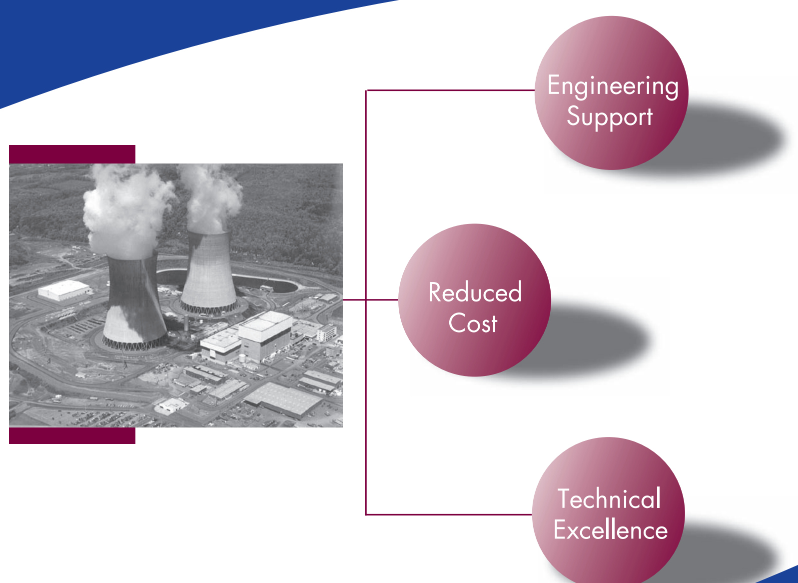
2012 TECHNICAL REPORT

Plant Engineering: Performance Diagnostic Test Program for the Nuclear Turbine Cycle at Korea Hydro & Nuclear Power Company



Plant Engineering: Performance Diagnostic Test Program for the Nuclear Turbine Cycle at Korea Hydro & Nuclear Power Company

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Abstract

Currently, many power generating companies are challenged to reduce operating costs, and at the same time, the cost of unit unavailability can be significant in today's power markets.

In the past decade, management of nuclear power plants, including Korea Hydro & Nuclear Power (KHNP), has been focused on reducing forced outage rates and nuclear-safety-related issues, with less attention paid to thermal performance. But recently, KHNP has been strongly challenged to increase unit thermal performance, as fossil power plants are forced to prepare for the impact of future fuel price increases and carbon taxes and cope with strengthened global environmental regulations.

This mission can be accomplished only through effective performance monitoring and maintenance activities, and in this regard, KHNP has started to perform periodic (a six-year cycle) diagnostic testing for turbine cycle performance at all of its nuclear units.

The purpose of this paper is to introduce KHNP's diagnostic testing methodology for nuclear turbine cycle performance.

Keywords
Full-scale test
Nuclear power plant
Thermal performance diagnostic test
Turbine cycle

Executive Summary

The performance diagnostic test program has been conducted at Korea Hydro & Nuclear Power's (KHNP's) nuclear power plants to determine the overall performance of the turbine cycle. This test program can be used in combination with the contractual acceptance test of new units to achieve benchmark performance parameters of overall turbine cycle and individual turbine cycle components in new and clean conditions.

In this report, Section 1, Introduction, discusses the newly considered diagnostic test program methodology for nuclear turbine cycle performance and the verification of its accuracy.

Section 2, Test Methodology, discusses modifying several test restrictions to conduct the full-scale test for nuclear turbine cycle performance. The thermal performance modeling is generated from recalculating the test heat balance at test conditions using Group 2 corrections. Verification that the diagnostic test at KHNP is sufficient to conduct in lieu of a full-scale test for nuclear turbine cycle performance is done by comparing the test result from the alternative test method with the modeling result.

In Section 3, Test Results and Achievement, performance gains are presented showing that over the past four years; the thermal performance diagnostic testing resulted in an increase of approximately 19.3 MW.

In Section 4, Conclusions and Lessons Learned, the test result from heat balance modeling and the result from the American Society of Mechanical Engineers Performance Test Code 6 Alternative Test Method differed by a maximum of about 0.06%. It is possible to provide accurate information about the turbine cycle and estimate the benefits or loss that result from any performance change.

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Section 1: Introduction

Many nuclear power plants presently evaluate turbine cycle performance by conducting the American Society of Mechanical Engineers (ASME) Performance Test Code (PTC) 6 test using an alternative test methodology. This test program provides limited information about the plant performance; that is, it provides only overall performance parameters such as corrected generator output and heat rate.

The ASME PTC 6 full-scale test method requires extensive thermal cycle measurement and heat balance calculations and provides detailed information on the turbine cycle and its components. However, this method has been very costly and physically impractical for nuclear power plants to use because in many cases new test connections for tracer injection and sampling and/or installation of temporary test piping sections for heater drain flow measurement are required to be installed prior to testing.

Korea Hydro & Nuclear Power (KHNP) has been developing a test program that precisely evaluates the performance level of the nuclear turbine cycle and its individual cycle components. This program refers to the guidelines of the ASME PTC 6 full-scale test method, but does not require the addition of test connections or temporary pipe sections to facilitate accurate measurement of process parameters. Instead, it employs less intrusive methods of data acquisition through the use of calibrated ultrasonic flow meters and application of well-established principles of thermodynamics. The new test program methodology has been applied to KHNP's nuclear power plant units since 2008.

In addition to the benchmark performance parameters of the overall turbine cycle and its individual components, the test program also provides the turbine cycle thermal performance modeling based on the actual unit performance parameters.

This report is composed of the following:

- Overall test program objectives
- Test methodology
- Thermal performance modeling from the test results
- Test results and achievement
- Lessons learned from the test program
- Test program summary and conclusions

Section 2: Test Methodology

Full-Scale Test and Alternative Test in ASME PTC 6

The ASME PTC 6 code contains the most commonly used technical guidelines to evaluate the performance of steam turbines and their thermal cycles. This code presents full-scale test and alternative test procedures. The full-scale test requires extensive thermal cycle measurements and heat balance calculations that provide detailed information on the individual sections of the steam turbine. The alternative test relies on fewer measurements and makes greater use of correction curves for cycle adjustment and heater performance with resultant cost savings over the full-scale test.

Test measurement for the full-scale test procedure allows performance evaluation of individual turbine cycle components, such as turbine individual sections, feedwater pump turbines, moisture separator/reheaters (MSRs), and feedwater heaters. Accurate performance testing of the condenser is also possible with additional temperature measurement of the condenser circulation water. The test results also provide the input data, that is, performance parameters of individual turbine cycle components, for the plant thermal performance modeling.

Constraints for Conducting the Full-Scale Test

In most nuclear power plants, the alternative test is conducted for steam turbine testing, as the nuclear steam turbines are operated in the wet steam region, which is the biggest constraint for conducting the full scale test. ASME PTC 6 code presents several methods for conducting the full-scale test in nuclear power plants, such as the tracer technique or heater drain flow and heat balance, but these methods are still cost and labor intensive and impractical to apply to existing power plants because, in many cases, new test provisions for tracer injection and sampling or temporary test pipe sections for heater drain flow measurement are required.

Under these technical restrictions, operators in nuclear power plants experience difficulties in monitoring and managing performance of individual turbine cycle components and their impact on the electric power output. As a result, the operators cannot cope effectively with plant anomalies related to the kW output performance.

Modifications to Conduct the Full-Scale Test

KHNP's diagnostic test program for nuclear turbine cycle performance is basically following the technical guidelines in the ASME PTC 6 full-scale test method with the following minimum deviations and modifications:

- Use calibrated ultrasonic flow meters to measure the MSR drain flows and high-pressure turbine exhaust steam condition to be determined using energy balance around the MSR.
- Use Keuffel and Esser (K&E) ship curve number 864-31 (dry region) and 50-in. (127 cm) radius curve (wet region) to estimate the extraction stage steam enthalpy as shown in Figure 2-1.
- Use the design moisture removal effectiveness of the low-pressure turbine moisture removal stage to estimate the moisture blowdown from the steam path.
- Assume a constant enthalpy process at the extraction line to the feedwater heaters and reheaters.
- Extend the test boundary from the steam turbine to the overall turbine cycle eliminating Group 1 correction.

The test results provide extensive information about the steam turbine and its cycle components, but the test is easier to conduct and more practical compared to the ASME PTC 6 full-scale test with minimal sacrifice of the test uncertainty. This test program can be used in combination with the contractual acceptance testing of new units to achieve benchmark performance parameters of overall turbine cycle and individual turbine cycle components in new and clean conditions. It can be periodically conducted on existing units to trace performance deterioration of individual turbine cycle components and their impact on electrical power output.

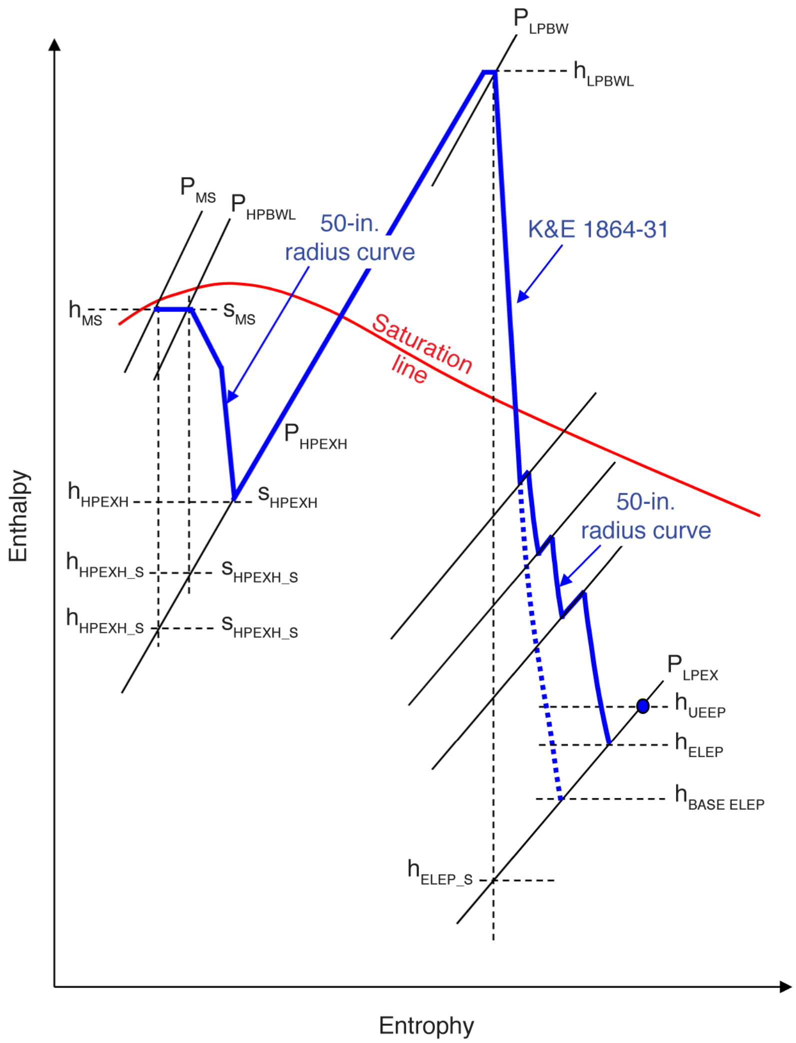

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Figure 2-1
Typical High-Pressure/Low-Pressure Turbine Steam Expansion Line

Thermal Performance Modeling from the Test Results

Figure 2-2 depicts the process for obtaining the thermal performance modeling by conducting this diagnostic test.

The performance parameters of individual turbine cycle components determined from the test cycle heat balance calculation are used as input data for the plant performance modeling.

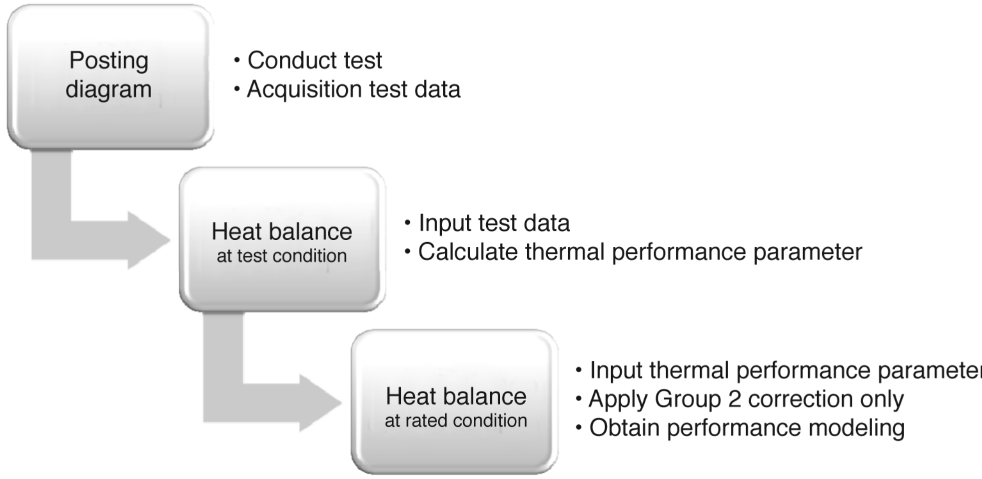

Figure 2-2
Process of How to Get the Thermal Performance Modeling

Modeling the plant thermal performance, the expected output when turbine cycle external variables (throttle pressure and moisture, low-pressure turbine exhaust pressure, generator power factor, and steam generator thermal power) are operated at rated conditions is predicted accurately by heat balance calculations when the following performance parameters resulting from the test cycle heat balance calculation are known:

- Turbine expansion line efficiency, which is determined for each turbine section or stage group by the:
  - Inlet volume flow
  - Steam conditions (a measure of moisture losses)
  - Pressure ratio
- Flow functions at each turbine bowl and extraction stage
- Cycle steam line pressure drop losses (includes turbine valves and moisture separator reheater)
- Extraction steam line pressure drop losses
- Moisture separator effectiveness
- Reheater thermal temperature difference
- Feedwater heater thermal temperature difference and drain cooler approach
- Feedwater pressure drop losses
- Feedwater pump turbine efficiency
- Feedwater pump efficiency or enthalpy rise
- Packing and valve stem leakage flows

Appendix B is provided as an example of plant performance modeling, which was conducted recently for Yong Gwang Nuclear Power Plant Unit #4 in 2011.

Model Verification

In order to have confidence in the results from the modeling, test results from the corrected kilowatt output calculation using performance correction curves should be cross-checked with the predicted kilowatt output.

Table 2-1 shows a sample comparison between results from the correction curves and the plant performance modeling. It was reported that the difference was a maximum of approximately 0.06%.

Table 2-1
Comparison of Each Test Result for 1,000-MW Korean Nuclear Power Plant's Performance Diagnostic Tests Since 2008

Test Results Corrected Kilowatt Output		From Correction Curves (a)	From Performance Modeling (b)	Difference	
				(a)-(b) kW	[(a)-(b)]/(a)
YGN #5 (Sep. 2008)	TR01	1,043,511	1,043,551	40	0.004%
	TR02	1,043,522	1,043,524	2	0.0002%
YGN #6 (Dec. 2008)	TR03	1,045,424	1,044,817	-607	0.06%
	TR04	1,045,855	1,045,294	-561	0.05%
UCN #4 (Jun. 2009)	TR01	1,048,146	1,047,882	264	0.03%
	TR02	1,047,832	1,047,729	103	0.01%
UCN #3 (Dec. 2009)	TR01	1,040,770	1,041,244	474	0.05%
	TR02	1,041,051	1,041,515	464	0.04%
UCN #6 (Apr. 2010)	TR01	1,042,423	1,042,863	439	0.04%
	TR02	1,042,160	1,042,435	275	0.03%
UCN #5 (Nov. 2010)	TR01	1,043,282	1,043,485	203	0.02%
	TR02	1,043,039	1,043,217	119	0.02%
YGN #3 (Jun. 2010)	TR01	1,048,359	1,047,843	-516	-0.05%
YGN #4 (Apr. 2011)	TR01	1,053,009	1,053,124	115	0.01%

Notes:

YGN is the Yong Gwang Nuclear Power Plant.

UCN is the Ulchin Nuclear Power Plant.

This plant performance modeling, achieved from the test cycle heat balance calculation, allows the plant performance engineer to more accurately simulate changes in kilowatt output and operating parameters when cycle modification and/or unit uprating is performed.

KHNP is now developing an on-line performance monitoring system that makes use of the test results from the field performance diagnostic test program. The modeling achieved from the field test also provides real-time expected kilowatt output and reference values to reconcile the station raw data when using the on-line performance monitoring system as a calculation engine. Using the reconciled values in lieu of the raw data will drastically reduce the overall uncertainty of the on-line performance calculation, which includes the reactor thermal power

Figure 2-3 illustrates briefly the test methodology process.

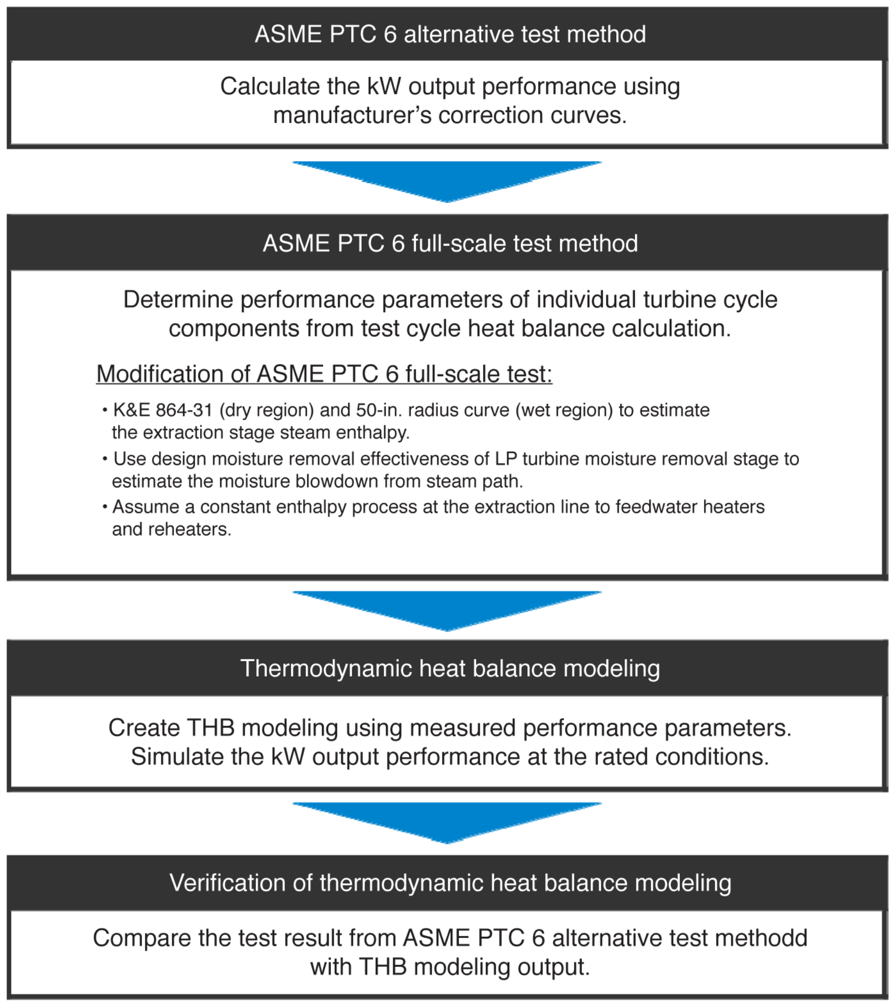

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Figure 2-3
Process of the Thermal Performance Diagnostic Methodology

Section 3: Test Results and Achievement

Testing and Computation of Test Results

All test runs are conducted near 100% rated thermal output and as close as possible to the rated steam conditions. Steam conditions during the above time periods should be stable and should not deviate from the permissible fluctuations of variables stated in Table 3.1 of ASME PTC 6-2004.

The test data recorded by the data acquisition system are stored on disk during each test point. Immediately following a test point, the data are printed, along with averaged values converted to engineering units, and corrections are made for water legs, barometric pressure, and instrumentation calibration. All reduced data are averaged for the appropriate test time period. Redundant readings are averaged. The average value of the test data for each test run is posted on the instrumentation diagram. All calculations are performed using International Formulation Committee of the 6th International Conference on the Properties of Steam, The 1967 IFC Formulation for Industrial Use (IFC 1967), or the International Association for the Properties of Water and Steam Industrial Formulation 1997 (IF 1997) steam properties, depending on the version used in the design heat balance. Appendix A details computation of test results of this test program for the Korean Standard Nuclear Power Plant Optimized Power Reactor-1000.

Calculation Using Performance Correction Curves

During the test runs, it would be unlikely that all plant operating conditions could be maintained exactly at the same values as the reference conditions. Therefore, it is necessary to correct test performance for the effect of such deviations.

The ASME PTC 6.0 code, which applies only to steam turbines, categorizes these corrections into two groups, Group 1 and Group 2 corrections. Group 1 includes corrections for variables primarily affecting feedwater heating systems, such as feedwater heater terminal temperature difference (TTD) and drain cooler approach (DCA), auxiliary extractions, enthalpy rise through pumps, extraction line pressure drop and heat losses, and generator power factor. Group 2 includes corrections for variables affecting turbine performance, such as throttle steam pressure, throttle steam quality and low-pressure turbine exhaust pressure.

The calculation procedure of the net electrical output in this performance diagnostic test program is similar to the ASME PTC 6 alternative test, which is commonly used for performance evaluation of nuclear steam turbines.

In this test program, the concern is not for the steam turbine, but for the overall turbine cycle. The correction procedure should be modified to extend the test boundary so that the feedwater heating systems are located inside the test boundary. In this case, most of Group 1 correction variables should be eliminated. Corrections applied to the computation of the net electrical output are as follows:

- Throttle steam pressure (at constant megawatts thermal [MWt])
- Throttle steam quality (at constant MWt)
- Low-pressure turbine exhaust pressure
- Generator power factor
- Steam generator thermal output (MWt) and deviation from the rated MWt
- Throttling loss at the high-pressure turbine governor valve

The ASME PTC 6 code proposes that unaccounted-for cycle losses should be assumed as steam generator leakage and, therefore, be subtracted from the measured final feedwater flow to obtain the steam generator outlet flow. However, in the nuclear power plant, it may be more reasonable to apply only the level change of the steam generator for calculation of the steam flow and use the calculated unaccounted-for cycle losses just for evaluating the cycle isolation conditions. This is because there are no vents and drains in the nuclear steam generation system except the continuous blowdown, which means that there is not a source of leakage when the blowdown system is isolated.

In the engineering judgment of the author, the PTC 6 proposal relating to the unaccounted-for cycle losses is applicable only to fossil-fired power plants.

Calculation Using Heat Balance Modeling

Because the nuclear steam turbines operate in the wet steam region, the steam quality of the turbine extraction steam to feedwater heaters or moisture separator reheaters must be determined for turbine cycle heat balance calculation. ASME PTC 6 code recommends several methods for determining steam quality in nuclear power plants, such as the tracer technique or heater drain flow and heat balance, these methods are still cost and labor intensive and impractical to apply to existing power plants because, in many cases, new test provisions for tracer injection and sampling or temporary test pipe sections for heater drain flow measurement are required.

This test program basically refers to the ASME PTC 6 full-scale test procedures, but some modifications as mentioned in Section 2 are made to best estimate the steam quality of the turbine extraction steam.

This approach allows full calculation of turbine cycle heat balance, and the calculations are easier and more practical to conduct relative to the ASME PTC 6 full scale test. Test uncertainty caused by using some design parameters is inevitable, but the magnitude of the increase in test uncertainty relative to the tracer technique or heater drain flow and heat balance method will be very small because stringent steam turbine performance characteristics are used. It should be noted that steam sampling and analysis in trace technique and heat drain flow measurement also yield relatively high measurement uncertainty.

Test Accomplishments

The main purpose of this performance diagnostic test program is to accurately estimate the performance parameters of individual turbine cycle components and plant thermal performance modeling for KHNP's operating nuclear power plant units. The test results will be used as the baseline data to monitor the performance deterioration of the turbine cycle and its components.

Conducting this test program over the past four years, there were additional accomplishments that resulted in the improvement of the electrical output as shown in Table 3-1. The improved electrical outputs have been identified after on-site actions to correct the detected output factors for each plant.

- Case 1: The Kori Nuclear Power Plant (KRN) #1 test program was conducted in 2008 and found the partial open state of a drain valve, which is the main steam-to-condenser dump line for checking the cycle isolation. There was about a 1.2-megawatt increase in output as soon as the plant operator closed the valve.
- Case 2: The Yong Gwang Nuclear Power Plant (YGN) #5 test program was conducted in 2008. A fault in the Y channel was identified in the steam generator #2 inlet feedwater flow nozzle after analysis of the flow nozzle calibration data in accordance with ASME PTC 6. Thus, it was recommended to change the average mode of the final feedwater channel into the X channel only. About a 3-megawatt increase was attained.
- Case 3: The Wolsong Nuclear Power Plant #3 test program was conducted in 2009 and identified that the condenser pressure limit (3.2 kPa [0.464 psi]) in winter required by the operating procedure resulted in an electrical output loss. About 1 megawatt increased per month in winter after removing the condenser pressure limit.
- Cases 4 and 5: The Ulchin Nuclear Power Plant (UCN) #3 & #4 test programs were conducted in 2009, and both units had the same problem. The second reheater heating steam isolation valves were only partially open, which should be fully open during normal operation. The valves were fully opened, resulting in about a 1.3-megawatt increase.
- Case 6: The YGN #3 test program was conducted in 2010 and found that the steam generator #2 inlet feedwater flow nozzle was fouled. Thus, the differential pressure transmitter in the field was corrected from the one installed for the diagnostic test, resulting in about a 2.5-megawatt increase.

- Case 7: The UCN #6 test program was conducted in 2010, which was the same as Case 2 above. This resulted in a 2.4-megawatt increase.
- Case 8: KRN #3 was conducted in 2011, which identified a low temperature indication of the steam generator inlet feedwater. The temperature in the field was an of average 1.38°C (2.48°F) lower than the test instrument, which resulted in about 0.35% over-estimation of the reactor thermal power. After checking for the signal loop of the related line, about a 1.9-megawatt increase was attained.
- Case 9: An overflow of the continuous vent for feedwater heaters in several nuclear power plants was found. The valve opening optimization was recommended. In 2010 YGN #3 showed the largest increase, about 1.5 megawatts.

Table 3-1
kW Output Improvement Resulting from Diagnostic Testing

No.	Unit	Cause of kW Output Loss by Test Result Analysis	Electrical Output Improvement	
			Modeling Simulation	On-Site Action
1	KRN #1	Partially open drain valve (main steam to condenser) found in cycle isolation	N/A	1.2 MW
2	YGN #5	Steam generator #2 inlet feedwater nozzle faulty on Y channel	3.275 MW	3.0 MW
3	WSN #3	Condenser pressure limit setting in winter	0.390 MW	1.0 MW
4	UCN #3	Partially open second reheater heating steam isolation valve	0.899 MW	0.9 MW
5	UCN #4	Partially open second reheater heating steam isolation valve	N/A	0.4 MW
6	YGN #3	Reactor thermal power overestimation from steam generator #2 inlet feedwater nozzle fouling	2.452 MW	2.5 MW
7	UCN #6	Steam generator #2 inlet feedwater nozzle faulty on Y channel	2.288 MW	2.4 MW

Table 3-1 (continued)
kW Output Improvement Resulting from Diagnostic Testing

No.	Unit	Cause of kW Output Loss by Test Result Analysis	Electrical Output Improvement	
			Modeling Simulation	On-Site Action
8	KRN #3	Reactor thermal power overestimation according to low indication of steam generator inlet feedwater temperature	N/A	1.9 MW
9	6 units not including YGN #3	Excess of feedwater heater continuous vent flow	1.481 MW for YGN #3	1.5 MW for YGN #3 and 4.5 MW for others
Total				19.3 MW

Section 4: Conclusions and Lessons Learned

Summary

In many nuclear power plants, turbine cycle performance is evaluated by conducting the ASME PTC 6 test using an alternative test methodology because the nuclear steam turbines operate in the wet steam region, which is the biggest limitation to conducting the full-scale test. Under these technical restrictions, operators in the nuclear power plant experience difficulties in managing performance histories of individual turbine cycle components and their impact on the electric power output. Therefore, they cannot deal effectively with plant anomalies that are related to the lost electrical output performance.

The diagnostic test program for nuclear turbine cycle performance at KHNP is basically following the technical guidelines in the ASME PTC 6 full-scale test method with minimum deviations and modification. Thus, it is possible to create thermal heat balance modeling by using measured performance parameters that are expected to simulate the electrical output performance at rated conditions.

This test program can be used in combination with the contractual acceptance test of new units to achieve benchmark performance parameters of overall turbine cycle and individual turbine cycle components in new and clean conditions. Also, if this test is periodically conducted on existing units, it can be used to identify performance deterioration of individual turbine cycle components and quantify their impact on the electrical output. KHNP is planning a six-year-cycle for this testing to obtain accurate absolute performance levels of the turbine cycle components and to update the modeling.

Conclusion

It is reported that the heat balance modeling was different by approximately a maximum of 0.06% from the test results from ASME PTC 6 alternative test method. Using the plant performance modeling created from the turbine cycle performance diagnostic test with precision test instrumentations, accurate information about the turbine cycle can be provided and the benefits or loss resulting from any performance change can be estimated.

Section 5: References

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2. ASME PTC 12.2 -1998 Performance Test Code on Steam Surface Condenser
3. F. G. Baily, J. A. Booth, K. C. Cotton, and E. H. Miller, "Predicting the Performance of 1800-rpm Large Steam Turbine Generators Operating with Light-Water-Cooled Reactors," GET-6020 (February 1973).
4. ASME PTC 12.4 -1992 Moisture Separator and Reheaters
5. ASME PTC 6A -2000 Appendix to PTC 6 , The Test Code for Steam Turbines
6. ASME PTC 12.1 -2000 Closed Feedwater Heaters
7. K. C. Cotton, *Evaluating and Improving Steam Turbine Performance.* Cotton Fact (1993).

Appendix A: Computation of Test Results

Corrected kW Output Calculation Using Performance Correction Curves

The measured electrical output is corrected using the following sample equation. The corrected kilowatt ($kW_{corrected}$) is the expected net electrical output when the correction variables operate at the reference conditions:

$$kW_{corrected} = \frac{kW_{Adjusted}}{CFK_1 \times CFK_2 \times CFK_3 \times CFK_{MWt} \times CFK_{Throttle}} \quad \text{Eq. A-1}$$

Where:

CFK_1 = kW output correction factor for throttle steam pressure

CFK_2 = kW output correction factor for throttle steam quality

CFK_3 = kW output correction factor for low-pressure turbine exhaust pressure

CFK_{MWt} = $MW_{thermal\_test}$ / $MW_{thermal\_\_design}$

$CFK_{throttle}$ = kW output correction factor for high-pressure control valve throttling

$kW_{adjusted}$ = Electrical output adjusted to nominal power factor, kW

Steam generator thermal output is defined as:

$$MW_{thermal} = (Q_{FFW} - Q_{SG\ LEVEL}) \times (h_{SGOUT} - h_{FFW}) + Q_b \times (h_{bf} - h_{FFW}) \quad \text{Eq. A-2}$$

Where:

Q_{FFW} = Steam generator inlet final feedwater flow

$Q_{SG\ LEVEL}$ = Change in steam generator drum level in mass flow

Q_b = Steam generator blowdown flow

h_{FFW} = Steam generator inlet final feedwater enthalpy

h_{SGOUT} = Steam generator outlet steam enthalpy

h_{bf} = Steam generator blowdown enthalpy

The correction factor for the high-pressure turbine governing valve throttling can be calculated using the method described in Paragraph 3-13 of the ASME PTC 6.0 code as shown below:

$$\%\Delta HR = \frac{w_n}{w_{mo}} \times \left(\frac{\Delta p_{mo} - \Delta p_{vwo}}{P_t} \right) \times k \qquad \text{Eq. A-3}$$

$$CFK_{throttle} = 1 - \%\Delta HR \qquad \text{Eq. A-4}$$

Where:

W_n/W_{mo}	=	Ratio of steam flow through control valve #4 to total steam flow
ΔP_{mo}	=	Pressure drop across control valve #4 during the test run
ΔP_{vwo}	=	Pressure drop across control valve #4 at the valve wide open condition
P_t	=	Throttle pressure
k	=	% effect on heat rate for 1% change in pressure drop (0.15 for nuclear)

Determination of the turbine cycle heat rate is meaningless in a nuclear power plant because the steam generator thermal output is bound up with the licensed reactor thermal power, and the test measured electrical output is to be corrected for the rated steam generator thermal output. From the equation below, the percent change in kW output is exactly same as the percent change in turbine cycle heat rate, and the corrected kW output should be interpreted as performance level rather than unit maximum capacity or specified conditions for heat rate calculation in the fossil power plant.

$$HR_{corrected} = \frac{Steam\ Generator\ Thermal\ Output(=Constant)}{kW_{corrected}} \qquad \text{Eq. A-5}$$

Test Cycle Heat Balance Calculation

The calculation procedures for performance parameters of steam turbine and individual turbine cycle components are as follows.

Steam Turbine Performance Parameters

From the Mollier Chart shown in Figure 2-1, the high-pressure turbine section efficiency is defined as:

$$\eta_{MS\ to\ HPELEP} = \frac{h_{MS} - h_{HPELEP}}{h_{MS} - h_{HX\,S2}} \qquad \text{Eq. A-6}$$

$$\eta_{BWL\ to\ HPELEP} = \frac{h_{BWL} - h_{HPELEP}}{h_{BWL} - h_{HXS1}} \quad \textit{Eq. A-7}$$

Where:

h_{MS} = Main steam enthalpy, kJ/kg

h_{HPBWL} = High-pressure turbine bowl steam enthalpy, kJ/kg

h_{HPELEP} = High-pressure turbine exhaust steam enthalpy, kJ/kg

h_{HXS1} = Isentropic enthalpy at expansion from main steam to high-pressure exhaust, kJ/kg

h_{HXS2} = Isentropic enthalpy at expansion from high-pressure bowl to high-pressure exhaust, kJ/kg

Because the high-pressure turbine section is operating in the wet region, its exhaust steam enthalpy, H_{HPEXH}, cannot be determined from direct measurement of steam pressure and temperature. As such, calibrated ultrasonic flow meters are used to measure the moisture separator reheater (MSR) drain flows, and the high-pressure turbine exhaust steam enthalpy is determined using a mass and energy balance calculation around the moisture separator reheater.

Steam enthalpy at the high-pressure turbine extraction stages is then determined from the expansion line of the 50-in.(127-cm) radius curve with design expansion slop reading the values at the measured extraction pressure.

From the Mollier Chart shown in Figure 2-1, the low-pressure turbine section efficiency is defined as:

$$\eta_{UEEP} = \frac{h_{LPBWL} - h_{LPUEEP}}{h_{LPBWL} - h_{LXS1}} \quad \textit{Eq. A-8}$$

$$\eta_{ELEP} = \frac{h_{LPBWL} - h_{LPELEP}}{h_{LPBWL} - h_{LXS1}} \quad \textit{Eq. A-9}$$

$$\eta_{BASE\ ELEP} = \frac{h_{LPBWL} - h_{BASE\ ELEP}}{h_{LPBWL} - h_{LXS1}} \quad \textit{Eq. A-10}$$

Where:

h_{LPBWL} = Low-pressure turbine bowl steam enthalpy

h_{LPELEP} = Low-pressure turbine expansion line end point

$h_{BASE\ ELEP}$ = Low-pressure turbine base expansion line end point

H_{LPUEEP} = Low-pressure turbine used energy end point

H_{LXS1} = Isentropic enthalpy for expansion from low-pressure bowl to low-pressure exhaust

The value of turbine efficiency that includes all the losses is based on the actual end point. The actual steam condition exiting at the condenser is called the *used energy end point* (UEEP). It is plotted in Figure 2-1 at the low-pressure turbine exhaust pressure, and the enthalpy, H_{UEEP}, is the enthalpy of the steam leaving the last stage.

Because the low-pressure turbine exhaust is in the wet region, the H_{UEEP} should be determined from the mass flow and energy balance calculation around the overall steam turbine including the generator power output, which is the same method as the ASME PTC 6 full-scale test procedure.

$$Q_{LP\ EXH} = Q_{MS} - \sum_{j=1}^{n} Q_{leakoff}\, j - \sum_{j=1}^{n} Q_{extraction}\, j - Q_{MS\ DRN} \qquad \text{Eq. A-11}$$

$$h_{UEEP} = \frac{Q_{MS} \times h_{MS} - \sum_{j=1}^{n} Q_{leakoff}\, j \times h_{leakoff}\, j - \sum_{j=1}^{n} Q_{extraction}\, j \times h_{extraction}\, j - Q_{MS\ DRN} \times h_{MS\ DRN} - KW_{shaft}}{Q_{LP\ EXH}}$$

Eq. A-12

Where:

$Q_{LP\ EXH}$	=	Low-pressure turbine exhaust steam flow to condenser
Q_{MS}	=	Main steam flow
h_{MS}	=	Main steam enthalpy
$Q_{leakoff}$	=	Control valve and turbine shaft leak-off steam flow
$h_{leakoff}$	=	Control valve and turbine shaft leak-off steam enthalpy
$Q_{extraction}$	=	Heater, moisture separator reheater, and feedwater pump turbine (fwpt) extraction steam flow
$h_{extraction}$	=	Heater, moisture separator reheater, and fwpt extraction steam enthalpy
$Q_{MS\ DRN}$	=	Moisture separator drain flow
$h_{MS\ DRN}$	=	Moisture separator drain enthalpy
KW_{Shaft}	=	Turbine shaft power (=kilowatt$_{measured}$ + fixed loss + generator loss)

Most of the velocity energy in the steam leaving the last stage of the low-pressure turbines is lost when the steam flow is turned down to the condenser in the exhaust hood. However, the expansion end-point efficiency does not take this loss into account. In fact, the expansion line end-point is a fictitious point that cannot be measured because it is based on the assumption of an imaginary turbine stage capable of utilizing the velocity energy leaving the last stage. The expansion line end-point efficiency is a very important performance parameter for the low-pressure turbines because the used energy end-point efficiency changes depending on the condenser pressure and turbine load; the expansion line end-point efficiency maintains a constant level regardless of operating conditions. The H_{ELEP} can be calculated from H_{UEEP} and the total exhaust loss curves submitted by turbine manufacture with following equation:

$$h_{ELEP} = h_{UEEP} - TEL \times (1 - 0.01Y) \times 0.87 \times (1 - 0.0065Y) \qquad \text{Eq. A-13}$$

Where:

TEL	=	Total exhaust loss (from exhaust loss curve)
Y	=	% weighted average moisture at the expansion line end point
0.87	=	Low-pressure exhaust fictitious stage dry efficiency
1-0.0065Y	=	Moisture loss correction

The flow function defined in the equation below is the flow passing capacity of the turbine stage, and if this value changes, the stage steam path must be the reason. Increased leakage areas, damage, and deposits to either rotating or stationary blades can cause the flow function to change.

$$\text{Flow Function } (K) = \frac{Q}{\sqrt{\frac{P}{V}}} = \text{a constant} \qquad \text{Eq. A-14}$$

Where:

Q	=	Steam flow into the next stage
P	=	Pressure of steam into the next stage
V	=	Specific volume of steam into the next stage

Moisture Separator Reheater Performance Parameters

During expansion through the high-pressure section, the moisture content in the steam increases to approximately 12% at the high-pressure turbine exhaust. Moisture in the cycle steam reduces the mechanical efficiency in the low-pressure turbines and causes erosion of low-pressure turbine components in the steam path. The function of moisture separator reheater, located between the high-pressure and low-pressure sections, is to remove moisture from the steam in the high-pressure turbine exhaust and to reheat the dried steam before it flows into the low-pressure turbine sections.

Four factors contribute to determining the overall thermal performance of the moisture separator reheater and its effect on the net electrical output. The two most important performance parameters are the pressure drop of the cycle steam as it passes through the moisture separator reheater and the moisture content, or quality, of the cycle steam leaving the moisture separator.

The other two, terminal temperature difference (TTD) and excess heating steam flow, have less effect on the net electrical output. It is desirable to have low cycle steam pressure drop, low moisture content in the cycle steam exiting moisture separator, and low thermal temperature difference to optimize the benefit of the moisture separator reheater's effect on the plant performance, because the reheater uses the cycle steam for a heating source. Also, excess steam flow should be held at a value as low as practical that is consistent with reheater design requirements in order to minimize a negative impact on the net electrical output.

Following are calculation procedures for the performance parameters of the moisture separator reheater configured in Figure A-1.

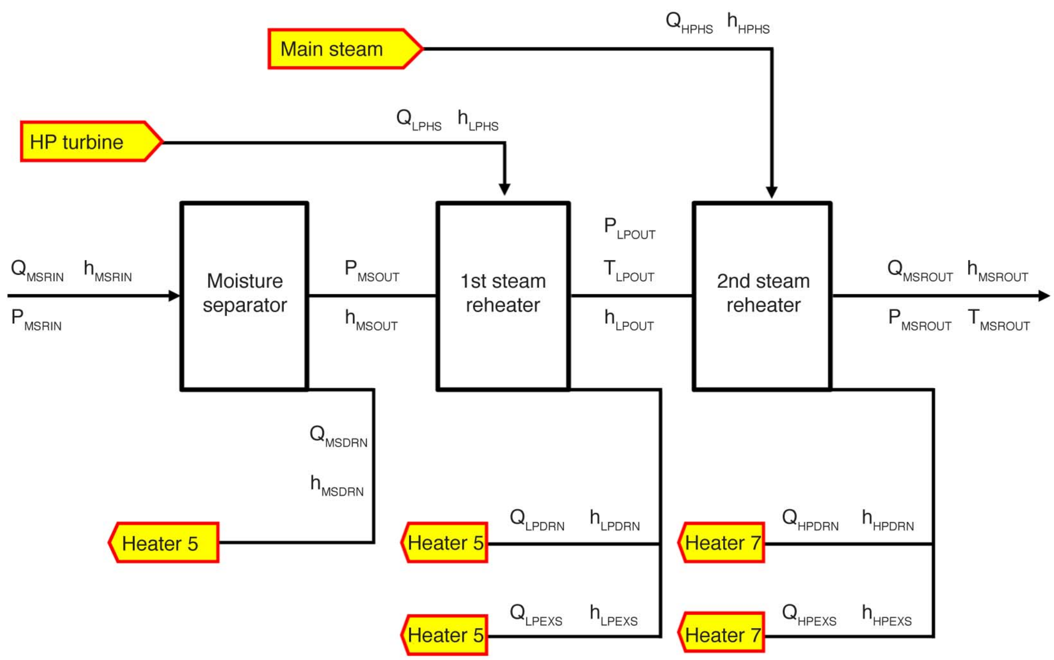


Figure A-1
Korean Standard Nuclear Power Plant Optimized Power Reactor-1000 Moisture Separator Reheater Configurations

Moisture Separation Effectiveness

One of the significant parameters involved in the moisture separator reheater performance calculation is the moisture separator outlet quality, or moisture separation effectiveness. As the moisture separator outlet quality decreases, increased surface amounts of reheater tube bundles are required to evaporate the moisture that is exiting the separator. This results in increased reheater thermal temperature and increased heating steam flow.

The moisture separator outlet quality, or moisture separation effectiveness, can be calculated from direct measurement of moisture separator drain flow or from the reheater energy balance method with following steps.

Step 1. Q_{MSDRN} is assumed for the first iteration

$$Q_{MSDRN\_Assumed} = Assumed\ Value \qquad \text{Eq. A-15}$$

Step 2. Using the thermal energy balance, determine the cycle steam interstage enthalpy at the moisture separator outlet.

$$h_{MSOUT} = h_{LPOUT} - \frac{Q_{LPDRN} \times (h_{LPHS} - h_{LPDRN})}{Q_{MSRIN} - Q_{MSDRN\_Assumed}} \qquad \text{Eq. A-16}$$

Where:

Q_{MSRIN} = Moisture separator reheater inlet cycle steam flow
= High-pressure turbine inlet throttle steam flow
- CV steam leak-off flow (from Design C-Factor)
- High-pressure interstage extraction steam flow to heater #6
- High-pressure interstage extraction steam flow to low-pressure reheater
- High-pressure interstage extraction steam flow to heater #5
- High-pressure shaft packing leak-off steam flow (from Design C-Factor)

Q_{MSDRN} = Moisture separator drain flow

h_{LPOUT} = Low-pressure reheater outlet cycle steam enthalpy
[Note: Refer to low-pressure (first reheater) thermal temperature difference calculation.]

h_{LPHS} = Low-pressure reheater heating steam enthalpy
[Note: From the high-pressure turbine expansion line]

Q_{LPDRN} = Low-pressure reheater drain flow

h_{LPDRN} = Low-pressure reheater drain enthalpy

Step 3. With the moisture separator reheater inlet cycle steam flow and moisture content, which are obtained from the turbine cycle heat balance calculation, an iterative method is used to determine the moisture separator drain flow for which difference with the value from following equation converges at 0 (zero).

$$Q_{MSDRN} = \frac{Q_{MSRIN} \times h_{MSRIN} - (Q_{MSRIN} - Q_{MSDRN\_Assumed}) \times h_{MSOUT}}{h_{MSDRN}} \qquad \text{Eq. A-17}$$

Where:

Q_{MSRIN} = Moisture separator reheater inlet cycle steam flow

h_{MSRIN} = Moisture separator reheater inlet cycle steam enthalpy

h_{MSOUT} = Moisture separator outlet cycle steam enthalpy

h_{MSDRN} = Moisture separator drain enthalpy

Step 4. Then the moisture separator effectiveness is determined from the following formula:

$$\eta_{MS} = \frac{Q_{MSDRN}}{M_{MSRIN}} \qquad \textit{Eq. A-18}$$

Where:

η_{MS} = Moisture separation effectiveness, %

Q_{MSDRN} = Moisture separator drain flow, kg/hr

M_{MSRIN} = Moisture content entering moisture separator, kg/hr

After moisture separation and removal, another function of the moisture separator reheater is to reheat the main cycle steam, performed in either one or two stages, before it enters the low-pressure turbine. The method for evaluating the performance of this heating function is by determination of the thermal temperature difference for each stage.

Terminal temperature difference for the low-pressure (or first stage) reheater is determined with following process:

$$TTD_{LP} = T_{LPHS\_SAT} - T_{LPOUT} \qquad \textit{Eq. A-19}$$

Where:

TTD_{LP}	=	Terminal temperature difference for the low-pressure reheater
T_{LPHS_SAT}	=	Saturation temperature of heating steam entering the low-pressure reheater
T_{LPOUT}	=	Temperature of cycle steam at the low-pressure reheater outlet

Using the thermal energy balance method, determine the cycle steam interstage enthalpy at the low pressure reheater outlet (h_{LPOUT}).

$$h_{LPOUT} = h_{MSROUT} - \frac{Q_{HPDRN} \times (h_{HPHS} - h_{HPDRN})}{Q_{MSRIN} - Q_{MSDRN}} \qquad \textit{Eq. A-20}$$

Where:

Q_{MSDRN}	=	Moisture separator drain flow
$h_{MSRROUT}$	=	Moisture separator reheater outlet cycle steam enthalpy
h_{HPHS}	=	High-pressure reheater heating steam enthalpy (= $h_{MAIN\ STEAM}$)
Q_{HPDRN}	=	High-pressure reheater drain flow
h_{HPDRN}	=	High-pressure reheater drain enthalpy

$$P_{LPOUT} = \frac{P_{MSOUT} - P_{MSROUT}}{2}$$ Eq. A-21

Where:

P_{LPOUT} = Pressure of cycle steam at low-pressure reheater outlet

P_{MSROUT} = Moisture separator reheater outlet cycle steam pressure

P_{MSOUT} = Moisture separator outlet cycle steam pressure

Then T_{LPOUT} which is a function of the cycle steam interstage pressure, P_{LPOUT}, and enthalpy, h_{LPOUT}, can be determined using the steam property table.

The thermal temperature difference of the high-pressure (second stage) reheater is determined with following equation:

$$TTD_{HP} = T_{HPHS\_SAT} - T_{MSROUT}$$ Eq. A-22

Where:

TTD_{HP} = Terminal temperature difference for the high-pressure reheater

T_{HPHS_SAT} = Saturation temperature of the heating steam entering the high-pressure reheater

T_{MSROUT} = Moisture separator reheater outlet cycle steam temperature

In this test program, design flow rate of excess steam in percentage of reheater drain flow is used for the reheater energy balance calculation, as wet steam flow measurement is not accurate and normally station permanent flow elements to measure this flow are not even installed.

Moisture separator reheater shell side cycle steam pressure drop and tube side heating steam pressure drop was also measured with static pressures transmitters. These parameters will be used for the plant performance modeling.

Feedwater Heater Performance Parameters

Most nuclear turbine cycles have two strings of high-pressure feedwater heaters and two or three strings of low-pressure feedwater heaters that preheat the feedwater using the heat available in turbine extraction steam.

The two most commonly used measures of feedwater performance are the terminal temperature difference and the drain cooler approach (DCA).

The thermal temperature difference of a feedwater heater is the difference between the temperature of the extraction steam being condensed on the shell side of the heater and the temperature of the feedwater leaving the heater.

$$TTD = T_{SAT} - T_{OUT} \quad \text{Eq. A-23}$$

Where:

T_{SAT} = Saturated steam temperature corresponding to the steam inlet pressure

T_{OUT} = Feedwater outlet temperature

The drain cooler approach of a feedwater heater is the difference between the sub-cooled condensate leaving the heater and the feedwater entering the heater. It can be a direct measure of the heat transfer capability of the drain cooler section of the heater.

$$DCA = T_{DRN} - T_{IN} \quad \text{Eq. A-24}$$

Where:

T_{DRN} = Drain cooling outlet temperature

T_{IN} = Feedwater inlet temperature

A more rigorous approach is to use the manufacturer design data in conjunction with basic heat transfer relationships to compute the expected thermal temperature difference and drain cooler approach at reference conditions. These calculations are outlined in detail in the ASME PTC 12.1 code, which outlines a calculation procedure to compute expected thermal temperature difference, drain cooler approach, and pressure drops for the feedwater heater for comparison with the actual measurements. Unfortunately, this code was not used in this test program because the detailed design information (listed here) required from the manufacturer was not available:

- Heat transfer surface areas for each heater zone
- Steam- and water-side fouling resistances
- Steam- and water-side film resistances
- Expected heat transfer rate for each zone
- Expected inlet and outlet pressures, temperatures, and flows for the extraction steam, feedwater flow, and drain flow

Pumps

The three most important pumps in the nuclear turbine cycle are the feedwater booster pump, the main feedwater pump, and the condensate pump. These pumps have a direct role in the thermodynamic efficiency of the turbine cycle because they increase the enthalpy of the feedwater. However, the enthalpy rise is very small relative to other components. Therefore, inefficiencies in these pumps play a very minor role in affecting the net electrical output. Reliability is generally of greater concern than efficiency for these pumps. The pumps efficiency is defined as:

$$\eta_{Pump} = \frac{h_{Suction} - h_{DXS1}}{h_{Suction} - h_{Discharge}}$$ *Eq. A-25*

Where:

$h_{Suction}$ = Feedwater enthalpy at the pump suction

$h_{Discharge}$ = Feedwater enthalpy at the pump discharge

h_{DXS1} = Isentropic enthalpy at pressure rise from the pump suction to the pump discharge

Condenser

A primary factor that affects the efficiency of the Rankin steam cycle is the temperature at which the waste heat is rejected to the environment. The waste heat is rejected to the environment by the condenser, and any performance deficiencies in the condenser have a significant impact on the turbine cycle performance.

There are several factors that contribute to determining the thermal performance of the condenser. These performance parameters are condenser duty, log mean temperature difference (LMTD), and overall heat transfer coefficient. These parameters have a direct impact on the condenser steam pressure at a given cooling water temperature and flow.

The first parameter required for condenser performance evaluation is the condenser duty, Q_a, which is the waste heat load that the condenser is rejecting to the cooling water.

In this test program, all energy inputs and outputs around the condenser were readily measured or calculated from the turbine cycle heat balance calculation, which makes it possible to compute the condenser duty as below.

$$Q_a = \left(Q_{LPEXH} + \sum_{i=0}^{n} Q_i \right) - Q_{HOTWELL}$$ *Eq. A-26*

Where:

Q_a = Condenser duty

Q_{LPEXH} = Heat into condenser from low-pressure exhaust steam

Q_i = Heat into condenser from other sources

$Q_{HOTWELL}$ = Heat out of condenser

Once the condenser duty has been determined, the actual heat transfer coefficient of the tube bundle is computed with following equation.

$$K_a = \frac{Qa}{A \times LMTD}$$ *Eq. A-27*

Where:

A = Heat exchange area

LMTD = Log mean temperature difference as defined below

$$LMTD = \frac{T_{OUT} - T_{IN}}{ln\left(\frac{T_{SAT} - T_{IN}}{T_{SAT} - T_{OUT}}\right)}$$ *Eq. A-28*

Where:

T_{SAT} = Saturation temperature of the turbine exhaust steam

T_{IN} = Condenser inlet cooling water temperature

T_{OUT} = Condenser outlet cooling water temperature

In this test program, the ASME PTC 12.2 code was also used to calculate the difference between the test adjusted and the design reference steam pressure.

Appendix B:Modeling Sample

Attachment 1: Measurement Posting Diagram for the Yong Gwang Nuclear Power Plant (YGN) #4

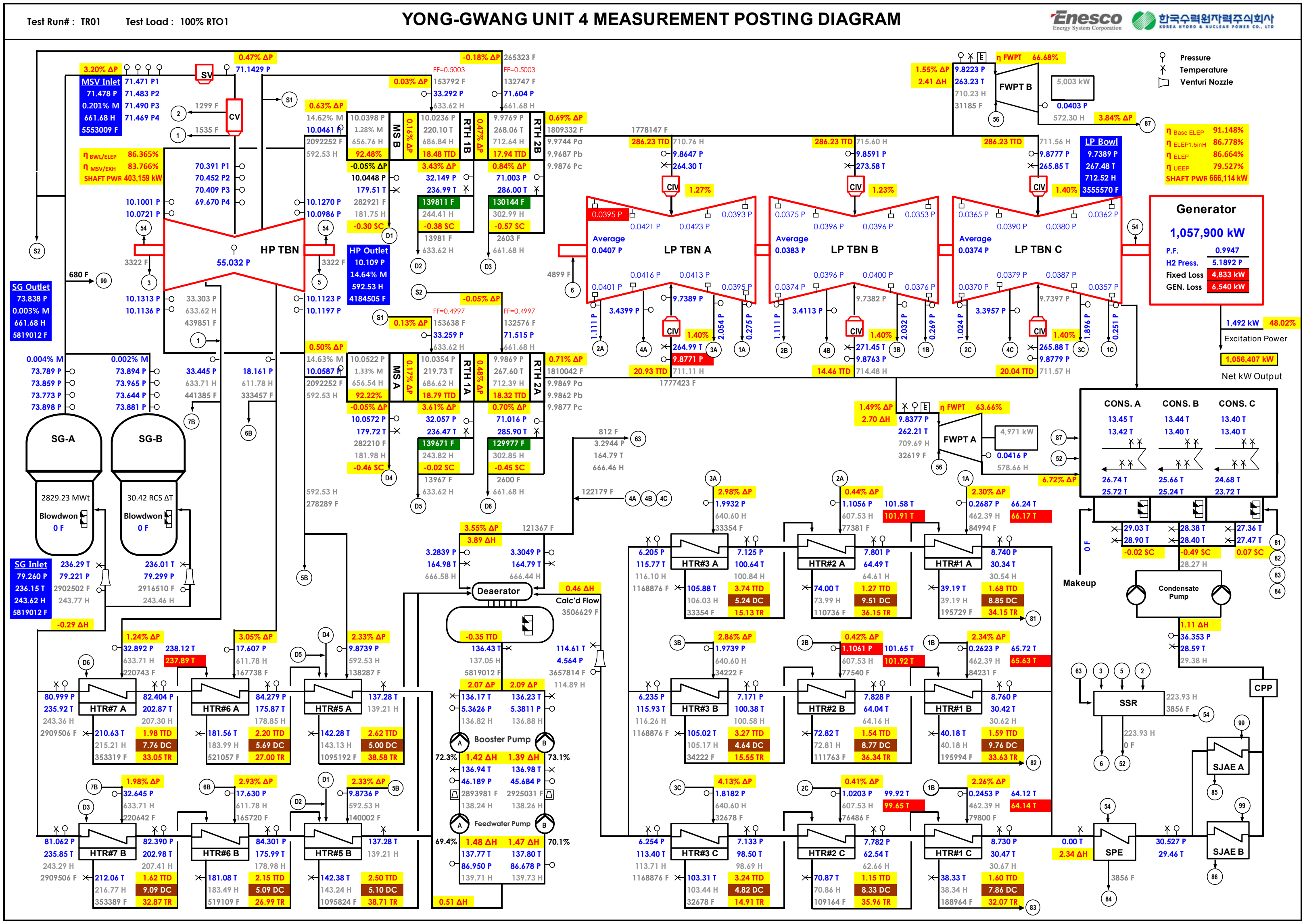
Attachment 2: Test Cycle Heat Balance at Test Condition for YGN #4

Attachment 3: Performance Data Calculated from Test Cycle Heat Balance at Test Condition for YGN #4

Attachment 4: Test Cycle Heat Balance at Rated Condition for YGN #4

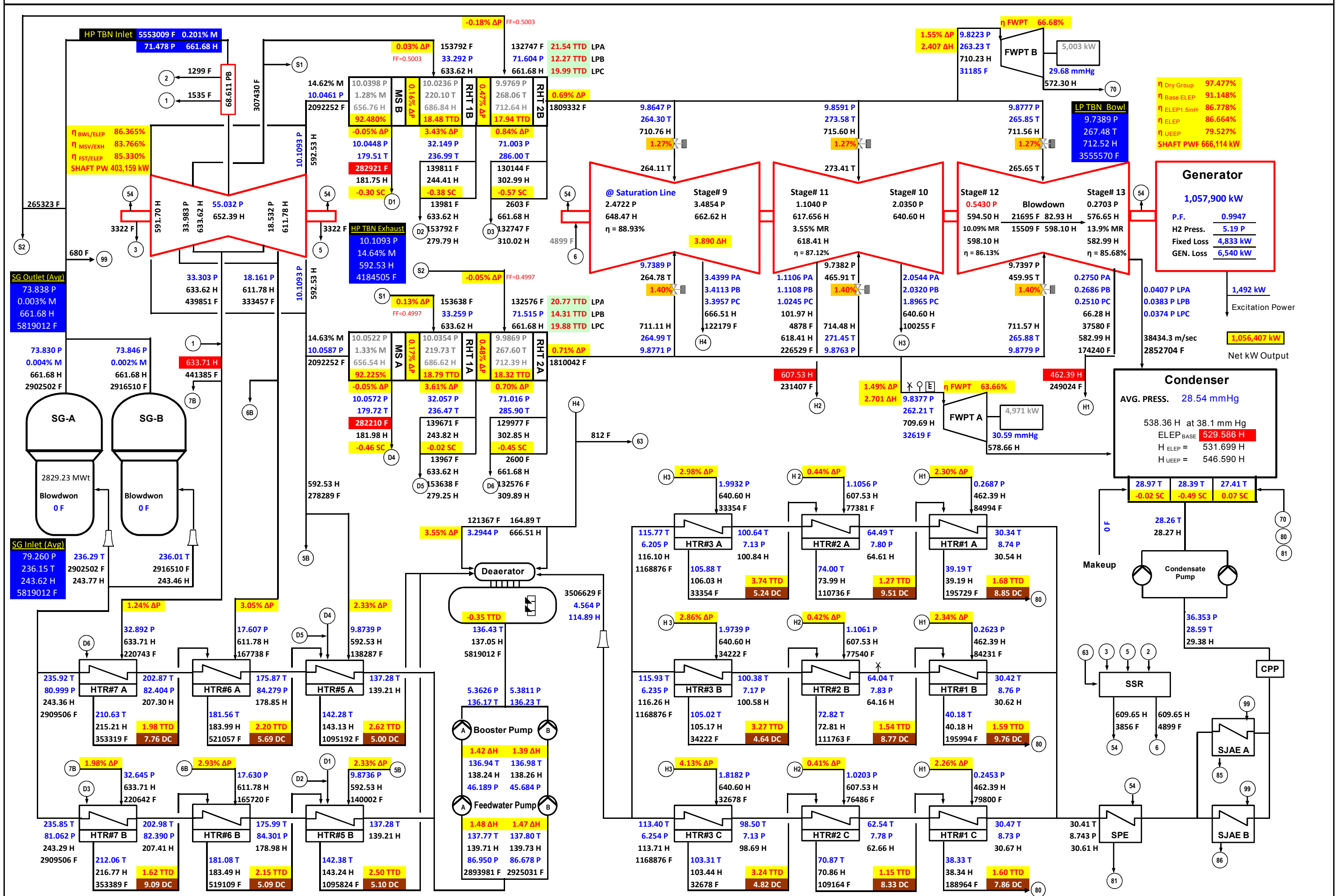
Attachment 5: Simulation Variables of Test Cycle Heat Balance at Rated Condition for YGN #4

Attachment 1



Attachment 2

YONG-GWANG UNIT 4 TEST CYCLE HEAT BALANCE AT TEST CONDITIONS



VALVE BEST POINT NET HEAT RATE = 5819012F x (661.68H - 243.62H) + 0F x (305.65H - 243.62H) / 1,057,900 kW = 2299.56 kcal/kwh

TEST POINT 100% RTO1 (TR01)

ON 1967 ASME STEAM TABLES
M - FLOW - kg/hr
P - PRESSURE AT - kg/cm2
H - ENTHALPY - kcal/kg
T - TEMPERATURE - C DEGREES

1054645 KW 38.1 MM HG ABS. 0.2 PCT MU
TC6F 43.0 IN. LSB 1800 RPM
72.768 ATMA 660.40 KCAL/KG 2 STAGE REHEAT
GEN 1219600 KVA 0.90 PF
MKS UNITS

Program by DOH-HOON, KOO

Attachment 3

Test Cycle Heat Balance Conversion

Turbine Inlet Throttle Steam Flow	5,553,009 F
Reactor MW Thermal	2,829.23 MWt

Turbine Inlet Steam Conditions

	SG A	SG B
Steam Generator Outlet Steam Pressure	73.830 P	73.846 P
Steam Generator Outlet Steam Moisture	0.0037%	0.0024%
Steam Generator Blowdown Flow	0.0 F	0.0 F

LP A Turibne Exhaust Pressure	29.94 mmHg
LP B Turibne Exhaust Pressure	28.18 mmHg
LP C Turibne Exhaust Pressure	27.49 mmHg
Generator Power Factor	0.9947

Current Throttle Flow Ratio 0.9799

Turbine Section Efficiency Level

HP Turbine Section	2.1812%
LP Turbine Section	2.3065%

Steam Turbine Stage Flow and Pressure

	Stgae Flow	Stage Press	Ethalpy
HP Turbine STG# 1	5,550,176	55.032 P	652.39 H
HP Turbine STG# 3	4,802,895	33.983 P	633.62 H
HP Turbine STG# 5	4,469,437	18.532 P	611.78 H
LP Turbine Bowl	3,555,570	9.7389 P	712.52 H
LP Turbine STG# 9	3,433,391	3.4854 P	662.62 H
LP Turbine STG# 10	3,333,136	2.0350 P	640.60 H
LP Turbine STG# 11	3,101,729	1.1040 P	618.41 H
LP Turbine STG# 12	3,064,525	0.5430 P	598.10 H
LP Turbine STG# 13	2,852,704	0.2703 P	582.99 H

Steam Turbine Extraction Stage Shell K-Factor

	Deisgn_K	Test_K	Difference
HP Turbine STG# 1	147,969	140,098	-7,871
HP Turbine STG# 3	199,603	193,158	-6,446
HP Turbine STG# 5	328,661	321,771	-6,889
LP Turbine Bowl	614,746	573,588	-41,158
LP Turbine STG# 9	1,444,465	1,383,490	-60,975
LP Turbine STG# 10	2,325,483	2,188,779	-136,704
LP Turbine STG# 11	3,918,580	3,627,253	-291,328
LP Turbine STG# 12	7,158,028	7,036,467	-121,561
LP Turbine STG# 13	12,726,924	12,778,234	51,310

CV & Packing Leakoff Flow C-Factor

Total Control Valve Leakoff Flow @VWO	0.05% MS Flow
N1LP Packing Leakoff Flow C-Factor	427.421
N2LP Packing Leakoff Flow C-Factor	427.421
LP TBN Seal Steam Supply Flow	4899 F
TBN Shaft End Packing to SPE Flow	3856 F

Stage to Flange Pressure Drop 2.00%

Steam Turbine Design Specification

VWO Throttle Steam Flow	5782335 F
VWO Throttle Steam Pressure	72.768 P
VWO Throttle Steam Enthalpy	660.40 H
VWO HP TBN Exhaust Pressure	9.962 P
VWO HP TBN Expansion Slope(ΔH/ΔS)	2632.894
TFR at First Admission Point	0.850
LP Turbine Exhaust Pressure	0.0518 P

Generator Design Specification

Maximum Generator Capacity in KVA	1219600 KVA
Rated Hydrogen Pressure	5.19 P

	Assumed	Difference
Moist. Separator B Drain Flow	282921 F	0 F
Moist. Separator A Drain Flow	282210 F	0 F
LP Section Base ELEP	529.586 H	0.000 H
LP Section Efficiency Level	2.30655	0.00
Mixture of Steam and Moisture to HTR #2	607.526 H	0.000 H
Ext. Steam to HTR #7 Aft Mixing w. CVLO#1	633.714 H	0.000 H
Total Ext. to HTR #1 Aft Mixing w. 12th Sta	462.387 H	0.000 H

% Deviation from Design Excitation Power 48.02%

Cycle Steam Pressure Drop

Steam Generator Outlet to MSV Inlet	3.20%
HP Exhaust to Moisture Separator Inlet	0.56%
MSR Outlet to Combined Intercept	1.10%
Through Combined Intercept Valve	1.35%

FWPT Performance Specification

	FWPT A	FWPT B
FWPT Efficiency	63.66%	66.68%
FWPT Exhaust Steam Pressure Drop	6.72%	3.84%
FWPT Driving Steam Pressure Drop	1.49%	1.55%
FWPT Driving Steam Hot Extraction Loss	2.70 ΔH	2.41 ΔH

Heater Performance Specification

	Ext Stm ΔP	TTD	DC	FW ΔP
Heater #7A	1.24%	1.98 ℃	7.76 ℃	1.71% ΔP
Heater #7B	1.98%	1.62 ℃	9.09 ℃	1.61% ΔP
Heater #6A	3.05%	2.20 ℃	5.69 ℃	2.23% ΔP
Heater #6B	2.93%	2.15 ℃	5.09 ℃	2.27% ΔP
Heater #5A	2.33%	2.62 ℃	5.00 ℃	2.92% ΔP
Heater #5B	2.33%	2.50 ℃	5.10 ℃	2.89% ΔP
Dearator	3.55%	-0.35 ℃	3.89 ΔH Hot Extraction	
Heater #3A	2.98%	3.74 ℃	5.24 ℃	12.92% ΔP
Heater #3B	2.86%	3.27 ℃	4.64 ℃	13.05% ΔP
Heater #3C	4.13%	3.24 ℃	4.82 ℃	12.32% ΔP
Heater #2A	0.44%	1.27 ℃	9.51 ℃	8.67% ΔP
Heater #2B	0.42%	1.54 ℃	8.77 ℃	8.39% ΔP
Heater #2C	0.41%	1.15 ℃	8.33 ℃	8.33% ΔP
Heater #1A	2.30%	1.68 ℃	8.85 ℃	10.77% ΔP
Heater #1B	2.34%	1.59 ℃	9.76 ℃	10.47% ΔP
Heater #1C	2.26%	1.60 ℃	7.86 ℃	11.00% ΔP

Condenser Performance Specification

	Condenser A	Condenser B	Condenser C
Condenser Subcooling	-0.02 ℃	-0.49 ℃	0.07 ℃

MSR Performance Specification

	MSR A	MSR B
HP Exhaust to Moisture Separator Inlet	0.50%	0.63%
2nd Reheater Heating Steam Pressure Drop	-0.05%	-0.18%
1st Reheater Heating Steam Pressure Drop	0.13%	0.03%
Shell PD through 1st Reheater	0.17%	0.16%
Shell PD through 2nd Reheater	0.48%	0.47%
Total Cycle Steam Pressure Drop	0.71%	0.69%
Through Combined Intercept Valve - LPA	1.40%	1.27%
Through Combined Intercept Valve - LPB	1.40%	1.27%
Through Combined Intercept Valve - LPC	1.40%	1.27%

Moisture Separator Effectiveness	92.22%	92.48%
1st Reheater TTD	16.74 ℃	16.89 ℃
2nd Reheater TTD to LP A	20.91 ℃	21.70 ℃
2nd Reheater TTD to LP B	14.44 ℃	12.42 ℃
2nd Reheater TTD to LP C	20.02 ℃	20.15 ℃
MS Drain Tank Subcooling	-0.46 ℃	-0.30 ℃
1st Reheater Drain Tank Subcooling	-0.02 ℃	-0.38 ℃
2nd Reheater Drain Tank Subcooling	-0.45 ℃	-0.57 ℃
1st Reheater Excess Steam	10.00%	10.00%
2nd Reheater Excess Steam	2.00%	2.00%
Tube PD through 1st Reheater	3.61%	3.43%
Tube PD through 2nd Reheater	0.70%	0.84%

Pump Performance Specification

SPE H Rise	
H Rise from Condenser Hotwell to HTR#1A	2.27 ΔH
H Rise from Condenser Hotwell to HTR#1B	2.34 ΔH
H Rise from Condenser Hotwell to HTR#1C	2.40 ΔH

Condensate Extraction Pump

- Pump Enthalpy Rise	1.11 ΔH
- Pump Discharge Pressure	36.353 P
- %DP From COP Dischare to HTR#1	75.95%
- %DP From HTR#1 Inlet to D/A Inlet	47.80%

Feedwater Booster Pump	FWBP A	FWBP B
- Pump Suction Head from D/A	2.07 ΔP	2.09 ΔP
- Pump Efficiency	72.31%	73.09%
- Pump Discharge Pressure	46.189 P	45.684 P

Main Feedwater Pump	MFWP A	MFWP B
- Pump Efficiency	69.41%	70.08%
- %DP From FWP Discharge to SG Out	15.08%	14.81%
	49.73%	**50.27%**

Additional Conditions

ΔH Between HTR#3 Outlet & D/A	0.46 ΔH
ΔH Between FWP & HTR#5A Inlet	0.51 ΔH
ΔH Between FWP & HTR#5B Inlet	0.51 ΔH
ΔH Between HTR#7 & SG-A Inlet	-0.45 ΔH
ΔH Between HTR#7 & SG-B Inlet	-0.14 ΔH
ΔH Between D/A Outlet & BFWP A Inlet	0.23 ΔH
ΔH Between D/A Outlet & BFWP B Inlet	0.18 ΔH
FW ΔP from HTR#7 Outlet to SG Inlet	2.18%
ΔP Between LP TBN Bowl & LP-A Inlet	0.00%
ΔP Between LP TBN Bowl & LP-B Inlet	0.01%
ΔP Between LP TBN Bowl & LP-C Inlet	-0.01%

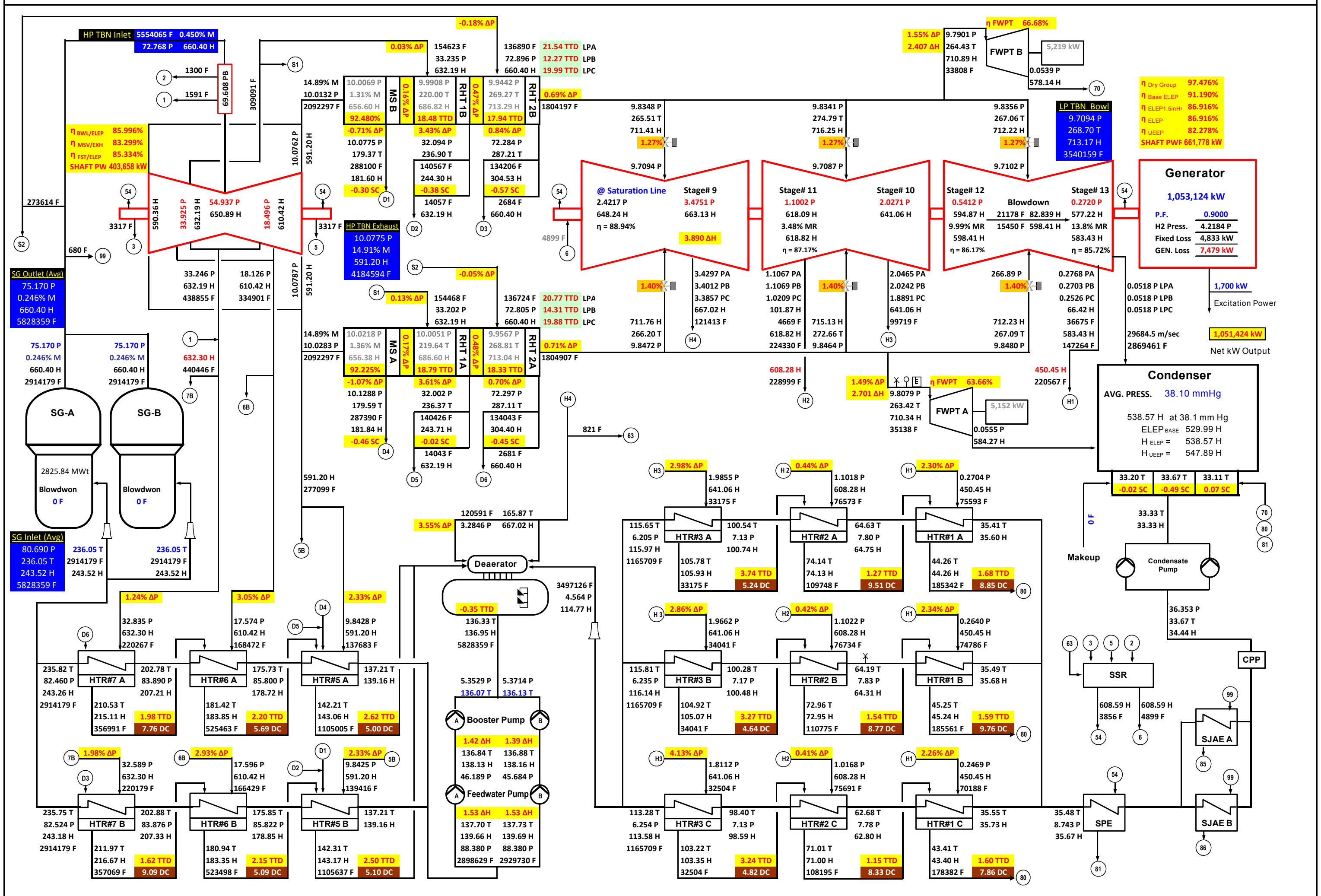
Flange Pressure Drop of LP TBN Stage Extraction for Each D/A & HTR#3,2,1

%DP from LP TBN A Stage#9 to D/A	1.30%
%DP from LP TBN B Stage#9 to D/A	2.12%
%DP from LP TBN C Stage#9 to D/A	2.57%
%DP from LP TBN A Stage#10 to HTR#3A Ext	-0.95%
%DP from LP TBN B Stage#10 to HTR#3B Ext	0.15%
%DP from LP TBN C Stage#10 to HTR#3C Ext	6.81%
%DP from LP TBN A Stage#11 to HTR#2A Ext	-0.59%
%DP from LP TBN B Stage#11 to HTR#2B Ext	-0.61%
%DP from LP TBN C Stage#11 to HTR#2C Ext	7.21%
%DP from LP TBN A Stage#13 to HTR#1A Ext	-1.75%
%DP from LP TBN B Stage#13 to HTR#1B Ext	0.62%
%DP from LP TBN C Stage#13 to HTR#1C Ext	7.13%

Enesco
Energy System Corporation

YONG-GWANG UNIT 4 TEST CYCLE HEAT BALANCE AT RATED CONDITIONS

한국수력원자력주식회사
KOREA HYDRO & NUCLEAR POWER CO., LTD



VALVE BEST POINT NET HEAT RATE = $\frac{5828359F \times (660.4H - 243.52H) + 0F \times (307.23H - 243.52H)}{1{,}053{,}124\ kW}$ = 2307.21 kcal/kwh

TEST POINT 100% RTO1 (TR01)

ON 1967 ASME STEAM TABLES
M - FLOW - kg/hr
P - PRESSURE AT - kg/cm2
H - ENTHALPY - kcal/kg
T - TEMPERATURE - C DEGREES

1054645 KW 38.1 MM HG ABS. 0.2 PCT MU
TC6F 43.0 IN. LSB 1800 RPM
72.768 ATMA 660.40 KCAL/KG 2 STAGE REHEAT
GEN 1219600 KVA 0.90 PF
MKS UNITS

Program by DOH-HOON, KOO

Attachment 5

OPTION	Run with Target MWt
	THB Conversion

Turbine Inlet Throttle Steam Flow 5,554,065 F

Reactor MW Thermal 2,825.84 MWt

0.000 MWt Diff

Turbine Inlet Steam Conditions

	SG A	SG B
Steam Generator Outlet Steam Pressure	75.170 P	75.170 P
Steam Generator Outlet Steam Moisture	0.2463%	0.2463%
Steam Generator Blowdown Flow	0.0 F	0.0 F

LP A Turibne Exhaust Pressure	38.10 mmHg
LP B Turibne Exhaust Pressure	38.10 mmHg
LP C Turibne Exhaust Pressure	38.10 mmHg
Generator Power Factor	0.9000

Current Throttle Flow Ratio 0.9605

Turbine Section Efficiency Level

HP Turbine Section	**2.1812%**
LP Turbine Section	**2.3065%**

Steam Turbine Stage Flow and Pressure

	Stgae Flow	Stage Press	Ethalpy
HP Turbine STG# 1	5,551,174	54.937 P	650.89 H
HP Turbine STG# 3	4,803,227	33.925 P	632.19 H
HP Turbine STG# 5	4,468,327	18.496 P	610.42 H
LP Turbine Bowl	3,540,159	9.7094 P	713.17 H
LP Turbine STG# 9	3,418,746	3.4751 P	663.13 H
LP Turbine STG# 10	3,319,027	2.0271 P	641.06 H
LP Turbine STG# 11	3,090,028	1.1002 P	618.82 H
LP Turbine STG# 12	3,053,400	0.5412 P	598.41 H
LP Turbine STG# 13	2,869,461	0.2720 P	583.43 H

Steam Turbine Extraction Stage Shell K-Factor

	Deisgn_K	Test_K	Difference
HP Turbine STG# 1	**140,098**	140,098	0
HP Turbine STG# 3	**193,158**	193,158	0
HP Turbine STG# 5	**321,771**	321,771	0
LP Turbine Bowl	**573,588**	573,588	0
LP Turbine STG# 9	**1,383,490**	1,383,490	0
LP Turbine STG# 10	**2,188,779**	2,188,779	0
LP Turbine STG# 11	**3,627,253**	3,627,253	0
LP Turbine STG# 12	**7,036,467**	7,036,467	0
LP Turbine STG# 13	**12,778,234**	12,778,233	0

CV & Packing Leakoff Flow C-Factor

Total Control Valve Leakoff Flow @VWO	0.05% MS Flow
N1LP Packing Leakoff Flow C-Factor	427.421
N2LP Packing Leakoff Flow C-Factor	427.421
LP TBN Seal Steam Supply Flow	4899 F
TBN Shaft End Packing to SPE Flow	3856 F

Stage to Flange Pressure Drop 2.00%

Steam Turbine Design Specification

VWO Throttle Steam Flow	5782335 F
VWO Throttle Steam Pressure	72.768 P
VWO Throttle Steam Enthalpy	660.40 H
VWO HP TBN Exhaust Pressure	9.962 P
VWO HP TBN Expansion Slope(ΔH/ΔS)	2632.894
TFR at First Admission Point	0.850
LP Turbine Exhaust Pressure	0.0518 P

Generator Design Specification

Maximum Generator Capacity in KVA	1219600 KVA
Rated Hydrogen Pressure	4.22 P

	Assumed	Difference
HP Exhaust Steam Flow	4184594 F	0 F
Generator Loss	7,479 kW	0 kW
Enthalpy of Extraction Steam to HTR#7	632.30 H	0.00 H
Enthalpy of Extraction Steam to HTR#2	608.28 H	0.00 H
Enthalpy of Extraction Steam to HTR#1	450.45 H	0.00 H

% Deviation from Design Excitation Power **48.02%**

Cycle Steam Pressure Drop

Steam Generator Outlet to MSV Inlet	**3.20%**
MSV Pressure Drop	**0.47%**
MSR Outlet to Combined Intercept	**1.10%**

FWPT Performance Specification

	FWPT A	FWPT B
FWPT Efficiency	**63.66%**	**66.68%**
FWPT Exhaust Steam Pressure Drop	**6.72%**	**3.84%**
FWPT Driving Steam Pressure Drop	**1.49%**	**1.55%**
FWPT Driving Steam Hot Extraction Loss	**2.70 ΔH**	**2.41 ΔH**

Heater Performance Specification

	Ext Stm ΔP	TTD	DC	FW ΔP
Heater #7A	**1.24%**	**1.98 ℃**	**7.76 ℃**	**1.71% ΔP**
Heater #7B	**1.98%**	**1.62 ℃**	**9.09 ℃**	**1.61% ΔP**
Heater #6A	**3.05%**	**2.20 ℃**	**5.69 ℃**	**2.23% ΔP**
Heater #6B	**2.93%**	**2.15 ℃**	**5.09 ℃**	**2.27% ΔP**
Heater #5A	**2.33%**	**2.62 ℃**	**5.00 ℃**	**2.92% ΔP**
Heater #5B	**2.33%**	**2.50 ℃**	**5.10 ℃**	**2.89% ΔP**
Dearator	**3.55%**	**-0.35 ℃**	**3.89 ΔH Hot Extraction**	
Heater #3A	**2.98%**	**3.74 ℃**	**5.24 ℃**	**12.92% ΔP**
Heater #3B	**2.86%**	**3.27 ℃**	**4.64 ℃**	**13.05% ΔP**
Heater #3C	**4.13%**	**3.24 ℃**	**4.82 ℃**	**12.32% ΔP**
Heater #2A	**0.44%**	**1.27 ℃**	**9.51 ℃**	**8.67% ΔP**
Heater #2B	**0.42%**	**1.54 ℃**	**8.77 ℃**	**8.39% ΔP**
Heater #2C	**0.41%**	**1.15 ℃**	**8.33 ℃**	**8.33% ΔP**
Heater #1A	**2.30%**	**1.68 ℃**	**8.85 ℃**	**10.77% ΔP**
Heater #1B	**2.34%**	**1.59 ℃**	**9.76 ℃**	**10.47% ΔP**
Heater #1C	**2.26%**	**1.60 ℃**	**7.86 ℃**	**11.00% ΔP**

Condenser Performance Specification

	Condenser A	Condenser B	Condenser C
Condenser Subcooling	**-0.02 ℃**	**-0.49 ℃**	**0.07 ℃**

MSR Performance Specification

	MSR A	MSR B
PD from HP Exhaust to MS Inlet	**0.50%**	**0.63%**
2nd Reheater Heating Steam Pressure Drop	**-0.05%**	**-0.18%**
1st Reheater Heating Steam Pressure Drop	**0.13%**	**0.03%**
Shell PD through 1st Reheater	**0.17%**	**0.16%**
Shell PD through 2nd Reheater	**0.48%**	**0.47%**
Total Cycle Steam Pressure Drop	**0.71%**	**0.69%**
Through Combined Intercept Valve - LPA	**1.40%**	**1.27%**
Through Combined Intercept Valve - LPB	**1.40%**	**1.27%**
Through Combined Intercept Valve - LPC	**1.40%**	**1.27%**

Moisture Separator Effectiveness	**92.22%**	**92.48%**
1st Reheater TTD	**16.74 ℃**	**16.89 ℃**
2nd Reheater TTD to LP A	**20.91 ℃**	**21.70 ℃**
2nd Reheater TTD to LP B	**14.44 ℃**	**12.42 ℃**
2nd Reheater TTD to LP C	**20.02 ℃**	**20.15 ℃**
MS Drain Tank Subcooling	**-0.46 ℃**	**-0.30 ℃**
1st Reheater Drain Tank Subcooling	**-0.02 ℃**	**-0.38 ℃**
2nd Reheater Drain Tank Subcooling	**-0.45 ℃**	**-0.57 ℃**
1st Reheater Excess Steam	**10.00%**	**10.00%**
2nd Reheater Excess Steam	**2.00%**	**2.00%**
Tube PD through 1st Reheater	**3.61%**	**3.43%**
Tube PD through 2nd Reheater	**0.70%**	**0.84%**

Pump Performance Specification

SPE H Rise

H Rise from Condenser Hotwell to HTR#1A	**2.27 ΔH**
H Rise from Condenser Hotwell to HTR#1B	**2.34 ΔH**
H Rise from Condenser Hotwell to HTR#1C	**2.40 ΔH**

Condensate Extraction Pump

- Pump Enthlpy Rise	**1.11 H**
- Pump Discharge Pressure	**36.353 P**
- %DP From COP Dischare to HTR#1	**75.95%**
- %DP From HTR#1 Inlet to D/A Inlet	**47.80%**

Feedwater Booster Pump	**FWBP A**	**FWBP B**
- Pump Suction Head from D/A	**2.07 ΔP**	**2.09 ΔP**
- Pump Efficiency	**72.31%**	**73.09%**
- Pump Discharge Pressure	**46.189 P**	**45.684 P**

Main Feedwater Pump	**MFWP A**	**MFWP B**
- Pump Efficiency	**69.41%**	**70.08%**
- %DP From FWP Discharge to SG Out	**14.95%**	
	49.73%	**50.27%**

Additional Conditions

ΔH Between HTR#3 Outlet & D/A	**0.462 ΔH**
ΔH Between FWP & HTR#5 Inlet	**0.513 ΔH**
ΔH Between HTR#7 & SG Inlet	**-0.292 ΔH**
ΔH Between D/A Outlet & BFWP A Inlet	**0.234 ΔH**
ΔH Between D/A Outlet & BFWP B Inlet	**0.175 ΔH**
FW ΔP from HTR#7 Outlet to SG Inlet	**2.185%**
ΔP Between LP TBN Bowl & LP-A Inlet	**0.000%**
ΔP Between LP TBN Bowl & LP-B Inlet	**0.008%**
ΔP Between LP TBN Bowl & LP-C Inlet	**-0.008%**

Flange Press. Drop of LP TBN Stage Extraction for Each D/A & HTR#3,2,1

%DP from LPA 9th Stage to D/A Extraction Nozzlle	**1.30% ΔP**
%DP from LPB 9th Stage to D/A Extraction Nozzlle	**2.12% ΔP**
%DP from LPC 9th Stage to D/A Extraction Nozzlle	**2.57% ΔP**
%DP from LPA 10th Stage to HTR#3A Extraction Nozzle	**-0.95% ΔP**
%DP from LPB 10th Stage to HTR#3B Extraction Nozzle	**0.15% ΔP**
%DP from LPC 10th Stage to HTR#3C Extraction Nozzle	**6.81% ΔP**
%DP from LPA 11th Stage to HTR#2A Extraction Nozzle	**-0.59% ΔP**
%DP from LPB 11th Stage to HTR#2B Extraction Nozzle	**-0.61% ΔP**
%DP from LPC 11th Stage to HTR#2C Extraction Nozzle	**7.21% ΔP**
%DP from LPA 13th Stage to HTR#1A Extraction Nozzle	**-1.75% ΔP**
%DP from LPB 13th Stage to HTR#1B Extraction Nozzle	**0.62% ΔP**
%DP from LPC 13th Stage to HTR#1C Extraction Nozzle	**7.13% ΔP**

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