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VERIFICATION FULL LOAD TEST OF 900 T/H (1,980,000 LB/H) LARGE MECHANICAL DRIVE STEAM TURBINE FOR ASU UP TO 200 MW (268,000 HP) BASED ON THE PRINCIPLE OF HIGH SCALE MODEL SIMILARITY

by
Osamu Isumi
Turbine Designing Section Manager

Takashi Niiyama
Design Engineer
Mitsubishi Heavy Industries, Ltd.
Hiroshima, Japan

Kazushi Mori
Chief Research Engineer
Mitsubishi Heavy Industries, Ltd.
Hyogo, Japan
and
Satoshi Hata
Group Manager
Mitsubishi Heavy Industries, Ltd.
Hiroshima, Japan

Osamu Isumi is the Turbine Designing Section Manager, Mitsubishi Heavy Industries, Ltd., in Hiroshima Japan. He has 18 years of experience in designing steam and gas turbines.

Mr. Isumi has a B.S. degree (Mechanical Engineering) from the Miyazaki University.

Takashi Niiyama is a Design Engineer in the Turbine Designing Section, Mitsubishi Heavy Industries, Ltd., in Hiroshima Japan. He has six years of experience in designing steam turbines.

Mr. Niiyama has B.S and M.S degrees (Mechanical Engineering) from Kyushu University.

Kazushi Mori is a Chief Research Engineer in the Takasago Research & Development Center, Mitsubishi Heavy Industries, Ltd., in Hyogo, Japan. He is a specialist in blade vibration with 14 years of experience in this field.

Mr. Mori has B.S. and M.S. degrees (Mechanical Engineering) from Kyoto University.

Satoshi Hata is a Group Manager within the Turbo Machinery Engineering Department, Mitsubishi Heavy Industries, Ltd., in Hiroshima, Japan. He has 26 years of experience in R&D for nuclear uranium centrifuges, turbomolecular pumps, and heavy-duty gas and steam turbines.

Mr. Hata has B.S., M.S. and Ph.D. degrees (Mechanical Engineering) from Kyushu Institute of Technology

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ABSTRACT

In 2009, the authors designed a compressor drive steam turbine for air separation units (ASU) applications up to 200 MW (268,000 hp), using large 500 mm (19.7 inch) blade rows, which have a low-pressure (LP) flow capacity of up to 900t/h (1,980,000 lb/h). These 500 mm (19.7 inch) blade rows were tested under actual load conditions between August and September 2008 at the authors' shop, based on the principle of high scale model similarity. In conclusion, excellent results were obtained and the reliability of the blade rows was verified.

INTRODUCTION

Considering market prospects for air separation business, the authors can expect air separation units (ASU) capacities to increase substantially. For the largest ASU, the air compressor steam turbine driver power will reach 200 MW (268,000 hp) and the turbine LP section flow will increase to as much as 900 t/h (1,980,000 lb/h). On the other hand, end users require a full load test to verify the integrity of rotating machinery. To get around the resulting limitation of shop testing facilities, the authors carried out a verification full load test on a large 900 t/h (1,980,000 lb/h) mechanical drive steam turbine for ASU, based on the principle of high scale model similarity. This was done in variable speed range between August and September 2008 at the authors' shop. Successful results were obtained and the reliability and performance of the blade rows were verified.

In this paper, the authors introduce the detailed rules of scale model similarity, the successful test results and state-of-the-art techniques for measuring actual vibration stress. This includes a comparison with a vibration response analysis based on three-dimensional (3D) finite element analysis (FEA) and an unsteady computational fluid dynamics (CFD) and rotating excitation test of an actual size 500mm (19.7 inch) double integral shroud blade (ISB) blade. The authors have completed the verification program for the mechanical drive steam turbine LP blades and have created a robust design of untuned blades for large ASU.

SPECIFICATION OF STEAM TURBINE FOR ASU

Figure 1 shows a typical outline of an ASU compressor and turbine module. In this steam turbine, the LP blade rows, (four stages total with a 500 mm (19.7 inch) L-0 stage height), will be applied. Furthermore, the authors have standardized three frame types for the 500 mm (19.7 inch) LP blade rows optimizing the base diameter, which can be selected based on required operating conditions (mainly LP steam flow). Table 1 shows typical dimensions of the three frame types of blade rows and their applicable operation range.

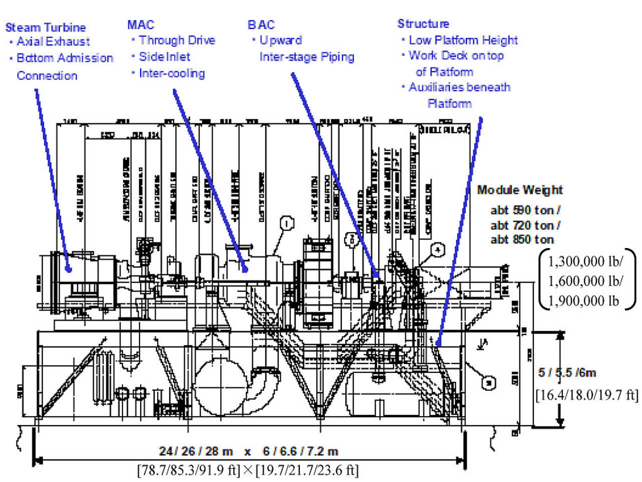

Figure 1. Outline of ASU Compressor Unit.

Table 1. Three Frame Type of LP Blade Rows.

	-	Frame10	Frame12	Frame14
Speed Range	rpm	3,600-4,725	3,200-4,200	2,880-3,780
LP flow	t/h (lb/h)	200-600 (441,000-1,320,000)	550-750 (1,210,000-1,650,000)	700-900 (1,540,000-1,980,000)
Exh. Press.	MPaA (inHgA)	0.02-0.06 (5.91-17.72)	0.02-0.06 (5.91-17.72)	0.02-0.06 (5.91-17.72)
Exh. mouth	mm (inch)	φ2,800 (φ110.2)	φ3,000 (φ118.1)	φ3,400 (φ133.9)
L-0 Base Dia	mm(inch)	1,016 (40.0)	1,318 (51.9)	1,500 (59.1)
L-0 Height	mm(inch)	500 (19.7)		
L-1 Base Dia	mm(inch)	1,054 (41.5)	1,334 (52.5)	1,538 (60.6)
L-1 Height	mm(inch)	340 (13.4)		
L-2 Base Dia	mm(inch)	1,080 (42.5)	1,308 (51.5)	1,564 (61.6)
L-2 Height	mm(inch)	240 (9.4)		
L-3 Base Dia	mm(inch)	1,064 (41.9)	1,270 (50.0)	1,548 (60.9)
L-3 Height	mm(inch)	170 (6.7)		

For reference, Figure 2 shows a cross section of the steam turbine for an actual planned ASU project, in which Frame 10 500 mm (19.7 inch) blade rows were applied. Table 2 shows the main specifications.

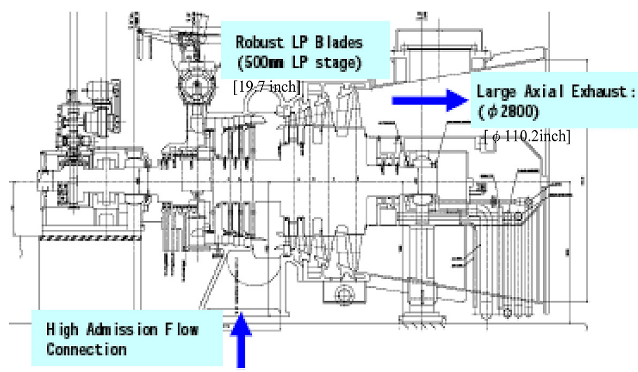

Figure 2. Cross Section of Steam Turbine (Frame10 Project-N).

Table 2. Main Specifications of Steam Turbine for ASU Compressor Driver (Actual Planned Project).

Actual Planned Project, 6train case		
Turbine Model no.	-	10MXL-8
LP Blades rows		Frame 10
Turbine rated	kW (hp)	83,200 (112,000)
Speed	rpm	3600 (const)
Inlet flow	t/h (lb/h)	Max.350 (772,000)
Inlet press.	MPaG (psig)	Nor.5.1 (740)
Inlet temp.	℃ (℉)	Nor. 365 (689)
Adm. flow	t/h (lb/h)	Max. 400 (882,000)
Adm. Press	MPaG (psig)	Nor. 1.4 (203)
Adm. temp	℃ (℉)	Nor. 255 (491)
Exh. flow	t/h (lb/h)	Max. 550 (1,210,000)
Exh. Press	MPaA (inHgA)	Nor. 0.03 (8.86)
Exh. temp	℃ (℉)	Nor. 69 (156.2)

WELL-PROVEN COMPONENTS

Table 3 shows the status of product development for each component of the compressor drive steam turbine. This clearly shows the authors have experience with all the main components except the LP blades.

Table 3. Well-Proven Components of Compressor Drive Turbine for ASU.

Components		Status (Notes)	Reference
Casing	Inlet Section	○	High pressure Experience 13.7MPa×540℃×750t/h (1,987 psig×1004℉×1,650,000 lb/h) Low pressure Experience 0.68MPa×332℃×138t/h (98.6 psig×629.6℉×304,000 lb/h)
Casing	Exhaust Section	○	Experience of axial exhaust φ2,800mm (φ110.2 inch) : 3 turbines φ3,000mm (φ118.1 inch) : 3 turbines φ3,400mm (φ133.9 inch) : 3 turbines
Nozzles		○	Experience (Proven Profile)
Blades	HP blades	○	Experience 13.7MPa×540℃×750t/h (1,987 psig×1004℉×1,650,000 lb/h)
Blades	LP blades	△ ↓ ○	Conducted rotating excitation test with actual size model Conducted full load test with 1/4 scale model
Rotor (Single flow)		○	Experience of Bearing Span is 5,200 mm (204.7 inch) Experience of Max tip diameter is φ2,540mm (100.0 inch)
Bearing	Inlet side	○	Experience, for 3,600 rpm, Journal φ360mm (14.2 inch) : 2turbines Thrust J-22.5 : 4turbines
Bearing	Exhaust side	○	Experience, for 3,600 rpm, Journal φ320mm (12.6 inch): 3turbines
Rotor Dynamics		○	Experience of Bearing Span is 5,200 mm (204.7 inch) API requirement can be satisfied

(Notes) ○: Well proven △: Need to be verified

Therefore, the authors conducted an excitation test on both LP blade rows, using an actual scale model, and an actual load test with a scale model to verify the LP blade reliability. With these tests, the authors could complete the verification program for the ASU turbine. (In this paper, the authors mainly introduce an actual load test with a scale model.)

PRINCIPLE OF HIGH SCALE MODEL SIMILARITY

The rotating blades of steam turbines are essential elements in the conversion of fluid force to mechanical force; therefore high operational reliability is a critical requirement. During the design/development stage, vibratory characteristics and actual blade strength were checked and verified with a rotating excitation laboratory test. However, an actual load test in a test turbine is the best method of verifying blade reliability. Due to the limited quantity of steam flow in the test facility, the actual load test is usually carried out using a scale model turbine instead of a full scale model. Scale model testing can be achieved by applying mechanical and aerodynamic laws of similarity. Table 4 shows a summary of the principles of similarity.

Table 4. Summary of Principle of Similarity.

Item	1/n Model	Note
Rotating Speed (Ω)	n	Adjusted by tip speed
Acceleration	n	$r\Omega^2$ $(1/n \times n^2 = n)$
Mass (m)	$1/n^3$	Volume
Centrifugal Force (F)	$1/n^2$	$mr\Omega^2$ $(1/n^3 \times n = 1/n^2)$
Centrifugal Stress (σ)	1/1	$mr\Omega^2/A$ $(1/n^2 \times n^2 = 1/1)$
Twist Angle	1/1	$\rho\ \Omega^2 l^2/G$ $(n^2 \times 1/n^2 = 1/1)$
Elongation (Δl)	1/n	$lmr\Omega^2/AE$ $(1/n \times 1/n^2 \times n^2 = 1/n)$
Displacement	1/n	
Length (l)	1/n	
Area (A)	$1/n^2$	
Velocity (U)	1/1	Adjusted by mach number
Weight Flow (G_o)	$1/n^2$	AU $(1/n^2 \times 1/1 = 1/n^2)$
Heat Drop (H)	1/1	Adjusted by mach number
Turbine Output (L)	$1/n^2$	G_oH $(1/n^2 \times 1/1 = 1/n^2)$
Turbine Torque (T)	$1/n^3$	L/Ω $(1/n^2 \times 1/n = 1/n^3)$
Resonant Stress (δ_v)	1/1	
Blade Frequency (ω)	n	$1/l^2\ \sqrt{(1/A)}$ $(n^2 \times \sqrt{(1/n^2)} = n)$
Reduced Frequency (ω_r)	1/1	$\omega \times C/U$ $(n \times 1/n = 1/1)$
Reynolds Number (Re)	1/n	$l \times U$ $(1/n \times 1/1 = 1/n)$
Ω_R/Ω_S	1/1	Resonant RPM Ω_R/ Blade Natural Frequency RPM Ω_S
Tolerance	1/n	
Damping (δ)	1/1	
Stimulus (s)	1/1	

r: Radius, G: Shear Modulus, ρ : Mass Density,
E: Elastic Modulus, C: Blade Cord width, I: Moment of Inertia

VERIFICATION OF LP BLADES FRAME AND SCALE RATIO FOR THE TEST TURBINE

During the initial planning of this study, the authors checked the most severe condition of the LP blade rows under planned operating conditions for each LP blade row type. Considering the simplest way to study the most severe conditions, the authors compared the steam mass flow for each radial blade flow path area because the blade pitch of each frame of LP blades rows is almost the same and this flow/area ratio is related to the loading per blade. For reference, the following planned projects were used as a case study sample.

- Frame 10, Project-N, max LP flow 580 t/h (1,280,000 lb/h)
- Frame 12, Project-A, max LP flow 750 t/h (1,650,000 lb/h)
- Frame 14, Project-M, max LP flow 900 t/h (1,980,000 lb/h)

The authors then confirmed that Frame 14 was the most severe case, so the goal for the test turbine was to verify the blade loading of Frame 14 case (max LP flow 900 t/h [1,980,000 lb/h]).

The authors then studied the scale ratio of the test turbine. At first, the authors checked the shop facility limitations, namely the shop condensate system, boiler, and driven equipment (test compressor). The authors then decided on the largest possible high scale model size for the test turbine, which would achieve the required blade loading within the limitations of the shop facilities and manufacturing constraints.

As a result, the authors decided on a one-fourth scale test turbine with Frame 14 LP blade rows. Figure 3 shows photos of the actual/scaled L-0 blade and Figure 4 shows photos of the test rotor. Table 5 shows a comparison of the operating conditions between actual and scaled turbines.

Figure 3. Actual and Scaled L-0 Blades.

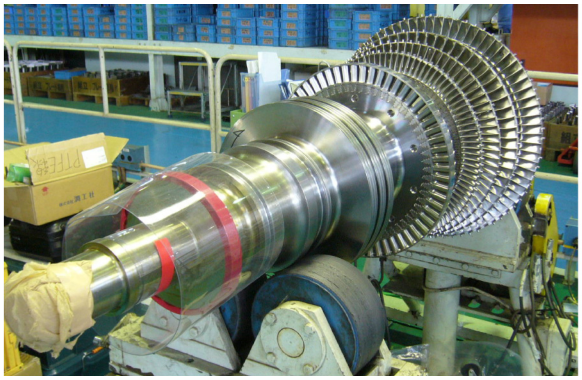

Figure 4. Turbine Rotor for Verification Test (One-Fourth Scale).

Table 5. Comparison of Operating Conditions Between Actual and Scaled Turbines.

Case		-	Frame14 Actual scale	Frame14 1/4 Scale
Flow		t/h(lb/h)	900 (1,980,000)	56.3 (124,000)
Speed		rpm	3600	14400
Stage Power	L-3	kW(hp)	23,200 (31,100)	1,400 (1,900)
	L-2	kW(hp)	26,500 (35,600)	1,700 (2,200)
	L-1	kW(hp)	22,200 (29,700)	1,400 (1,900)
	L-0	kW(hp)	36,100 (48,400)	2,300 (3,000)
Blade Height	L-3	mm(inch)	170 (6.7)	42.5 (1.67)
	L-2	mm(inch)	240 (9.4)	60 (2.36)
	L-1	mm(inch)	340 (13.4)	85 (3.35)
	L-0	mm(inch)	500 (19.7)	125 (4.92)
L-0 Base Dia.		mm(inch)	1,500 (59.1)	375 (14.8)

VERIFICATION TEST OF LP BLADE ROWS

Tapping Test

Before the actual load test, a tapping test of the LP blade rows at the free-free condition was carried out. Table 6 shows the tapping test results for the L-0 stage (scaled blade).

Table 6. Scaled Blade Tapping Test Results.

Mode		Deviation from calculated Natural Frequency					
		Calc.	No1	No2	No3	No4	No5
1st	%	100.0	102.7	100.8	102.7	102.7	100.8
2nd	%	100.0	101.4	100.7	102.9	101.4	101.4
3rd	%	100.0	99.5	98.3	100.6	100.6	98.9

The authors conducted a 3-D model study of the LP blades (actual/scaled model); Figure 5 shows the natural frequency mode shape of the scaled L-0 stage blades. As a result, measured values for each mode shows a ±3 percent variation, compared with the calculated value.

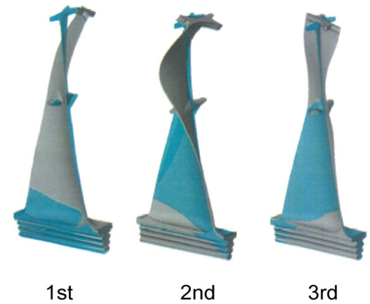

Figure 5. Natural Frequency of Scaled Blades (L-0 Stage, Free-Free).

Test Turbine

Figure 6 shows the test facility setup schematic used for the scaled 500 mm (19.7 inch) LP blade row verification. Table 7 shows the major specifications of the test turbine. Figure 7 shows details of the manufactured test turbine and compressor.

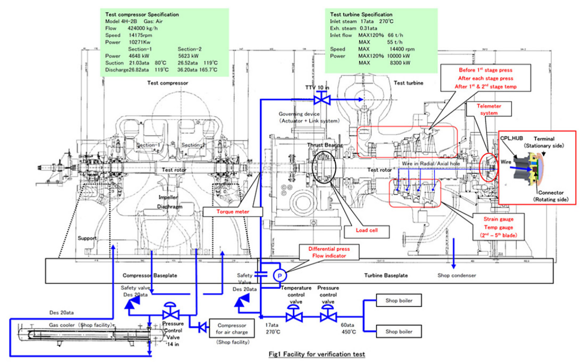

Figure 6. Test Setup Schematic for LP Blade Rows Verification.

Table 7. Test Turbine Specifications.

Speed	rpm	11,520-14,400
Power	kW (hp)	7,800 (100%) (10,500)
Inlet steam Flow	t/h (lb/h)	56.3 (100%) (124,000)
Inlet steam press	MPaG (psig)	1.67 (242.2)
Inlet steam press	℃ (℉)	270 (518)
Exhaust press	MPaA (inHgA)	Nor 0.03 (8.86)

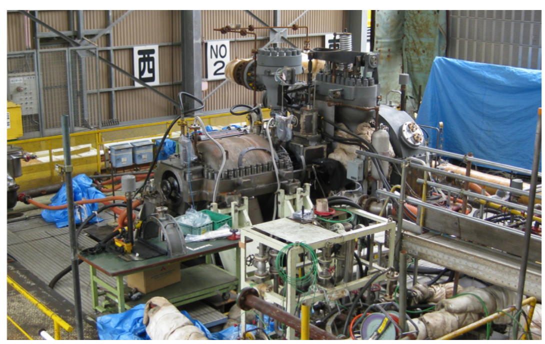

Figure 7. Test Turbine and Compressor for Verification of 500 MM (19.7 Inch) LP Blades Rows.

The test turbine was planned as a five-stage straight condensing type, with one control stage and four LP stages. Its size was selected considering a steam flow, which could be verified with blade loading at the most severe case. Adjustment and control of the inlet steam conditions were made through the use of a pressure control valve and temperature decreasing device on the inlet steam line. Additionally, the exhaust pressure was adjusted, and a vent valve was applied on the condenser. A two-stage centrifugal type impeller was selected for the test compressor and air was used as the operating fluid in a closed loop.

Special Measurement

During the test, the authors took some special measurements for verification of the LP blade rows, including excited blade frequencies, vibration stresses, and flow path performance, in addition to normal measurements ("speed," "steam flow, pressure, temperature on piping," and "shaft vibration," etc.).

In particular, to verify the LP blade rows vibration response, the authors measured the blade vibration stress using "strain gauges" and "thermocouples" installed on the blades with a "telemetry system." Figure 8 shows the special wiring for the blade rows verification. "Strain gauges" were installed on the blade profile, tip, or mean edge. Six strain gauges were installed on each stage (L-0 to L-3 stage). The authors utilized a state-of-the-art, induced electromotive type telemetry system (no battery/self generation type) and measured vibration response through this system. Figure 9 shows the test rotor with the telemetry system installed. For reference, the cross section of the telemetry system is shown. During measurement, the authors were able to check the vibration response of the blades through this system. Figure 10 shows a photo taken during measurement of the vibration response.

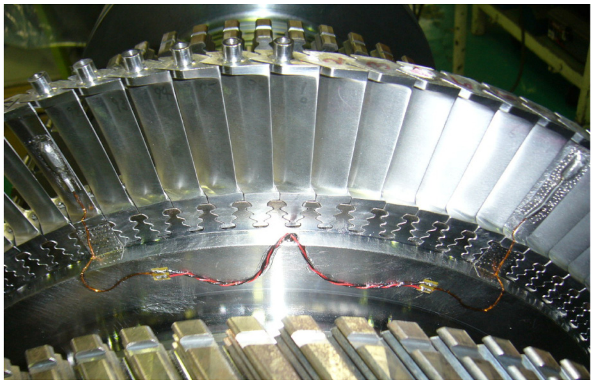

Figure 8. Special Wiring for Blade Rows Verification ("Strain Gauges" and "Thermocouples").

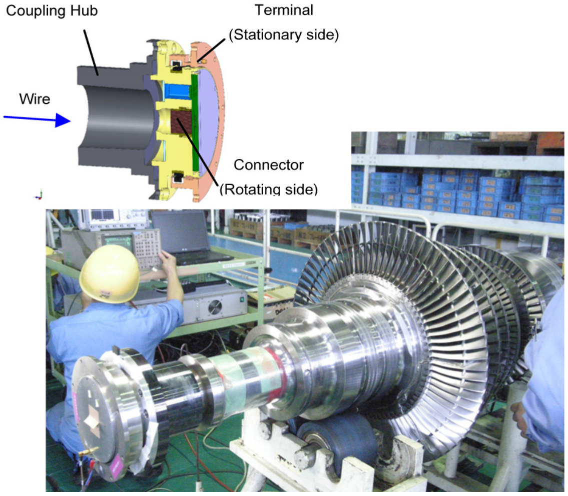

Figure 9. Rotor Installed Telemetry System.

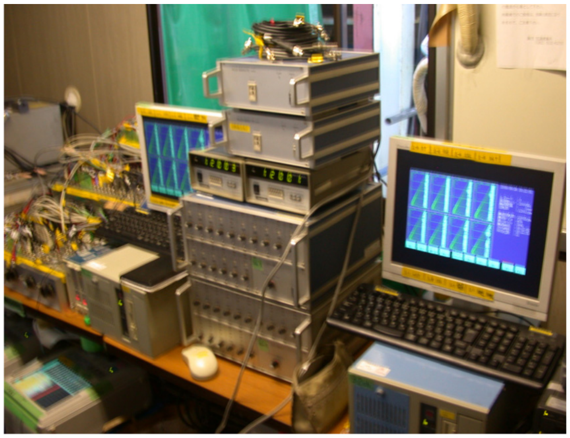

Figure 10. Measurement Facilities in Verification Test .

For verification of LP blade row performance, the authors measured the pressure and temperature of each stage in the turbine casing and the turbine power through the coupling torque meter.

Test Conditions

Figure 11 shows the applicable operation range of the type 14 LP blade rows in the one-fourth scale test. The red hatched line shows the applicable range of these LP blade rows and the pink triangles and blue circles show the operating (verified) conditions during the verification test. (At the pink points, the vibration response and performance were verified. At the blue points, only the vibration response was verified, because the L-3 stage condition was wet.) As one can see, the measured points were covered at almost all the applicable operating ranges.

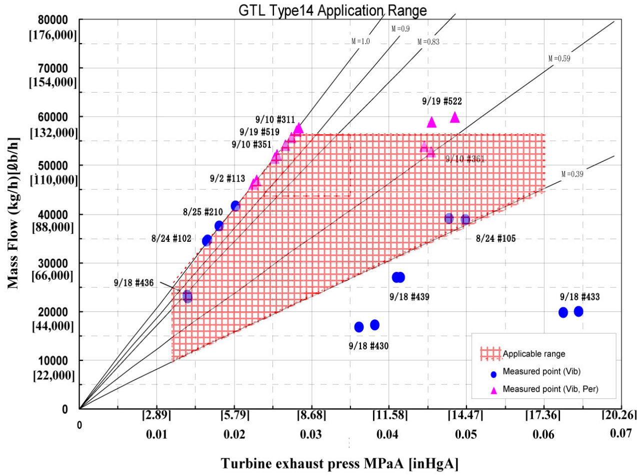

Figure 11. Operating Conditions.

Regarding the method of measurement, at first, turbine power and exhaust pressure at max continuous revolution (MCR, 14,400 rpm) were adjusted until a stable operating condition was confirmed. The speed was then decreased to minimum governor speed (MGS, 11,520 rpm), and then increased to MCR again until stable operating conditions were confirmed. During the stable condition at MCR, performance measurement was conducted, and as the speed changed, blade vibration response was measured (two times for each measurement).

The authors carried out these measurements for seven days, with 50 hours of operation in total. Also 1.5 hours of 110 percent load test at harmonic resonant speed was conducted to verify blade stress cycles in excess of 1×10^7.

VERIFICATION TEST RESULTS OF LP BLADE ROWS

Blade Vibration Response

Figures 12 and 13 show the measured Campbell diagrams of the L-0, L-1 stage blades. In these diagrams, the size of the circles represent the vibration stress levels. The authors evaluated the vibration response stress level of the harmonic resonance to low vibration mode (below 20 harmonic), nozzle wake resonance to high vibration mode (60 harmonic for L-0 stage, 74 harmonic for L-1 stage), and random vibration level from low exhaust vacuum condition (L-0 stage), under a wide range of operating conditions.

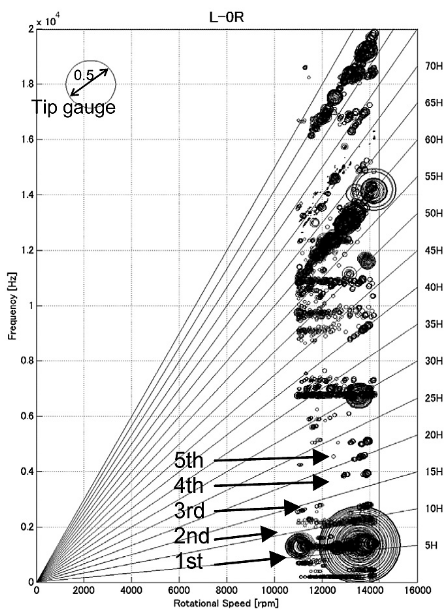

Figure 12. Campbell Diagram (L-0 Stage).

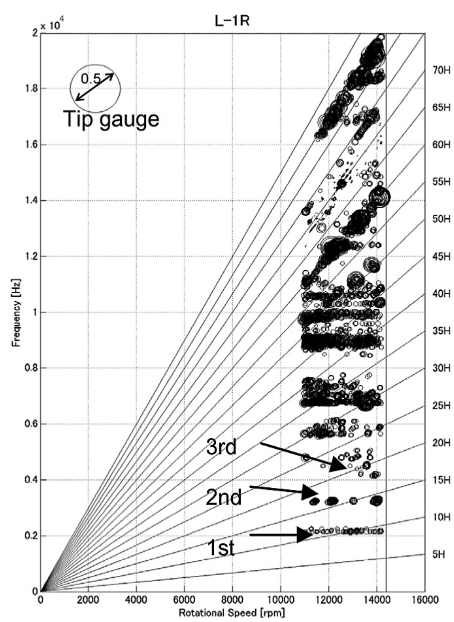

Figure 13. Campbell Diagram (L-1 Stage).

The authors then checked and obtained vibration response of harmonic resonance to low vibration mode (below 20 harmonic) test results. Figures 14 and 15 show the Campbell diagram for comparison of measured results and the studied model. The authors were able to confirm the natural frequency of the low vibration mode of the scaled blades, which agreed with the study results. (Other LP stage results also agreed.)

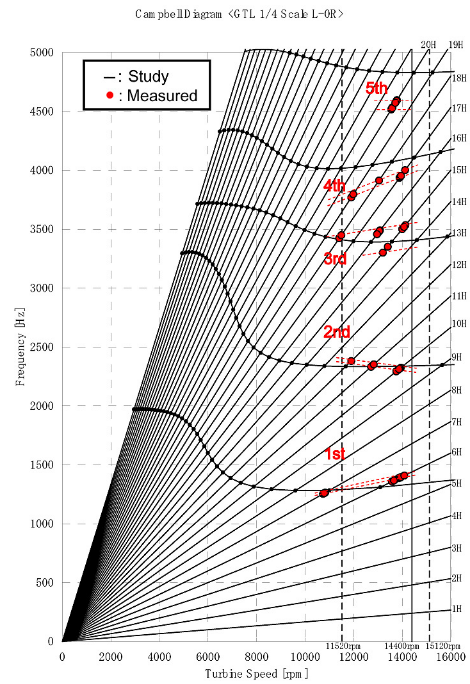

Figure 14. Campbell Diagram (L-0 Stage).

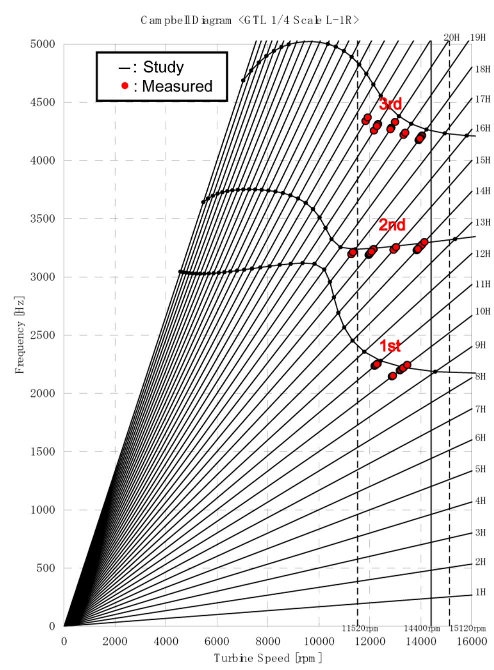

Figure 15. Campbell Diagram (L-1 Stage).

Regarding the vibration response stress evaluation, the authors tabulated the largest resonant stress measured during the verification test for each vibration mode. A safety margin for the L-0 stage for each vibration mode was calculated as shown in Table 8. A safety margin was calculated (based on allowable stress) at the weakest point stress, which was considered as an actual stress measured point (tip or mean edge) at each vibration mode. As a result, the authors evaluated the safety margin of all the LP blades.

Table 8. Vibration Stress Level per Each Mode for L0 Stage.

Mode	Resonant harmonic	Safety margin (allow. stress/ stress @ weakest point)
1st	6H	6.9
2nd	10H	38.9
3rd	15H	34.4
4th	17H	79.6
5th	20H	29.4
14th	60H	44.3
15th		33.6
16th		15.8
17th		32.9
18th		10.8

As reported in Table 8, the first mode safety margin for the L-0 stage was the lowest, but still had a suitable factor of safety. For reference, the authors show the vibration mode analysis result (first mode, most severe case) in Figure 16 and the stress contour diagram in Figure 17. The authors calculated a safety margin using these analysis results.

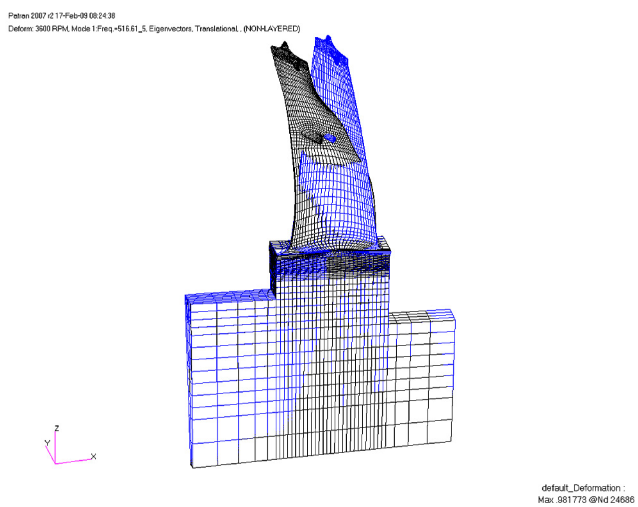

Figure 16. Vibration Mode (First Mode, L-0 Stage).

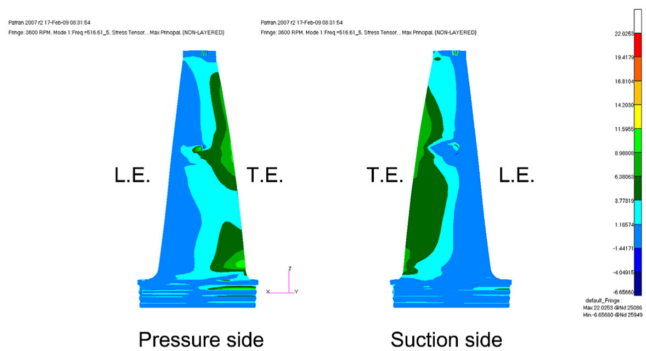

Figure 17. Stress Contour Diagram (L-0 Stage) First Mode.

Figure 18 shows the Goodman diagram for the L-0 stage. The authors completed the stress level evaluation by plotting the result on a Goodman diagram. As a result, the authors were able to verify that the LP blade rows have enough strength margin against harmonic resonance to low vibration mode, nozzle wake resonance to high vibration mode, and random vibration with low exhaust vacuum condition (L-0 stage) through all the applicable operation ranges.

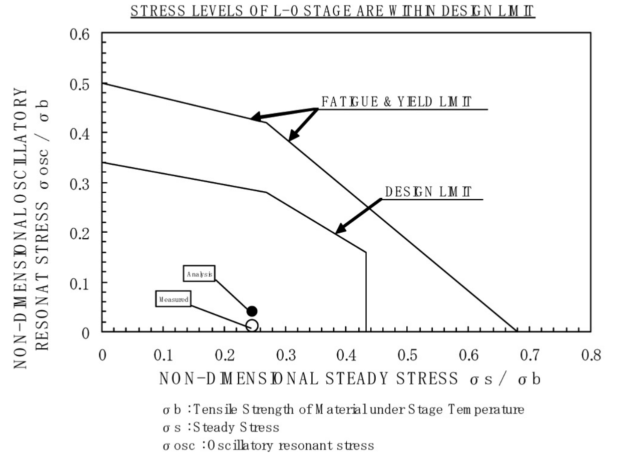

Figure 18. Goodman Diagram (L-0 Stage).

Performance

Figures 19 and 20 show measured stage efficiencies of L-2 to L-0 stage and L-3 stage. (Because L-3 stage outlet was in a dry condition the evaluation was separated.) The turbine performance was almost equal to the studied results with 3D CFD and the authors confirmed that the LP blade rows had a high level of thermal efficiency.

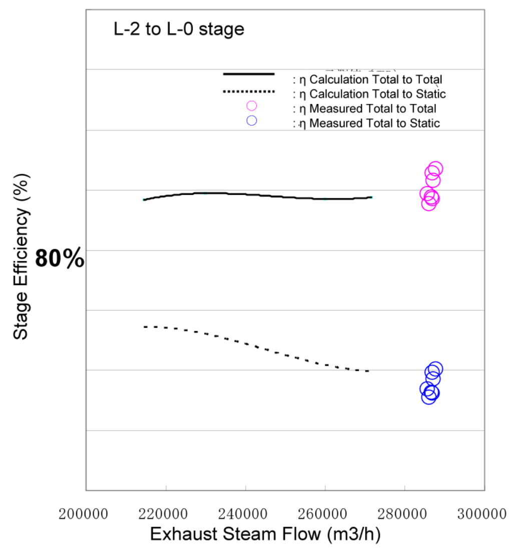

Figure 19. Turbine Efficiency of L-2 to L-0 Stage.

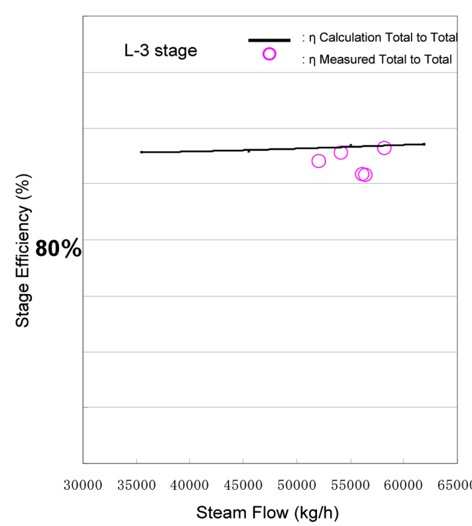

Figure 20. Turbine Efficiency of L-3 Stage.

CONCLUSION

An actual load test using a scale model turbine based on the principle of high scale model similarity was completed successfully in the authors' shop, and the authors were able to verify that 500 mm (19.7 inch) blade rows have a high level of reliability and thermal efficiency.

The authors have completed a verification program for ASU drive steam turbines up to 200 MW (268,000 hp).

The authors are confident that utilizing this turbine can offer great benefits to clients in the ASU market.

ACKNOWLEDGEMENT

The authors gratefully wish to acknowledge the following individuals for their contribution and technical assistance in analyzing and reviewing the results and for their expert advice and leadership in practical application and testing: M. Saga, N. Nagai, Y. Kaneko, and T. Miyawaki, of Mitsubishi Heavy Industries, Ltd.

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