the safety for static and dynamic strength as well as against fluttering and random excitation	• Simulation analysis by FEM, comparative analysis on measurements of the model turbine and the existing units • Examination by rotor vibration test on the actual unit blades (**Fig. 7**, **Fig. 8**)
Turbine efficiency	To minimize the internal losses To introduce no cooling external steam to the inlet of intermediate-pressure turbine	• Simulation of the optimum blade planning • Model test of the boltex cooling method • Analysis by FEM on the temperature distribution of the intermediate-pressure rotor
Strength of the parts exposed to 566°C of the reheat steam	To minimize the alterative bend of the 12%-Cr intermediate-pressure rotor	• Circumferential creep strain velocity test on the center part of the actual rotor material • Analysis and evalution by FEM
	To improve the quality of the overlay welding on the intermediate-pressure rotor	• Welding test of the model rotor
	To examine the safety for the strength of the intermediate-pressure inner casing	• Stress analysis by FEM on steady state and non-steady state
Stability of the combined rotor system	To make the field balance unnecessary and to limit the amplitude of shaft vibration to less than 50μm (p-p) at rated speed	• Simulation analysis on the total rotor system for: 1 Determination of the optimum form and gap of the bearing 2 Determination of the optimum coupling phase of each rotor with regard to the thermal stability test and the single unit balance test
	To examine the safety for the reinforced bearing metal (ϕ560)	• Trial manufacture and test of the ϕ560 reinforced bearing metal by the centrifugal casting method
Reliability of the major steam valves	To examine the safety of the casting valve casing and to improve this quality.	• Casting test of the actual size model (**Fig. 9**) • Stress analysis on steady state and non-steady state
	To examine the safety of the valve against vibratory acceleration of fluid	• Measurements of the vibratory acceleration on the spindle of the existing units • Analysis and evaluation by similar theory
Measures against earthquakes	To examine the strength and safety at maximum acceleration 0.4g	• Dynamic analysis of the response of the turbine generator supports, steam pipings and valves to the earthquake wave • Analysis for the strength of each connection of the turbine
Operating characteristics of the turbine	To omit the operating test at the shop and adjustment during the field test	• To simulate the following items on various start-up conditions by the reliability expectation system on the turbine operation: 1 shaft vibration 2 bearing metal temperature 3 variation of gaps in axial and radial directions and to compare with the restrictive conditions
Completeness on shipment from the shop	To minimize assembly and adjustment work on site, and to shorten installation period and to improve total quality	• Assembly of the condenser with optimum block construction • Package type BFPT • Console type lubricating oil system • Unit type construction of the accessory piping • Unit type construction of the instrument piping around the heater

Table 3 Specifications of the turbine bearing

Item \ Measuring point	#1	#2	#3	#4	#5
Bearing diameter D (mm)	280	380	560	560	560
Bearing length L (mm)	140	290	500	560	450
Bearing load (N)	34,590	172,000	543,200	789,300	394,100
Bearing type	Double wedge bearing		special double wedge bearing		



with fire-resistant fluid of 3.2Mpa pressure.

3. Design Review and Quality Assurance in Development

Reliability of the turbine components and its functional features are well-proven on numerous operations. Before manufacturing, a design review was throughly performed to meake sure of the functions and high quality as Fuji Electric's record-making configuration. We selected the 401 review items which mainly included past major defects. These items were thoroughly analyzed and examined from the viewpoint of performance and reliability.

Major review items by model simulations and finite element method (FEM) are listed in **Table 2.**

As for stabilizing the combined rotor system, double wedge bearings for both sides of the rigid high-pressure

Fig. 7 Rotating vibration test of the 1,050mm blade with 3,000rpm

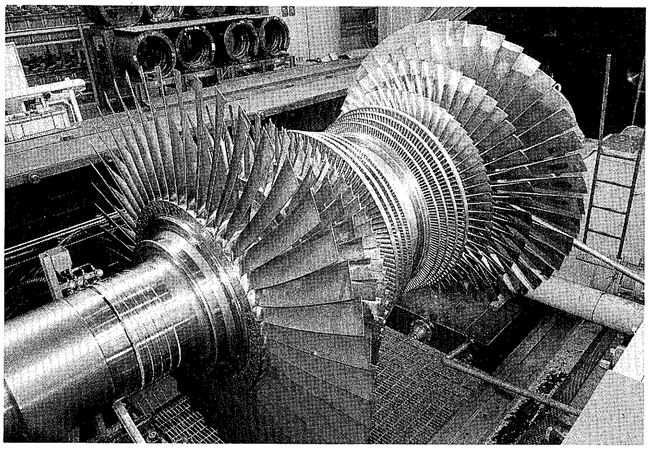


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Fig. 8 Campbell diagram of the 1,050 L.S.B.

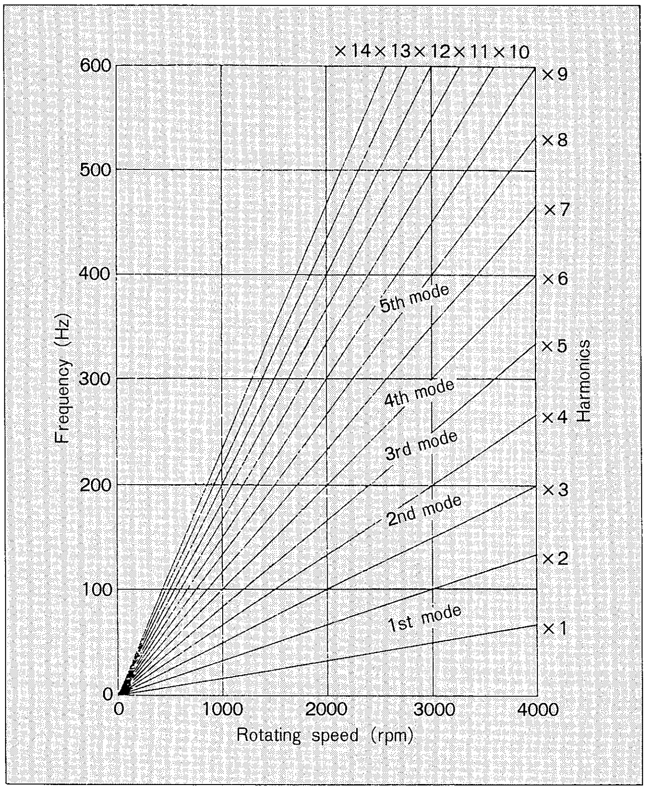


Table 4 Operating record of the turbine

Item	Measuring point / Load	#1	#2	#3	#4	#5
Shaft vibration (μm)p-p	No-load	11	22	11	5	3
	300MW	20	21	19	4	3
	600MW	20	24	15	4	2
Bearing metal (°C) temperature	600MW	67.6	68.7	80.1	91.9	80.0

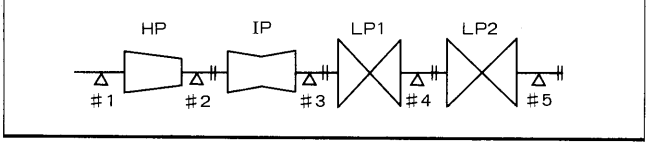
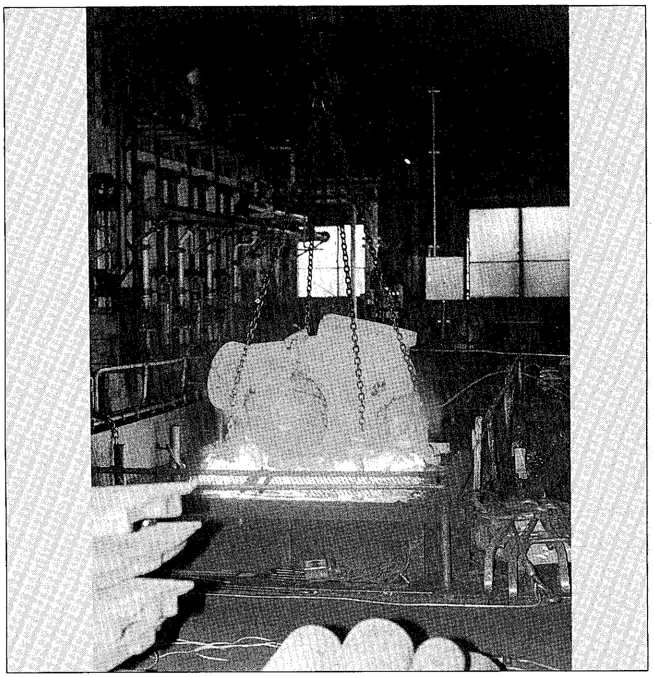


Fig. 9 Hardening of the prototype for the main steam valve casing



rotor and the special double wedge bearings for both sides of the elastic, low-pressure rotor are arranged, optimizing the arrangement of the bearing gaps to keep it coordinated with the generator rotor (**Table 3**).

The influx guide ring has a boltex cooling construction to keep high performance and to prevent the intermediate-pressure rotor from bending (**Fig. 10**).

4. Results of Test Operation

Since the initial steam injection on December 18, 1992, various site tests have been successfully performed. The performance test in the middle of April 1993 has proved quite satisfactory as described below.

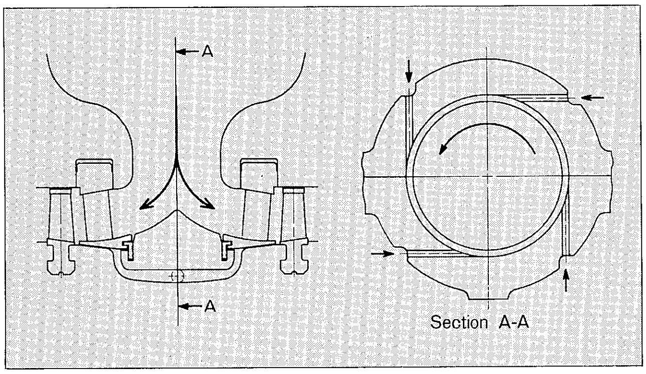
4.1 Shaft vibration and bearing metal temperature

The measurements of shaft vibration and bearing metal temperature are given in **Table 4**. Without field-balancing, the shaft vibrations on all the bearing points and loads were less than 30μm (p-p) and the bearing metal temperatures were not more than 93°C. These results were quite satisfactory.

4.2 Differential expansion of the turbine

The differential expansion of the high-pressure turbine tended to "rotor-long" at the low-speed heat soak operation on the first steam injection. The heat delivery to the outer casing of the high-pressure turbine was less than expected level. This was solved by adjusting the program to trim the openings of the CV and ICV at the low-speed heat soak operation. This gave the same pressure (about 1 Mpa)

Fig. 10 Guide ring of the intermediate-pressure turbine admission nozzle



as a plant with a low-pressure bypass circuit, in which Fuji a plant with a low-pressure bypass circuit, in which Fuji Electric has numerous experiences.

4.3 Turbine efficiency

Measurements of the turbine heat rate on the performance test are given in **Table 5**. Results were quite satisfactory at each load.

Table 5 Results of heat rate on performance test

Test load (coal fired)	MW	600	450	300
Design value	kcal/kWh	1,830	1,850	1,909
Measured value	kcal/kWh	1,812.6	1,836.6	1,902.8
Deviation (relative value)	%	0.957 (good)	0.731 (good)	0.333 (good)

5. Conclusion

The technical features and the results of test operation of Noshiro Thermal Power Plant's Unit 1 Steam Turbine have been shown. During its test operation period, this unit, which holds Fuji Electric's rating record, succesfully passed major target goals, and can be considered a milestone in realizing higher efficiency and reliability.

We sincerely acknowledge advice given by Tohoku Electric Power Co., Inc. during the period from planning to test operation of the unit.