Wakamatsu High-Temperature Turbine Verification Test Project (A Study of a Large-Capacity Coal-Fired USC Plant) *

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This paper reviews the design of the EPDC's Wakamatsu high-temperature turbine and the operating performance of its first-step verification test, STEP I, which features main/reheat steam temperatures of 593°C/593°C. In designing this turbine, a number of new concepts for elevated temperature use were incorporated, and several types of new heat-resistant steels were used in the major parts. Since the initial startup in November 1986, the operating performance of the Wakamatsu high-temperature turbine STEP I has so far demonstrated high reliability, and many valuable data are being obtained, proving the practicality of a large-capacity USC turbine.

Key Words: Steam Turbine, Fossil-Fuel-Fired Power Generation, Ultrasuper-critical, Wakamatsu, High Temperature

1. Introduction

The ultrasupercritical pressure (USC) power-generating system which realizes improved efficiency by raising steam temperature and pressure in a thermal power plant is one of the potential systems by which to utilize abundant coal reserves effectively.

Two basic steam conditions (shown in Table 1) can be given as steam conditions for large-capacity

atmospheric pressure, pulverized coal-fired USC power plants⁽¹⁾. For basic steam condition No.1, all major parts of a turbine can be made of ferritic materials. At 649 °C of basic steam condition No.2, austenitic materials must be used for the rotor and the inner casing. Technology for the manufacture of these parts in the required sizes has been developed^{(2)~(4)}.

Figure 1 shows the relative ratio of improvement in power-generating efficiency due to improvements in steam condition based on 24.1 MPa, 538/538°C supercritical pressure power plants which are prevalent at present. An improvement of about 6% is expected with basic steam condition No.1, and about 8% with basic steam condition No.2.

In order to prove that a 1 000-MW-class USC turbine can be realized and to collect data conducive to designing the turbine, the Wakamatsu High-Temperature Turbine Verification Test project was started. The project was conducted in two stages, STEP I and STEP II.

This paper reports the design of the Wakamatsu high-temperature turbine and the achievements made so far, including the operation results of STEP I.

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Table 1 Steam conditions of large-capacity USC plant and Wakamatsu High-Temperature Turbine Verification Test project

Large Capacity USC Plant					Wakamatsu High-Temperature Turbine Verification Test Project			
Steam Condition	Main Steam Pressure MPa	Main Steam Temperature °C	First Reheat Temperature °C	Second Reheat Temperature °C	STEP	Main Steam Pressure MPa	Main Steam Temperature °C	Reheat Temperature °C
Basic Steam Condition No. 1	31.0	593	593	593	STEP I	10.0	593	593
Basic Steam Condition No. 2	34.5	649	593	593	STEP II	10.0	649	593

Fig. 1 Efficiency vs steam condition

2. Design of Wakamatsu High-Temperature Turbine

2.1 Summary of the project

In this project, the existing Wakamatsu No.2 75-MW turbine is modified into a 50-MW high-temperature turbine, replacing the HP-IP combined element with a new one and using the existing LP element in order to actually generate power. The schedule for the project is shown in Fig. 2.

2.2 Conceptual design of 1000-MW USC turbine

In designing the Wakamatsu high-temperature turbine, a conceptual design of a 1000-MW USC turbine was worked out to determine the required specifications (sizes of major parts, operating temperature and strength of materials at operating temperature). Two types, as shown in Fig. 3, are conceivable

Fig. 2 Schedule of Wakamatsu High-Temperature Turbine Verification Test project

Fig. 3 Composition of 1000-MW USC turbine

for a 1000-MW USC turbine. In type (b), the highest-temperature, highest-pressure part of the VHP element of type (a) is separated as a SP element, and the remainder is combined with the HP element. As the steam leakage loss increases, type (b) is somewhat disadvantageous in its performance as compared with type (a)⁽¹⁾.

For basic steam condition No.1, type (a) can be used if a rotor material with a creep rupture strength of 98 MPa or more at 593 °C and a casing material of 69 MPa or more can be manufactured in large sizes⁽¹⁾. Generally, austenitic materials are difficult to manufacture in large sizes and have the disadvantage that thermal stress and differential expansion are likely to increase during operation. Also considering that the highest pressure rotor is likely to cause unstable vibration, it is reasonable to use type (b).

2.3 Design of Wakamatsu high-temperature turbine

Because the Wakamatsu high-temperature turbine is a modification of an existing turbine and because its output is limited to 50-MW, a main steam pressure of 10.0 MPa was used, as in the existing unit, but efforts were made so that the verification test could be conducted with the temperature in the high-temperature parts and the centrifugal force as close to those of a 1 000-MW-class USC turbine as possible.

In designing the Wakamatsu high-temperature turbine, attention was paid to the following points:

(a) to secure high reliability by employing an appropriate construction and selecting proper materials;

(b) to achieve high turbine efficiency overcoming the restrictions due to modifications of an existing unit and reduction in output,

(c) to enable collection of data useful for the design of 1 000-MW-class USC turbines.

2.3.1 Construction

a. Differential expansion

Differential expansion does not pose a problem in STEP I since 12Cr steel is used as the rotor material. However, in STEP II, in which an austenitic steel with a thermal expansion coefficient about 1.4 times that of low CrMo steel is used as the rotor material, a transient differential expansion as large as about 17 mm is produced in the coupling between the HP-IP rotor and the LP rotor. Since axial clearance is fixed in the existing LP turbine, this differential expansion cannot be accommodated. To solve this problem, a diaphragm coupling is provided between the HP-IP rotor and the LP rotor.

b. High-temperature design of main steam inlet

Austenitic materials are used for the main steam stop valve, steam control valve and main steam inlet pipe in STEP I so that they can withstand a main steam temperature of 649°C in STEP II. High main steam temperature coupled with a large thermal expansion coefficient of the material causes a large thermal expansion of the main steam inlet pipe. To reduce thermal stress due to thermal expansion, a large loop was provided in the main steam inlet pipe

to make the pipe more flexible.

The main steam inlet pipe was inserted into the turbine in a radial direction, as shown in Fig. 4, to eliminate eccentricity with the inner casing and nozzle chamber due to differential expansion. A flexible skirt was provided at the connection to the outer casing to reduce thermal stress, and a dissimilar metal joint with a short piece of incolloy in between was used to reduce thermal stress resulting from the difference in thermal expansion between the austenitic steel and low CrMo steel.

The nozzle chamber was supported by the horizontal joint of the casing to eliminate eccentricity with moving blades. To reduce the thermal stress of the nozzle chamber which receives the highest-temperature, highest-pressure steam, a round section construction of a forged and welded type was employed so that a thin wall design could be made^{(1),(5)}.

c. High-temperature design of moving blades

A triple-pin root-type design was used for control stage moving blades. Manufactured as sets of three by an electric discharge process, the blades are high in toughness and resistance to vibration. For high- and intermediate-pressure stages, integral shroud-type blades were used. These blades are resistant to creep because they do not have tenons for fastening several moving blades into groups with shrouds.

d. Rotor cooling

In STEP I, the control-stage outlet steam flow is reversed and led to high-pressure stages via the outside of the nozzle chamber wall. On the inside of the nozzle chamber wall, a flow separator is provided so that control-stage outlet steam led through pumping holes in the control-stage disc will flow over the rotor surface to cool the rotor at the control stage (see Fig. 14 in 3.4).

In both STEP I and STEP II, a cooling steam path is provided at the blade groove bottom of the IP

Fig. 4 High-temperature design for main steam inlet part

turbine inlet No.1- and No.2 -stage moving blades so that cooling steam will be led from the HP turbine via the IP dummy ring to cool the rotor at the IP inlet (see Fig. 15 in 3.4). The temperature of the cooling steam can be changed by controlling the steam flow bypassing to the HP exhaust via an HP dummy ring leakage steam bypass pipe provided in the middle of the HP dummy ring, as shown in Fig. 5⁽⁶⁾.

2.3.2 Materials Table 2 shows the materials

used for Wakamatsu high-temperature turbine STEP I and the materials which will be used for STEP II. These materials were developed to meet the required specifications of materials, which were determined in accordance with a conceptional design of a 1 000-MW USC turbine which was made with a balance between design and materials.

a. Materials for STEP I

A high temperature of 593°C is close to the limit

Table 2 Material selection for Wakamatsu high-temperature turbine

Parts	Turbine	Wakamatsu High-Temperature Turbine	
		Conventional Turbine 538/566°C	STEP I 593/593°C STEP II 649/593°C
Main Steam Inlet Pipe		2 1/4 CrMo Steel	316H (Forged & Bored)
MSV, GV		2 1/4 CrMo Forged Steel	316H (Forged) *Stem W545, Bushing A286 *Nitriding
Reheat Steam Inlet Pipe		2 1/4 CrMo Steel	oHCM9M (Forged & Bored)
RSV, ICV		RSV: 2 1/4 CrMo Cast Steel ICV: 2 1/4 CrMo Forged Steel	oHCM9M (Forged) *Stem Refractaloy 26, Bushing Type 422 *Nitriding
Inlet Sleeve		2 1/4 CrMo Forged Steel	316H (Forged Pipe)/Alloy 800H/2 1/4 CrMo (Pipe) /2 1/4 CrMo Outer Casing
Rotor		CrMoV Steel	o12 Cr Steel (TMK 1) oModified A286
HP-IP Inner Casing		2 1/4 CrMo Cast Steel	o12 Cr Cast Steel 316H (Forged)
Outer Casing		1 1/4 CrMo Cast Steel	2 1/4 CrMo Cast Steel
Nozzle Chamber		2 1/4 CrMo Cast Steel	oHCM9M (Forged) *Welded Type 316H (Forged) *Welded Type
HP Blade Ring		1 1/4 CrMo Cast Steel	o12 Cr Cast Steel 316H (Forged)
HP Dummy Ring			
IP #1 Blade Ring		1 1/4 CrMo Cast Steel	12 Cr Cast Steel
IP Dummy Ring			
IP #2 Blade Ring		1 1/4 CrMo Cast Steel	1 1/4 CrMo Cast Steel
LP Dummy Ring		1 1/4 CrMo Cast Steel	2 1/4 CrMo Cast Steel
Moving Blade		Type 422, AISI 403	Refractaloy 26, Type 422, AISI 403 Refractaloy 26
Stationary Blade		AISI 403	oHCM9M, AISI 403 316H
Bolt (High Temperature Only)		Type 422	Refractaloy 26 for Ferritic Component W545 for Austenitic Component

oDeveloped by MHI

Fig. 5 Temperature change method for IP inlet cooling steam

temperature for ferritic steel, but a USC turbine composed mainly of ferritic steel is greatly advantageous in operation and economy as compared with a turbine made of austenitic steel, so 9 or 12Cr steels were developed and used for STEP I.

(1) New 12Cr Rotor (TMK1). Steam temperatures of 566°C and 582°C are the limits for the conventional HP/IP rotor materials CrMoV steel and 12CrMoVNbN steel, respectively.⁽⁶⁾ Jointly with Professor Fujita of Tokyo University and Kobe Steel, Ltd., Mitsubishi Heavy Industries, Ltd. developed a new 12Cr rotor TMK1 which can withstand 593°C. Figure 6 shows a rotor made of TMK1 for STEP I, which is equivalent to the size of the VHP rotor of the 1 000-MW USC turbine. The TMK1 features (a) low C, (b) 1.5% Mo, (c) adoption of the ESR process, and has a creep rupture strength well above the value required for 1 000 MW-USC turbines, as shown in Fig. 7^{(7)~(10)}.

Moreover, the TMK2 with $W\%/Mo\% > 3$,

Fig. 6 Wakamatsu high-temperature turbine rotor for STEP I (TMK1)

Fig. 7 Creep rupture master curve of TMK1 rotor and working stresses

$Mo\% + 1/2W\% = 1 \sim 1.5\%$ was developed using W. Since this material has a higher creep rupture strength than TMK1 on the high-temperature, long-time side, it has become possible to manufacture a turbine of mainly ferritic materials for a USC plant with a steam temperature of 621°C in between steam temperatures of STEP I and STEP II⁽¹¹⁾.

(2) New 12Cr Cast Steel (MJC12). Since a steam temperature of 566°C is the limit for 2 1/4CrMo steel, Mitsubishi Heavy Industries, Ltd. developed a new 12Cr cast steel MJC12 jointly with the Japan Casting & Forging Corporation. This material has a creep rupture strength which permits it to be used at a steam temperature about 40~50°C higher than 2 1/4 CrMo cast steel.⁽²⁾

(3) HCM9M. 9Cr2Mo HCM9M is a highly weldable, high-strength steel which Mitsubishi Heavy Industries, Ltd. developed jointly with Sumitomo Metal Industries, Ltd. as a material for boiler tubes and main steam pipes, and was used for nozzle chamber, reheat stop valves and intercept valve chambers.

b. Materials for STEP II

To withstand 649°C, it is imperative to use austenitic steel. Austenitic steel was difficult to manufacture at large sizes, so manufacturing technology was also developed.

(1) Modified A286 Rotor. A problem incurred in the manufacture of a large-size A286 rotor is a segregation called a "freckle". In cooperation with Kobe Steel, Ltd., Mitsubishi Heavy Industries, Ltd. developed a modified A286 rotor material which does not produce freckles. Figure 8 shows a modified A286 rotor being machined for STEP II. It is the world's largest austenitic rotor and has a size equivalent to the VHP rotor of the 1 000-MW USC turbine of type (a) shown in Fig. 3. It is intended to clarify the problem of thermal stress and differential expansion quantitatively from the operation of STEP II in conjunction with a study of unstable vibration, in order to

Fig. 8 Wakamatsu high-temperature turbine rotor for STEP II (modified A286)

Table 3 Chemical composition of modified A286 rotor

Component	C	Si	Mn	P	S	Ni	Cr	Mo	V	Al	Ti	B
Spec.	0.010 ~0.030	0.35 max.	0.30 ~0.80	0.020 max.	0.010 max.	25.00 ~27.00	14.00 ~16.00	1.00 ~1.50	0.10 ~0.50	0.35 max.	1.15 ~1.60	0.0020 ~0.0100
Actual	0.014	0.12	0.58	0.012	0.0002	25.89	14.91	1.24	0.25	0.16	1.39	0.0038

determine the optimum configuration of a 1 000-MW USC turbine for basic steam condition No.2. Ti in the modified A286 rotor material is reduced, as shown in Table 3, by lowering the high-temperature creep rupture strength to the required minimum; thus, coupled with an improvement in manufacturing techniques, the formation of freckles is suppressed^{(3),(4)}.

(2) SUSF316H Forged Steel. This material was used for high-temperature parts such as nozzle chamber and HP inner casing. Its specifications are the same as those of the JIS material. In manufacturing the nozzle chamber, an electron beam welding process which produces little distortion was used.

c. Others

If the reheat steam temperature is 593°C, LP inlet temperature exceeds 400°C, which is in the range of tempering embrittlement for the conventional 3 1/2Ni rotor material. However, development of a super-clean rotor material featuring (a) low Mn and Si, and (b) low impurities such as P, Sn, Sb and S has made it possible to manufacture LP rotors which are not subject to embrittlement at 400°C or higher^{(1),(12)}.

In the Wakamatsu high-temperature turbine, the LP element of the existing 75 MW turbine is used at 50 MW, so LP inlet temperature is sufficiently low and the existing LP rotor does not cause any problems.

2.3.3 Measurement of operation data To obtain data conducive to designing 1 000-MW-class USC turbines, many instruments are fitted to the Wakamatsu high-temperature turbine, as shown in Fig. 9. In addition to the metal temperature of the rotor center bore at 8 points, the rotor surface temperature at the control stage and the IP inlet is detected by thermocouples and measured by a telemeter. Moreover, strain gauges are fitted to measure stress in inner casing bolts, and so forth^{(6),(13)}. To collect and process data, a data logger is installed. Some of the data are analyzed on line and are displayed on a CRT.

3. Operation Results of STEP I

3.1 Operation History

November 30, 1986	Initial steam admission
December 3, 1986	Initial synchronization
December 22, 1986	Initial operation at 50 MW
March 20, 1987	Passed inspection before commercial operation

Table 4 Results of turbine performance tests

		Load	50MW	37.5MW	25MW
Item	Unit				
	Target				
Turbine Heat Rate	kJ/kwh	8,566	8,662	9,085	
	Actual	kJ/kWh	8,415	8,583	8,792
Comparison (Relative Value)		%	+1.8	+0.9	+3.2

Fig. 9 Temperature and stress measuring points for verification test

Operating hours of turbine	11 075 hours	to June 16, 1989
Power-generating hours	10 746 hours	
Number of turbine starts and stops	177 times	

3.2 Performance

A performance test of the high-temperature turbine showed that the turbine heat rate was lower than the design value, as shown in Table 4⁽¹⁴⁾. In a performance test in which the main steam/reheat steam temperature was actually changed, it was confirmed that turbine heat rate improves as steam temperature rises, as shown in Table 5.

3.3 Reliability

Examples of the CRT display indicating shaft vibration, bearing temperature, differential expansion, thrust and casing top and bottom temperature differences are given in Figs. 10~13.

All of these values representing the reliability of the turbine are stable and well below the allowable ranges.

Table 5 Results of steam temperature changing tests

Item	Steam Condition Unit	Rated	Reduced (Case-1)	Reduced (Case-2)
Main/Reheat Steam Temp.	°C	593/593	566/566	538/538
Turbine Heat Rate Comparison (Relative Value)	%	Base	-1.8	-3.3

Fig. 10 Rotor vibration • Bearing temperature (CRT display)

Allowable limit Rotor vibration 12.5/100 mm
Bearing oil temperature 77°C
Bearing metal temperature 107°C

Fig. 11 Differential expansion • Casing expansion (CRT display)

Allowable limit Differential expansion +9.1 mm, -1.0 mm

3.4 Rotor cooling effect

Examples of the CRT display indicating metal temperature and steam temperature near the control stage are given in Fig. 14. The high-temperature leakage steam from the nozzle is isolated from the rotor surface by the flow separator installed on the inside of the nozzle chamber wall. Control stage outlet steam led by pumping holes in the control stage disc flows on the inner circumference side of the flow separator. Thus, the rotor temperature was lowered.

Examples of the CRT display indicating metal temperature and steam temperature at the IP inlet are given in Fig. 15. The cooling steam temperature at the IP inlet can be changed by changing the opening of the

Fig. 12 Thrust • Rotor position • Thrust bearing temperature (CRT display)

Allowable limit Thrust bearing metal temp. 99°C
Thrust bearing oil temp. 77°C
Rotor position ± 0.9 mm
Thrust intensity 1.0 MPa

Fig. 13 Casing top and bottom temperature difference (CRT display)

Allowable limit $\pm 42^\circ\text{C}$

Fig. 14 Control-stage metal and steam temperatures (CRT display)

Fig. 15 IP inlet metal and steam temperatures (CRT display)

HP dummy ring leakage steam bypass valve, as mentioned in 2.3.1 d. This is shown in Fig. 16. When the HP dummy ring leakage steam bypass valve is fully closed, the ratio of HP dummy ring leakage steam in IP cooling steam increases and the cooling steam temperature rises. On the other hand, when the HP dummy ring leakage steam bypass valve is opened, the cooling steam temperature falls. With the change in IP cooling steam temperature, the rotor metal temperature changes, and so does the temperature of cooling steam flowing along the internal circumference of the No.2- and No.3-stage stationary blades. These results of temperature measurement verified the effectiveness of the IP inlet cooling method. Furthermore, the heat transfer coefficient in the rotor surface was verified by a comparison between the rotor temperature distribution analysis

Fig. 16 Results of IP cooling steam temperature changing test

by the finite-element method and the temperature measured. Figure 17 shows the calculation results of rotor temperature distribution at the IP inlet.

3.5 Inner casing temperature distribution

Figure 18 shows the calculation results of HP inner casing temperature distribution. It was found that heat transfer coefficients on the inside and outside must be increased more than originally estimated in order for the calculation results to agree with the temperature actually measured. The reason for this seems to be that the actual local flow velocity is about twice the average velocity used to obtain the value originally estimated.

Figure 19 shows the calculation results of the IP inner casing temperature distribution. As in the case of the HP inner casing, it was found that heat transfer coefficients must be increased more than originally estimated in order for the calculation results to agree with the temperature actually measured. Nevertheless, the effectiveness of the thermal shield installed on the inside of the inner casing is remarkable, and the metal temperature is 535°C or lower against the steam temperature of 593°C on the inside, indicating that the metal temperature is lower by 20°C or more than when the thermal shield is not provided⁽¹³⁾.

3.6 Bolt stress

Figure 20 shows the IP inner casing bolt stresses measured, which agree with the values obtained by calculation from the bolt and casing temperature measured. It is seen that the tension of the bolts is reduced by a difference of thermal expansion as temperature rises, because bolts made of refractory 26

Fig. 17 Calculation results of IP rotor temperature distribution

Fig. 18 Calculation results of HP inner casing temperature distribution

Fig. 19 Calculation results of IP inner casing temperature distribution

Fig. 20 Results of IP inner casing bolt stress measurement

with a large thermal expansion coefficient are used for the IP inner casing. Anticipating this reduction in tension at high temperatures, bolts are tightened to a higher tension when the turbine is assembled. In STEP II, an improved 12Cr bolt material derived from the development of 12Cr rotors for 593°C and 621°C will be used instead of refractory 26.

3.7 Development of large-capacity USC turbine

The following results obtained from the STEP I verification test indicate that it is possible to realize 1 000-MW-class USC turbines for basic steam condition No.1.

(1) The high-temperature turbine materials developed here can be used for large-capacity USC turbines.

(2) The effectiveness of rotor cooling has been verified and heat transfer rates in various parts can be obtained accurately, so the calculation accuracy of temperature and stress of large-capacity USC turbines has been improved and a highly reliable design can be made.

On the other hand, the problems in increasing turbine capacity are

(a) as steam pressure rises, the wall thickness of the pressure vessels becomes larger and thermal stress increases;

(b) as steam pressure rises and output becomes larger, the highest-pressure rotor becomes more susceptible to unstable vibration.

Problem (a) can be solved by (2) of the above results, and (b) can be solved separately as a vibra-

tion problem.

4. Conclusions

(1) Aiming at the realization of atmospheric-pressure, pulverized-coal-fired large-capacity USC power plants, two basic steam conditions were set up, and the Wakamatsu High-Temperature Turbine Verification Test project was started.

(2) New materials and a new design developed to meet the specifications determined by a conceptual design of a 1 000-MW USC turbine were incorporated into the Wakamatsu high-temperature turbine simulating a VHP element of a 1 000-MW USC turbine.

(3) Wakamatsu high-temperature turbine STEP I (main steam/reheat steam temperature 593/593°C) has been in successful operation since the end of 1986. Its performance is higher than the design value, and the values of turbine supervisory items indicating reliability are stable at levels lower than the allowable ranges. Measurements of temperature and stress in various parts of the turbine have clarified the effectiveness of rotor cooling and heat transfer coefficients, and other valuable data useful for designing large-capacity USC turbines have been obtained, increasing the possibility of realizing 1 000-MW-class USC turbines.

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