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Thermal Performance Monitoring and Optimization in Nuclear Power Plants

Experience and Lessons Learned

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International Atomic Energy Agency

THERMAL PERFORMANCE
MONITORING AND OPTIMIZATION
IN NUCLEAR POWER PLANTS

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The Agency's Statute was approved on 23 October 1956 by the Conference on the Statute of the IAEA held at United Nations Headquarters, New York; it entered into force on 29 July 1957. The Headquarters of the Agency are situated in Vienna. Its principal objective is "to accelerate and enlarge the contribution of atomic energy to peace, health and prosperity throughout the world".

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EXPERIENCE AND LESSONS LEARNED

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FOREWORD

Thermal performance monitoring and optimization are vital attributes of efficient and reliable nuclear power plant operation. This publication explores and provides information on various aspects of thermal performance management in nuclear power plant operation and on good practices in this area. The publication highlights the importance of having an appropriate thermal performance programme, especially in the context of improving plant economic competitiveness. Good practices, case studies, challenges and lessons learned in thermal performance monitoring and optimization in nuclear power plants to enhance efficiency and reliability are also discussed.

The publication is aimed at Member State representatives involved in the thermal performance engineering or operation of nuclear power plants. These might include individuals engaged in engineering of heat balance optimization and heat reject optimization, including the development of balance of plant and use of performance curves on the basis of thermodynamic models. Suppliers of tools applicable to thermal performance testing and monitoring in nuclear power plants and members of academia who focus on thermal performance management of nuclear power plants may also find value in the content.

The IAEA expresses its appreciation to the relevant Member States for their valuable contributions and to the individuals who provided data and shared their experience on the subject. The IAEA officer responsible for this publication was E. Bradley of the Division of Nuclear Power.

CONTENTS

1. INTRODUCTION

1.1. BACKGROUND

Economic competitiveness with other energy sources as well as high reliability and performance of the balance of plant (BOP) systems are important goals of nuclear power plant (NPP) operation. Thermal performance is a key indicator for measuring BOP system efficiency, reliability and maintenance quality. Rising operating costs and increased competition from other plants have increased the need to improve thermal performance for efficient electricity generation. The integration of thermal performance assessment capability into modernization of operating NPPs and new NPPs designs is critical to ensuring the optimal efficiency in plant operation.

Thermal performance is the overall evaluation of NPP electrical production with respect to its energy consumption expressed in heat rate (kJ/kWh). An improved heat rate will result in more electricity being provided to the grid and can result in increased revenue from NPP operation.

The evaluation objectives of thermal performance include assessment of the steam conversion system (main steam/main feedwater system), flow capacity and efficiency of the turbine, condenser, moisture separator reheater (MSR) systems, heater regenerative system and pumps (e.g., circulating pumps, feed pumps, condensate pumps), among others.

Thermal performance of the NPPs will naturally decrease due to the heat loss and the aging of the BOP system such as turbine, MSR, feedwater system and condenser unless thermal performance is continuously monitored and improved under a structured programme to identify issues and review the design for thermal efficiency.

The basis of a thermal performance improvement programme is to complete periodic field performance diagnostics which support:

- Improved plant efficiency;
- Improved plant electrical output – additional revenue;
- Early performance degradation recovery;
- Early identification of reliability issues;
- Short payback period;
- Maximize use of original capital investment;
- State of the art technology – increased asset value.

The efforts behind improving thermal performance require a broad understanding of NPP design, operation, maintenance, ambient conditions, thermal sciences, etc. To be successful, a holistic view ought to be taken to ensure results are both cost-effective and do not create problems elsewhere in the plant.

1.2. OBJECTIVE

The objective of the publication is to provide the best practices and practical experiences related to the monitoring and optimization of plant thermal performance to enhance efficiency and reliability. All the methods available including those described in this publication are to be considered estimates.

The publication describes the essential elements of a thermal performance programme, providing guidelines on the design of the BOP systems for new build NPPs and improvements to an existing programme for operating NPPs.

It focuses on the roles and responsibilities of a thermal performance engineers (TPEs) and their interface with other site organizations.

1.3. SCOPE

The scope of the publication is:

— Thermal system performance maintenance and improvement plans
— Thermal system test and monitoring procedures:
  - High accuracy testing that is done infrequently.
  - Routine testing done with a frequency that ranges from monthly to quarterly.
  - On-line monitoring system to provide plant operations and maintenance personnel with immediate feedback in real time on plant performance.

Major parameters and systems to be monitored for thermal performance include 1) steam turbine monitoring models, steam enthalpy, turbine flow capacity, 2) error heat balance method, characteristic flow area, variable condition calculation, 3) condenser back pressure, 4) accurate thermal performance history files, 5) MSR system etc.

The publication looks at core tasks the TPE performs, including development and use of performance curves and thermodynamic models, and at his or her role in system monitoring, work management, and modification support. It details the importance of properly communicating plant performance both internally to plant management and externally to regulatory and government agencies, as well as programme assessment. This publication is intended for NPPs owner operators, engineers and specialists to maintain and improve thermal system efficiency.

1.4. STRUCTURE

This publication is comprised of 7 Sections containing information related to thermal performance monitoring and optimization in NPPs operation:

— Section 2 introduces the thermal performance programme its applicability and programmatic aspects and discusses safety and performance culture.
— Section 3 presents key performance indicators, monitoring timing, input data, cycle isolation and analysis methods for monitoring thermal performance.
— Section 4 is devoted to key component performance namely reactor and steam generator measurement and performance and uncertainties determination.
— Section 5 is focusing on key component performance on the turbine cycle side.
— Section 6 is describing recovery or improvement of electrical power output.
— Section 7 is concludes by summarizing all methods.

2. THERMAL PERFORMANCE PROGRAMME

The programme overview will briefly describe the various aspects of a typical thermal performance programme at a NPP. This section will provide a summary of the objectives, methodology, processes and unique considerations of a thermal performance programme.

2.1. DISCUSSION ON SAFETY AND PERFORMANCE CULTURE

The ability to safely produce electricity is at the centre of the TPEs work description. It is often difficult at NPP to devote the necessary resources to monitoring and improving plant efficiency in the wake of the overriding concern for safety. The focus of the TPE is to maximize plant efficiency while maintaining proper consideration for safety. For instance, the performance engineer ensures the instrumentation and calculation of core thermal power are accurate and remains cognizant of issues that can result in non-conservative calculations. However, it is also important to operate the NPP at the maximum electrical output which would include providing support to maintain full reactor power. Plant management is required to recognize the challenge and provide the necessary support, oversight and guidance to ensure the TPE help the NPP achieve safe and efficient plant operation.

2.2. OBJECTIVE OF THERMAL PERFORMANCE PROGRAMME

2.2.1. Thermal efficiency optimization

2.2.1.1. Increasing MWe output

Thermal efficiency is the ratio of the amount of energy available to generate electricity to the amount of electricity generated. Thus, any improvement in the amount of electricity generated at a given reactor power is an improvement in plant efficiency. Increasing plant electrical generation is the focus of a thermal performance programme. While in some sense this is the basic purpose of all site personnel, it is uniquely the direct focus of the thermal performance programme. Safely and cost effectively increasing the electrical output of the NPP will add value to the entire NPP by reducing the overall cost of producing electricity. This focus will consist of identifying generation losses and determining how these losses can be recovered. Due to changing conditions outside of the plant operator's control, identifying reduced efficiency is a detailed and complicated endeavour. The target generation is influenced by various physical conditions such as weather, cooling body (river, lake or ocean) or operational restrictions. Therefore, the discussion of increasing electrical output is based on a standard set of reference conditions. Hence the need to provide a means of determining the expected generation to which actual generation can be compared.

2.2.1.2. Plant modification to improve power output

In addition to identifying and correcting problems causing reduced generation, plant value can be improved by modifying the plant design, operation or configuration to increase the generation of the plant. These changes may or may not necessarily result in improvements to plant efficiency; but result in an overall improvement in the cost to produce a given amount of electricity. For example, a modification which increases the thermal power delivered by the reactor may not improve the thermal efficiency but will increase the plant's electrical generation. The one time and recurring costs (e.g. increased maintenance costs) of these modifications are compared to the benefit of the new generation to justify the change. These changes may take the form of power uprates, equipment improvements, operational changes or improved maintenance practices.

2.2.2. Reactor power measurement accuracy

As stated above the plant efficiency is a ratio of reactor power to electrical generation, also called heat rate (HR). Therefore, it is crucial for the TPE to have and understanding of both the numerator (reactor power) and the denominator (electrical generation) of the ratio. TPEs are in a unique position in that they are aware of all aspects of the plant with respect to electrical production. Other plant staff focus on specific areas such as reactivity control, equipment maintenance or system health. The TPE is tasked with understanding all the influences on plant generation and how the individual plant systems work together in the whole plant. In a sense the thermal performance 'system' includes the integration of nearly all plant systems. This will start at the reactor and end at the cooling tower or body of cooling water. Since the BOP systems are affected by the amount of energy being produced by the reactor, those BOP systems provide information about the reactor. It is because of this relationship that the TPE can monitor reactor power in a way that the typical reactor engineer cannot. Thus, a good understanding of the core thermal power calculation is essential for the TPE.

2.3. APPLICABILITY OF THERMAL PERFORMANCE

2.3.1. Normal operation

Various plant circumstances can influence the methodology employed by the TPE. These would include normal operation as defined by the basic plant design, conditions where operational changes are made for long term operation to maintain safety limits, changes to plant design such as turbine replacement or operational changes due to influences outside the control of the plant such as grid demand changes. These conditions are discussed in this section to provide an understanding of the issues to be considered by the performance engineer. Normal operation is based on the as installed plant. This may or may not coincide with the initial design of the plant. There may be limitations placed in the plant operation such as transformer or grid capability. Sometimes a plant will be limited due to equipment degradation such as feedwater heater limitations or cooling system capability. Even if there are such limitations a baseline needs to be established based on the as built or 'as is' condition of the plant. This based line will be used to provide target plant parameters which the performance engineer can compare against actual plant measurements. Periodic evaluation of the plant in the normal operating condition will provide a way to identify and correct any deviation from optimum performance.

2.3.2. Long term operation

Consideration of plant operation over the long term is fundamental aspect of the TPE's responsibilities. While overall plant efficiency is the primary task of the TPE, consideration for the long-term reliability and performance of the plant is also typically included. Feedwater heater operation is an example. While the heater performance may be improved by lowering the heater level, such operation may ultimately cause drain cooler damage thus requiring the heater to be replaced or result in unreliable overall plant operation.

Some plants have reduced primary temperature in order to prevent damage to the steam generators (SGs). This reduced temperature results in lower steam pressure and thus reduced overall plant efficiency. Therefore, when monitoring a plant over the long-term, such consideration of operational realities is typically a factor in key performance parameters.

2.3.3. New design verification test

When designing a plant, thermal performance monitoring is not always adequately implemented. Many plants have very limited instrumentation which makes monitoring and troubleshooting thermal performance issues more difficult. Attention is advised to be given to what parameters are measured as well as the location of those measurements.

The following instrument suggestions are provided for consideration:

- Measure heat exchanger outlet temperatures far enough downstream of the tube sheet to allow adequate mixing to prevent erroneous temperature measurements due to stratification. Outlet temperatures are the most susceptible to this problem.
- Measure throttle pressure near the turbine stop valves.
- Measure cooling water flow to the condenser (bulk and individual water box). At many plants there is not a measurement of this flow, thus hampering the ability to determine condenser performance.
- Install basket tips for condenser pressure measurements. Additionally, providing proper tubing slope helps prevent erroneous readings.
- Measure heater drain flow.
- Measure heating steam supply flow to reheaters for each stage in the case of two stage reheaters.
- Measure moisture separator drain flow.
- Provide density compensation for steam flow measurements.
- Ultrasonic feedwater flow measurements to be provided as well as flow measurements based on differential pressure.
- Place temperature instruments downstream of cycle isolation valves. These instruments are typically monitored by the plant computer.

2.3.4. Retrofit

Consulting the TPE early in the process of any major modification such as turbine retrofit or major component replacement is a good practice. Often, the TPE will be the most knowledgeable person regarding current plant operation and operational parameters that will be used as input for a design change. Additionally, the acceptance test following such plant modifications will require input from the TPE. This testing may require specific plant modifications, including:

- Installation of pressure taps downstream of control valves;
- Validation or installation of pressure taps for extraction points;
- Installation of tracer injection and sample points for MSRs;
- Installation of basket tips for low pressure hood pressure measurement;
- Installation of pressure transducers to measure feed pump turbine steam flow;
- Ability to measure feedwater pump seal flow;
- Installation of flow measurement devices on heater drains;
- Pressure measurements at heater shells;
- Reheater drain flow measurement;
- Moisture separator drain flow measurement;
- Feed pump turbine exhaust pressure measurement;
- Hot well temperature measurement;

— Ability to temporary isolate leaking turbine bypass valves.

Another consideration is when a plant is undertaking multiple major changes during the turbine outages or when changes occur between the contract (design) and the installed heat balance. Scrutiny needs to be applied to the final heat balance to identify any changes that resulted from the design process especially where it interfaces with the other components that may be changed. It is very important that the current plant conditions are provided to the turbine vendor for the purposes of designing the new turbine. This is especially true if just the low pressure (LP), interim pressure or high pressure (HP) turbines are being replaced.

2.3.5. Load following operation

As the power industry changes there are scenarios when NPPs will be forced to reduce load to accommodate grid control requirements such as an increase in wind, solar or hydroelectricity at certain times. Typically, the NPP will reduce power to a fixed load (e.g. 60%) depending on the particular plant design and stability restrictions. Areas to be considered are primarily centred around the ability to maintain stable safe operation at the lower load for an extended period. If there is a requirement to change load often then other considerations are recommended to be considered, such as transient considerations regarding reactor control and stability of the secondary systems.

2.4. PROGRAMMATIC CONSIDERATIONS

This section will provide an overview of items that are advised to be included in a thermal performance programme. Starting from the goals and expectations and how to achieve the goals.

2.4.1. Policy (goal expectation of management level)

There are three types of goals that are considered for the thermal performance programme:

- Programmatic goals;
- Plant efficiency, in some instances, for base load, plant efficiency will be replaced by actual plant generation as compared to expected generation.
- Production goals.

The programmatic goals look at the procedures and practices of the thermal performance programme. Does the programme meet industry standards? Is the programme effective in the identification and reporting of thermal performance issues?

The plant efficiency and production goals have essentially two considerations. Firstly, is the plant generation at or near the baseline value corrected for expected conditions based on environmental conditions; namely circulating water inlet temperature or atmospheric conditions? Secondly, how does the plant production compare to its capacity factor (CF)? Capacity factor is mostly influenced by reliability and outage duration.

The goals associated with CF are determined in conjunction with business planning and are most affected by plant outages. The TPE typically has the necessary information to project performance with respect to the established goals as well as providing significant input to the setting of the goals.

An important element of the thermal performance programme is a good understanding of the ability to measure and determine a baseline for monitoring. Proper understanding of the plant design and operational limitations is necessary for establishing realistic goals. The plant baseline values may have come from a thermal kit or a design document that never actually represented the as-built condition of the plant. If a good baseline value has not been rigorously determined it will be impossible to set a realistic plant efficiency goal.

Since the plant generation is affected by environmental conditions it is necessary to review the plant efficiency against an expected value for the given conditions. The conditions can be atmospheric parameters such as wet-bulb temperature or humidity if a cooling tower is used or cooling water inlet temperature if the plant is a once-through design.

The TPEs responsibility with respect to plant management is to accurately report either programmatic or thermal performance issues which would affect the ability of the plant to generate rated electricity. In addition to reporting problems, possible solutions ought to be provided with adequate cost benefit analysis to allow management to make informed decisions regarding thermal performance issues. Often the TPE is required to report to senior plant management regarding possible projects to improve plant generation or correct plant efficiency problems. The TPE may have input to the key indicators which are associated with all plant goals therefore, frequent communication will aid management understanding of these indicators. Topics for the meetings with plant management may include:

- Ability and accuracy of the indicators to represent actual performance.
- How much time it took to identify and recover lost megawatts.
- Negative or positive trends from previous years, and reasons for change.
- Validity of baseline and reference values.
- Performance improvements for equipment and the programme.
- Industry comparisons and advancements.

One aspect of setting plant goals related to lost generation is the overall uncertainty of the measurement and calculation process. The best baseline process may not use design documents to establish plant generation goals. Often the design thermal kit provided by the vendor may not represent the actual plant configuration or operation. Sometimes historical plant data or results of a code-based turbine test are used as a basis for establishing expected plant output which ought to be corrected for environmental conditions. Many plants will set threshold values for lost generation to encompass the random uncertainty of the process. A curve is generated with an uncertainty band around the expected generation for a given set of atmospheric or condenser inlet conditions.

2.4.2. Programme key performance indicator, metrics

Programme key metrics can be summed up in an evaluation of the overall programme health which considers all the aspects of the programme and rates them on an objective set of criteria. These evaluations can be divided into process issues and component issues as shown in the Table 1 and Table 2 below.

TABLE 1. PROCESS AND COMPONENT ISSUES EVALUATION [1]

Programme process health colour criteria	Comments
1a. Resources – staffing ☒ Green: Required staffing meets actual ☐ White: Required staff with backup in training ☐ Yellow: Understaffed with active requisition ☐ Red: Understaffed with no plan	1 Technical specialist 1 Engineer 2 Analyst
1b. Resources - training/qualification ☐ Green: Fully trained/qualified backup programme owner ☒ White: Fully trained owner, backup named ☐ Yellow: Training is not current, training in progress ☐ Red: No trained staff	Backup is in the process of being trained.
2a. Procedures – adequacy ☒ Green: Best practice process ☐ White: Current and meets all requirements ☐ Yellow: More than 3 outstanding changes ☐ Red: Missing for critical or complex task	TNC thermal performance procedures for tracking plant thermal performance identified as strength by Exelon PEER Assessment and other utilities have used our procedures as a basis for their programmes.
2b. Procedures – compliance ☒ Green: No valid notifications on procedure noncompliance. ☐ White: Positive trend. in noncompliance notifications ☐ Yellow: Negative trend in noncompliance notifications ☐ Red: No plan for improving negative trend	None for this period
3a. Implementation - assessments Internal ☒ Green: QA satisfactory assessments, no open issues ☐ White: Few QA findings of low consequence ☐ Yellow: Unresolved QA issues or findings ☐ Red: Numerous unresolved issues and findings	None for this period
3b. Implementation - assessments external (period) ☒ Green: No NRC/INPO; satisfactory ☐ White: Minor NRC/INPO findings or observations ☐ Yellow: Significant NRC/INPO findings or observations ☐ Red: NRC violation(s)	None for this period
3c. Implementation- Self-assessments ☒ Green: Strengths Identified or minor improvement areas ☐ White: Meets requirements ☐ Yellow: Few findings with low risk ☐ Red: Numerous unresolved issues and findings	No assessments for this period
4. Programme monitoring trends ☒Green: No trends in alert or action ☐ White: No trends in action ☐ Yellow: < 2 trends in action ☐ Red: >2 trends in action	All performance trends are up to date and issues are identified and tracked via notifications. ‘alerts’ ‘actions’ are not applicable to the thermal performance programme. The thermal performance indicator is at or above the goal of 99.7%. There are no units that have unidentified losses greater than 1 MWe on a regular basis. This would constitute an equivalent to an alert or action level for the thermal performance programme.

TABLE 1. PROCESS AND COMPONENT ISSUES EVALUATION (cont.) [1]

Programme process health colour criteria	**Comments**
5. Outstanding programme corrective action items (NUCR) ☒ Green: None older than 120 days ☐ White: Average age less than 120 days ☐ Yellow: Average age greater than 120 days ☐ Red: Average age greater than 120 days for more than one health report cycle	Currently no outstanding corrective action programme items older than 120 days.
6. Open programme improvement items (NUTS) ☒ Green: None ☐ White: = 1 ☐ Yellow: </= 3 ☐ Red: >3	There are currently no programme improvement NUTS orders.

TABLE 2. PROCESS AND COMPONENT ISSUES EVALUATION [1]

Programme component health colour criteria	**Comments**
1. Forced Derates or transients ☐ Green: 0 Forced derates or transients ☐ White: 1 ☐ Yellow: 2 ☒ Red: $\geq$ 3	The TNC nuclear forced loss rate is 12.3% primarily due to equipment failure issues.
2. Operations and maintenance concerns ☒ Green: 0 Concerns ☐ White: 1 ☐ Yellow: 2 ☐ Red: $\geq$ 3	There are currently no operator concerns identified with the thermal performance programme
3Component Monitoring trends ☐ Green: 0 ☐ White: 1 ☐ Yellow: 2 ☐ Red: >2	
4. CM and PM assessment ☒ Green: 2 Open CMs ☐ White: 2 □ x <= 5 ☐ Yellow: 6 □ x <= 10 > 18 months ☐ Red: > 10 for longer than 18 months	Currently 2 open CM’s associated with components affecting plant thermal performance.
5. Long standing operating, design, or licensing basis issues ☒ Green: 0 issues ☐ White: 1 ☐ Yellow: 2 ☐ Red: > 2	Currently no long-standing issues affecting components related to thermal performance.
6. Material condition ☐ Green: 1 degraded item / leak ☒ White: 2 <= x <=5 ☐ Yellow: 5 ☐ Red: >5	3 items identified as degraded resulting in losses or reduced ability to monitor performance.

Once evaluated the indicators can be combined to establish overall plant health as shown below in Figure 1. This indicator provides management with a clear picture of the status of the thermal performance programme and along with the detail in Table 1 and Table 2 above where improvements need to be made.

Thermal performance programme health status

Programme owner: **MAXEENE MEGAWATT**

Jan 2017 - Jun 2017	Jun 2017 - Dec 2017	Jan 2018 - Jun 2018
WHITE/ IMPROVING	WHITE / IMPROVING	WHITE / IMPROVING

Projected yellow	Projected white	Projected green
N/A	N/A	12/2019

FIG. 1. Plant health programme layout

2.4.3. Organization roles and responsibility

In a programmatically efficient situation, the TPE is the established holder of the thermal performance programme and all plant organizations recognize this responsibility. All questions about unit thermal performance are directed to the TPE. It is important that the TPE is consulted for operating and design configuration decisions that may affect unit thermal performance. The TPE is also responsible to maintain the plant data required for generation reporting and measuring thermal efficiency.

Maintaining TPE awareness of the latest techniques in monitoring and trending, instrumentation and new component designs to improve plant efficiency is important to a successful thermal performance programme. This is best accomplished through participation in the various industry groups and reading of the latest literature associated with thermal performance. It is often the case that a new design or method is not intuitive and requires a good understand of the plant cycle to recognize the benefit. The following are suggested areas to consider:

- New technologies associated with a component;
- Common failure mechanisms and their causes;
- New or enhanced preventive maintenance techniques;
- Suitable replacement items (components, subcomponents, and parts) due to obsolescence;
- New regulatory requirements or industry operating guidelines;
- Modification experiences and lessons learned;
- Early warning of component degradation.

2.4.4. Interface of internal and external organization

Operations interface – Operations is ultimately responsible for the safe and reliable operation of the power plant and has first-hand knowledge of the systems important to thermal performance. The TPE is encouraged to be familiar with the operations staff and periodically (on a weekly basis as a minimum) visit the control room to check displays and field any questions the operators may have about thermal performance. The TPE’s relationship to the operations department is vital to the success of the thermal performance programme. The operators often are first to observe performance declines or questionable changes in parameters, and they are advised to consult the TPE immediately in those instances.

If an on-line thermal performance monitoring system is implemented, the operators may desire access to information from it and can work with the TPE to resolve plant efficiency problems.

Maintenance and outage interface – Maintenance and outage work to address thermal performance deficiencies is not always given the appropriate priority to ensure timely completion. Therefore, communication of well-documented cost/benefit information to the maintenance and outage management staff is necessary. A grading system for prioritizing thermal performance improvements and issues is defined, such that work orders and design changes are priority-coded according to economic/reliability value to help communicate the importance of the items. It is also important for the TPE to own any maintenance generated as a result of the thermal performance programme. This means following through on the effectiveness of the maintenance and determining the benefits received as a result of the maintenance. For example, if the TPE identifies a leaking valve they may want to be present when the valve is inspected to visually see the condition of the seat and disc.

Engineering interface – The TPE is either assigned from the system engineering or programmes engineering organization. The advantage of being in the system engineering organization facilitates a strong communication link with the system and component engineers for problem identification and resolution at the system/component level. In addition, it is necessary to develop a good interface between the TPE and the design engineering staff to ensure that design modifications are properly evaluated for thermal performance impact. The TPE also interacts with other engineering programmes such as the flow-accelerated corrosion, air-operated valve, and SG programmes to address known leakage paths and performance problems. The relationship between the TPE and the project engineering organization is important to ensure that any efficiency related projects are adequately evaluated for their effect on plant thermal performance. The TPE has the 'big picture' view of the NPP and can evaluate a project with respect to all the variables impacted. For example: NPP desired to install a SG blowdown recovery system. The design was almost completed when the TPE observed that the location of the heat exchanger cooling water return would result in reduced generation due to the temperature effects on the cycle. Additionally, the TPE is advised to be involved in major plant component replacement acceptance testing especially where increased generation is part of the guarantee.

Chemistry interface – The plant chemistry department maintains condensate/feedwater/circulating water chemistry, including dissolved oxygen levels, within appropriate limits for control of corrosion in the fluid system components. Many component efficiencies are affected by changes in plant chemistry due to build-up of corrosion or oxide layers. Circulating water systems including cooling towers and condensers can be greatly affected by changes in biocide control or chemistry causing biofouling or scaling on condenser tubes or cooling tower fill. Therefore, proper operation of these systems is important to maintaining the overall efficiency of the plant.

Business analysis/generation planning interface – It is important that this programme interface directly with the business analysis/generation planning organization to establish measurable and achievable annual performance goals. This interface will also ensure that critical parameters are consistently selected and used to report unit thermal performance. For programme monitoring and overall programme effectiveness, the TPE and the business analysis/generation planning representatives are recommended to meet with the plant management at least once a year to review the following items:

- Thermal performance indicator trends versus goals;
- Goal – setting for the following year, including thermal performance and generation goals;
- Programme effectiveness.

To complete the monitoring efforts and fully evaluate the effectiveness of the thermal performance program, the TPE and responsible plant personnel review the trends in comparison to the established goals. Comparisons to other indicators such as heat rates for other plants are necessary to be made. Important areas to review for both direct and indirect monitoring include the following:

- Accuracy of the indicators to represent actual performance;
- Elapsed time required to identify and recover lost megawatts;
- Negative or positive trends from previous years, and reasons for change;
- Validity of baseline and reference values;
- Performance improvements for equipment and the programme;
- Industry comparisons and advancements.

2.4.5. Communication and reporting

When reporting results, it is recommended to include the process for recovering identified lost generation. This kind of reporting ought to include identification of the loss, work responsibility, due date, estimated loss in MWe (or revenue) and any explanations required. The losses may be categorized as short term or long-term losses. See example of the thermal performance status report in Figure. 2.

Unit 1 - Thermal Performance Status				MW Accounting	Main Menu
Identified Short Term Losses	**Work Doc.**	**Responsibility**	**Due Date**	**Est. MWe**	**Explanation**
Throttle Steam Pressure MW Loss	3336900	Outage	5/25/2019	-0.85	
Top Final FW Temp MW Loss	3336901	Engineering	5/1/2018	-0.02	
MSR TTD MW Loss	3336902	Work Control	6/6/2018	0.06	
Condenser Efficiency MW Loss	3336903	Work Control	7/9/2018	-1.87	
Condenser Subcooling MW Loss	3336904	Outage	3/6/2019	-0.08	
Thermal Power MW Loss	3336905	Engineering	5/1/2018	-0.37	
Blowdown Flow MW Loss	3336906	Chemistry	6/6/2018	0.09	
Generator H2 Pressure MW Loss	3336907	Work Control	7/9/2018	0.02	
Generator H2 Purity MW Loss	3336908	Outage	3/6/2019	0.00	
Generator MVA MW Loss	3336909	Engineering	5/1/2018	-0.52	
MSR Pressure Drop MW Loss	3336910	Work Control	6/6/2018	-0.07	
FPT Flow MW Loss	3336911	Work Control	7/9/2018	-0.96	
Identified Long Term Losses	**Work Doc.**	**Responsibility**	**Due Date**	**Est. MWe**	**Explanation**
Turbine first Stage blockage	3340098	Outage	05/25/19	4.5	Retrofit scheduled for 2019
Comments					

FIG. 2. Thermal performance status report [1]

2.4.6. Peer review, program assessment

Peer reviews and assessments provide a means to verify the thermal performance programme meetings intended goals. An effective assessment needs to cover the following areas:

- Structure – what are the essential elements of the thermal performance programme.
- Ownership – is there clear ownership of the programme.
- Training – qualification of the TPE, plan for continuing training and succession planning.
- Design output basis (design heat balance) and regulatory basis (if applicable).
- Implementation – procedures for monitoring and reporting.
- Testing and inspection including capacity determination tests, acceptance testing and cycle isolation.
- Configuration management – of programme elements, models design basis.
- Performance monitoring and trending.
- Self-assessment and benchmarking.
- Use of industry operating experience.

2.4.6.1. Assessment plan

The assessment plan will include the areas of the programme to be assessed and how they will be assessed. The plan may include:

(a) The development of a specific set of questions to ask each responsible department.

(b) Scheduling of interviews of personnel from the Operations, Maintenance, Work Management, Chemistry, Finance, and Sr. Management.

Many sites will have a formal and informal self-assessment process with a procedure and qualification required for the assessment team leader. The assessor may need to use this process or at least part of this process when planning and implementing the self-assessment for their thermal performance programme. There are many variations of the self-assessment process, so it is recommended to determine how to combine the requirements of their plant with the information provided in this section.

2.4.6.2. The assessment of the programme scope may include the following elements

— Review historical plant data;
— Review current plant data;
— Review applicable plant documentation;
— Key component specifications;
— Uncertainty calculation;
— Baseline validation data;
— Acceptance testing;
— Review current thermal performance calculations;
— Review historical feed flow measurements;
— Review plant procedures;
— Review organization interfaces;
— Review secondary plant history (major BOP modifications and events);
— Review and validate the current plant thermodynamic computer model;
— Review the core thermal power calculation;
— Evaluate the current plant thermal performance programme including generation accounting;
— Long term planning for thermal performance improvements. Provide an evaluation of possible modifications to increase overall plant generation. This evaluation will consider all major BOP components.

2.4.6.3. Thermal performance program design

— Programme structure – Provides an assessment of the basic components of the programme and management oversight of the programme.
— Programme scope – Provides an assessment of the programme basis including industry documents (for example the EPRI Thermal Performance Handbook [2]) and the basis of what structures, systems and components are included or excluded from the programme.
— Design basis and regulatory basis – Provides an assessment of how the programme design and regulatory basis is included if applicable. It may include an assessment of

the baseline design data of the plant used to determine the design output of the plant. It may also assess the quality of the secondary model.
- Configuration control – Provides an assessment of the use of design documents as appended from time to time that establish the baseline values for the programme. It may also assess the contents and organization of the programme notebook if used.

2.4.6.4. Thermal performance program execution

- Implementation – Provides an assessment of how well the overall programme is being implemented by the TPE and if all aspects of the programme are being implemented per the programme procedure.
- Training and qualification – Provide an assessment of the training and qualifications of the TPE, use of continuing training and succession planning for the programme.
- Performance monitoring – Provides an assessment of the effectiveness of the trending and monitoring portion of the programme, how well the various software data management and analysis tools are being used, as well as the accuracy of thermal performance calculations.
- Programme ownership – Provides an assessment of how well the TPE owns the programme, manages the interdepartmental communications, and keeps the management chain of command informed of problems and issues.

2.4.6.5. Thermal performance programme results

- Management oversight of results – Provides assessment of how well management is providing leadership to the programme using performance indicators and observations. It also looks at the adequacy of internal and external reporting.

- Self-assessment and benchmarking – Provide an assessment of how well the programme is being reviewed against industry standards, the frequency of the assessments, industry experience, participation in industry owner and peer groups, and closure review of previously identified programme deficiencies.

2.4.6.6. Attributes of a self-assessment

The self-assessment is a less formal assessment led by the TPE at the site with limited management involvement. It may include additional multi-department involvement at the site level. A self-assessment normally consists of the following attributes:

- Informal assessment with a department leader as management sponsor;
- Lead by the TPE or their section leader;
- Short duration, two to three days;
- No outside peer on-site, may use peer help to develop a self-assessment plan;
- Limited plan, with limited review of interdepartmental interactions;
- Small team, may include operations;
- Limited final report.

2.4.6.7. Attributes of a peer-assessment

The peer assessment is a more formal and in-depth assessment involving industry peers and multi-department involvement. The assessment is likely to be conducted by a person from the plant or fleet with training in the performance of self-assessments. A peer assessment normally consists of the following attributes:

- Formal assessment with director or senior executive level management sponsor;
- Team leader to be fully trained full-time assessment leader;
- One to two-week duration;
- Use of industry peer from industry, fleet or contractor;
- Extensive communications review with multi-department interviews;
- Large team with multi-department involvement;
- Comprehensive final report.

The formal peer review assessment will provide a summary of the overall programme effectiveness. This overall summary may include an objective measurement of the following areas as shown in Table 3. This assessment can be displayed graphically as follows in Figure 3.

TABLE 3. PEER REVIEW ASSESSMENT OF PROGRAMME EFFECTIVENESS

Programme design	Programme structure	0.80
	Programme scope	0.75
	Programme design and regulatory bases	1.00
	Programme configuration control	0.75
Programme execution	Programme implementation	0.65
	Programme training and qualification	0.60
	Programme performance monitoring	1.00
	Programme ownership	0.83
Programme results	Programme management oversight	0.71
	Programme self-assessment and benchmarking	0.58

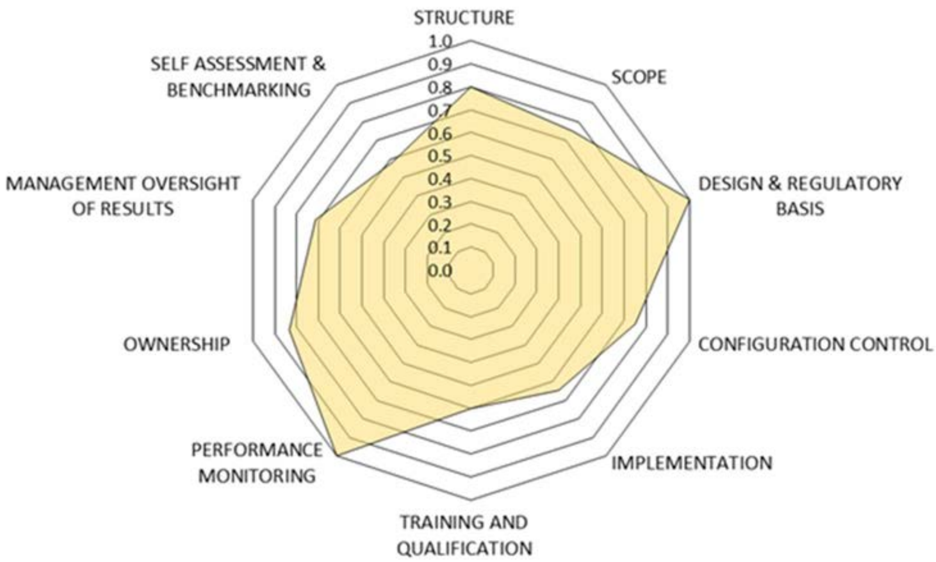

FIG. 3. The graphical peer review assessment [1]

The following is an example of an outline for a formal peer assessment report:

(a) Executive summary;

(b) Areas reviewed and associated methodology;
- Abbreviations;
- Inputs for assessment;
- Overall program evaluation results.

(c) Programmatic assessment;
- Programme strengths;
- Recommendations for program improvement;
  - Improve operations thermal performance indicator;
  - Add circulating water inlet temperature to operations report.
- Work control / maintenance coding;
- Improve operating experience process;
- Improve interface with operations;
- Establish a qualified backup programme owner;
- Cycle isolation monitoring improvement.

(d) Current plant thermal performance
- Performance summary;
- Lost megawatt accounting;
- Measurement of generation;
- Feedwater heaters (same general form for all components as applicable);
  - Design;
  - Performance summary;
  - LP heater performance;
  - HP heater performance;
  - Recent performance trends;
  - Planned FW heater maintenance activities;
  - Drain valve problems;
  - Recommendations.
- Condensers;
- Cooling towers;
- Steam turbines;
- Core thermal power;
- Blowdown system;
- Moisture separator reheaters.

(e) Summary of recommendations
- Programme recommendations;
- Feedwater heater recommendations;
- Condenser recommendations;
- Cooling tower recommendations;
- Steam turbine recommendations;
- Blowdown system recommendations;
- MSR recommendations;
- Core thermal power recommendations.

2.4.7. Procedure (implementation procedure)

In order to implement a thermal performance programme a procedure is required to define the goals, scope and processes necessary to generate electricity safely and efficiently. It is advised that the procedure covers the following areas:

- Purpose;
- Scope;
- Definitions;
- Baseline document and modelling;
- Measurable performance goals;
- Continuous monitoring and trending;
- Structured search and recovery methods;
- Clear lines of communication and reporting;
- Structured search and recovery methods.

2.4.8. Monitoring

2.4.8.1. Benchmarking (baselining)

Before comparisons and evaluations of plant performance can be made, standard definitions and baseline values for key performance indicators (KPIs) need to be established. Baseline values are necessary because of the variable operation of the unit compared to the steady-state design condition and are needed for corrections, comparisons, and troubleshooting. The baseline provides the optimum performance that the plant can achieve.

Baseline reference values are established for the performance of equipment important to overall cycle efficiency, parameters used to adjust for (or correct) equipment performance degradation, and correction factors used to adjust performance data back to baseline/reference ambient conditions.

The following design or baseline values are developed and documented by each plant using design calculations, verified deviations from design for actual operating conditions, baseline testing, and historical performance trends. The baseline may change when the unit is modified, requiring that a new baseline be established.

Sources for baseline information:

- Turbine vendor thermal or heat kit;
- Unit acceptance test;
- Performance test;
- Historical operating data;
- Post outage data, especially in the case of a major change (clean condenser, etc.).

The baseline is typically established at 100% power and reference ambient condenser cooling water temperature. The following is advised to be considered when establishing the baseline.

- Best achievable heat rate (kJ/kWh);
- Gross electrical output, MWe;
- Turbine backpressure and associated condenser cleanliness;

— Main steam pressure(s) (turbine throttle, impulse/first stage pressures);
— Feedwater temperature(s) and flow rates (final feedwater temperature, feedwater heater terminal temperature differences (TTDs), drain cooler approaches (DCAs), heater drain flow rates);
— Moisture separator reheater temperatures (heating steam temperature, hot reheat temperature, and TTDs, if applicable);
— Correction curves for off-normal conditions, particularly for changes in ambient conditions and the resulting effect on turbine backpressure and heat rate will be developed to adjust the baseline for standard conditions.

2.4.8.2. Baseline guidance

A thermal kit is a set of information including heat balance diagrams and specific information describing the design performance of the turbine cycle. Turbine vendors provide thermal kits upon the initial unit's construction and after any major modifications to the turbines. Generally thermal kits are provided with heat balance diagrams and other data that cover operation over a range of power levels (valve wide open, 100%, 75%. 50%, 25%).

The most accurate means of establishing a good baseline for a unit is with a performance test. Performance testing is similar to ASME Performance Test Code (PTC) -6 which can be used to establish a reliable baseline. However, any special configurations for the test need to be evaluated with respect to their effect on the 'normal' operational baseline. These tests will use high accuracy instrumentation which is temporarily installed in the plant to measure flows, pressures, temperatures, etc. The plant is placed in its optimum configuration during this test. Corrections can be made to the normal plant instrumentation to bridge the gap between a code test and normal monitoring of the plant with respect to the baseline.

Another method to establish a baseline especially after the plant has been modified is to build a thermodynamic model of the plant. An example of using a model containing vendor baseline information is for applications for testing and correcting to standard conditions. The boundary conditions (e.g. throttle pressure and moisture, reheat pressure drop, condenser back pressure, etc.) represent a design value to which corrections can be applied while performing studies to correct to a design standard basis.

The initial model can be modified to represent the current plant configuration and behaviour. The model is 'tuned' to performance test data or to current reliable plant data to provide best achievable results that are more representative of plant behaviour versus using original design data. The process will result in a representation of plant operation that can be used as a target for performance parameters during plant operation.

In the absence of an accurate vendor thermal kit or performance test, the plant establishes a baseline using historical operational data. This approach is only achievable if a good archive of these values has been maintained. Baseline or target information is maintained by the thermal performance programme documents developed by the TPE. These values need to be reviewed and updated periodically to reflect significant changes in the plant which would influence the ability to monitor thermal performance.

2.4.8.3. Quick search, recovery, and feedback analysis

The typical nuclear steam cycle consists of at least five main systems (main steam, extraction steam, condensate, main feedwater, and heater drains and vents) and thousands of components.

Because of the complexity of the steam cycle, the root cause determination for any given thermal performance problem or decline can be very difficult. For this reason, the thermal performance programme uses a structured approach to lost megawatt search and recovery efforts.

The beginning of the search and recovery effort will start with action values. Action values are performance deviation thresholds at which the search and recover efforts are started. Action values will vary depending on the plant design, quality of measurement system and operational characteristics. The following action values are an example based on industry experience and current instrument accuracy and uncertainty bands:

When thermal performance goals are not met or the unaccounted MWe exceed some established action value of MWe on a sustained basis. The value of when action is initiated will vary based on the ability to measure plant generation which is primarily a function of the watt hour metering, condenser pressure/cooling water inlet temperature and confidence in the correction curves. Typical action MWe values range from 1 to 3 MWe.

Rapid recovery of unit thermal performance depends on the application of structured methods for problem identification and corrective action. The thermal performance programme identifies the critical performance parameters (throttle pressure, backpressure, final feedwater temperature, extraction pressures, heater TTDs and DCAs, and so forth) and evaluates the MWe sensitivity of each.

The TPE needs to develop and maintain a list of all cycle bypass isolation valves that isolate high-energy steam, feedwater, condensate, or drain flows from the condenser during normal, full-power operation.

These flow paths may also be evaluated using the heat balance model or other methods, such as special testing, to quantify their potential contribution to megawatt loss events.

An investigation team led by the TPE and assisted by other site personnel will investigate the condition reports when the thermal performance losses are excessive. To assist in a structured search, a logical problem analysis technique is recommended to be adopted and applied. Guidance is provided in various industry documents including EPRI specific to thermal performance troubleshooting.

Depending on the nature and severity of the loss it is advisable to form a troubleshooting team to provide an objective approach. In some cases, it is helpful to acquire outside resources such as representatives from other plants in the utility or even outside the utility.

2.4.9. Training

Job-specific initial and professional development training for the TPE is specified in the Thermal Performance Monitoring Qualification Guide (Appendix D) [1]. This training addresses the five thermal performance programme fundamental elements and includes requirements for specific knowledge as well as demonstrated proficiencies.

Continuing training is accomplished through participation in industry forums and annual meetings, as discussed within the communications section. Likewise, participation in peer reviews and evaluations of programmes, methods, experience, problems, and improvements of other plants and utilities is encouraged to broaden the knowledge base and keep abreast of technological developments. The following areas are advised to be included in the training programme.

(a) Generic thermal performance skills
- Spreadsheet/database software;
- Heat balance software user training;
- Steam turbine performance precision test instrumentation and data acquisition;
- ASME (or applicable) performance test code turbine testing;
- Use of computer databases;
- Applied engineering fundamentals;
- Thermal performance program document;
- Heat balance drawing series.

(b) Plant specific operating procedures
- Determination of steam and water leakage into condenser via isolation valves;
- Moisture separator reheater performance tests;
- Main feedwater flow determination using feedwater ultrasonic flowmeter calorimetric calculation;
- Turbine cycle performance test ASME PTC 6 [3]
- Feedwater heater performance test ASME PTC 12.1 [4]
- Turbine pressure ratios;
- Cycle performance losses due to steam and water leakage;
- Turbine back-pressure measurement;
- Cooling tower performance pest;
- Turbine cycle performance test/condenser performance;
- Feedwater heater level optimization test;
- Turbine generator supervisory instrumentation;
- Turbine cycle performance monitoring instruction.

(c) Knowledge requirements
- Thermodynamic principles;
  - Carnot cycle;
  - Rankine cycle;
  - First and Second Law;
  - Role of assumptions;
  - Mass calculations;
  - Pressure and fluid flow equations;
  - Steam prosperities;
  - Heat transfer.
  - Component equations.
- Component understanding – Turbines, MSRs, condenser, feedwater heater, reactor, SG, cooling towers;
  - Purpose;
  - Principle of operation;
  - Structures;
  - Performance parameters;
  - Identification of component losses.
- Generation metering;

- Instrumentation – flow, pressure, temperature;
  - Types;
  - Failure modes;
  - Troubleshooting;
  - Usage;
  - Calibration.
- Calculation of core thermal power;
- Cycle isolation calculation methods;
- Statistics and uncertainty;
- Design basis;
- Design calculations;
- Heat balances;
- Heat balance software models
- Software control;
- Backpressure and condenser correction methodology;
- Baseline tests and reports;
- Basis sheets;
- Performance goals, including yearly goal setting and goal documentation;
- Monitoring and trending;
  - Daily review;
  - Weekly turbine cycle thermal performance report and megawatt accounting;
  - Thermal performance evaluation spreadsheets that include daily performance indicators (monthly report, lookup tables, trends), constants, and baseline;
  - Adverse trends;
  - Comparison to goal and action values;
  - Corrective actions;
  - Monthly report to business plan spreadsheet;
  - Integrated control system BOP software, constants, condenser backpressure curves, log, accumulators, and trend groups;
  - Predictive maintenance monitoring using acoustics and IR imaging;
  - Related BOP procedures and interface with system engineers (major indicator versus specific equipment).

(d) Search and recovery
- Plant plan of action procedures or guides;
- Trends (power indicators, turbine cycle complete pre/post database review);
- Process versus precision data, verification of important inputs, and specific equipment tests;
- Major turbine cycle performance tests;
- Heat balance comparisons and sensitivity (what-if) studies;
- Steam path audits;
- Outage preparation and inspections;
- Fault tree analysis.

(e) Communications, administration, and responsibilities
- Management responsibilities and interface;
- TPE peer team, EPRI Plant Performance Enhancement Programme (P^2EP) [5] coordinator;

- Industry codes, references, support, EPRI, ASME;
- Audits (peer self-assessments, INPO, EPRI);
- Training for task performance evaluation, training of system engineers, operations, and managers;
- Testing equipment, measuring and test equipment, and accuracy;
- Programme health reports, problem evaluation reports, and system notebook.

2.4.10. Implementation

2.4.10.1. Data flow from measurements to key performance indicator to actions to follow-up.

In order to keep the overall process efficient, it is helpful for the TPE to evaluate the flow of information from the information resource point (e.g. plant computer) to the various end products such as KPIs or work documents to resolve an issue. In Figure 4 below, there is an example of the data flow process that utilizes the various sources of input for data analysis and troubleshooting. Then the process moves to reporting and support functions. Each plant will have its own processes, supporting functions and reporting requirements which will fit into the overall thermal performance programme.

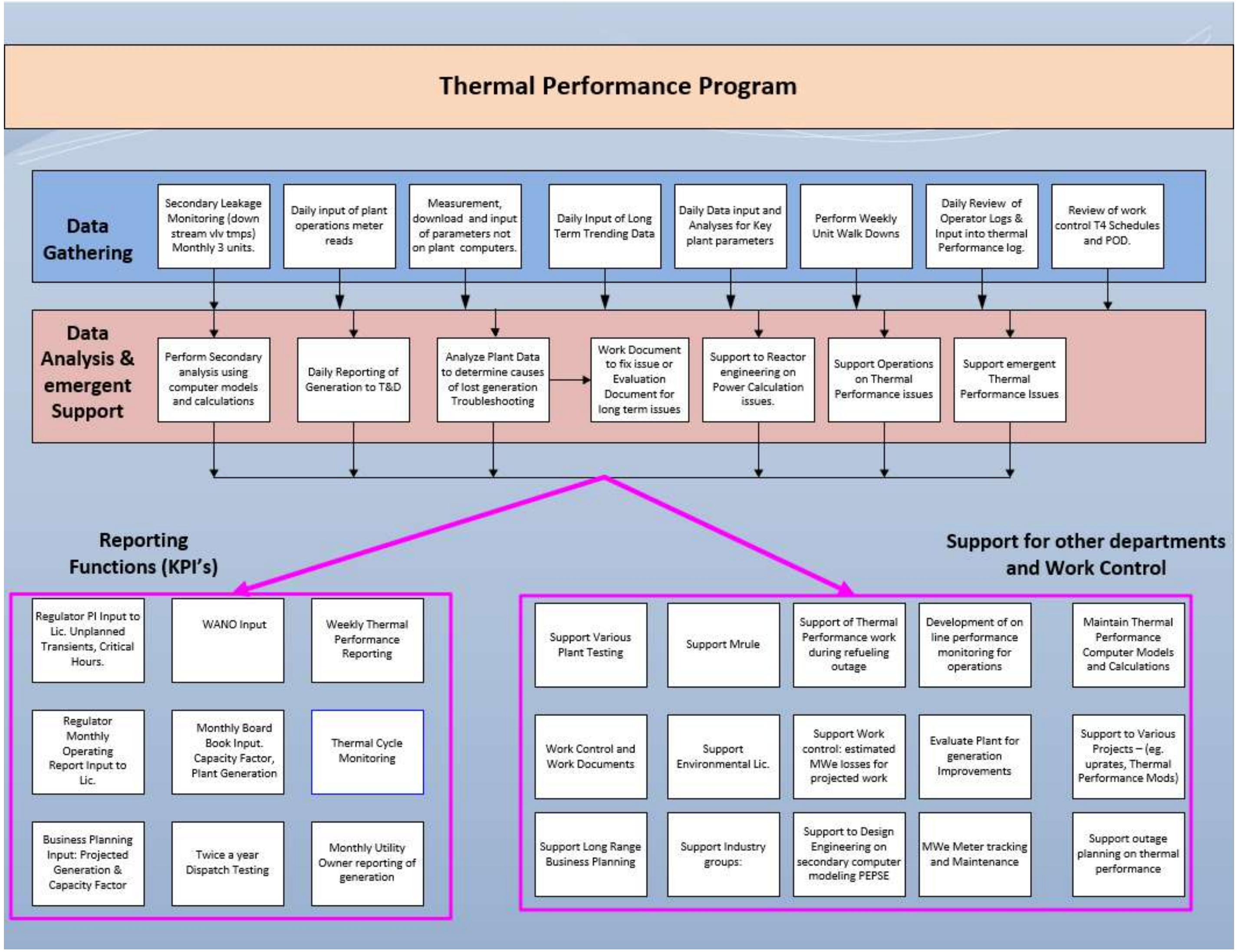

FIG. 4. Thermal performance programme – data flow process

2.4.10.2. *Daily practice for thermal performance engineer*

Trending is one of the basic functions of thermal performance. Long term and daily trending and heat rate trend along with advanced pattern recognition trending provides a means to evaluate the thermal cycle, quantify losses, and predict the future condition of the thermal cycle.

Every plant needs to have a thermal performance trending system. These can be developed internally using a commercially available spreadsheet programme with input from the plant process computer via a separate data historian. In other cases, third party software package can be acquired for trending. Many third-party thermal performance evaluation software programmes contain a data historian system that provides a good set of trending tools as well as the ability to develop presentation ready charts for reports and management meetings.

The trending process needs to be automated to efficiently and effectively monitor the hundreds of trends that are part of a normal thermal performance programme. The list of evaluations below is an example of trends that may be included in the monitoring programme:

(a) Daily performance evaluation

The thermal performance programme needs to be set up to include a daily review of important plant parameters. The data snapshot is usually collected at the same time every day and is filtered for less than full power operation and off-normal line-ups or conditions. The data collection period varies from 10 minutes to 2 hours depending on how much the data needs to be smoothed and the preference of the engineer.

A separate spreadsheet needs to contain design/as-built component information to facilitate accurate calculations using the snapshot data. This information can then be changed based on component maintenance such as tube plugging information, expected feedwater heater TTDs and DCAs temperature and other information pertinent information.

A separate spreadsheet or data base containing initial cycle conditions is also helpful in the analysis process for correcting heat rate or MW to the design condition.

Automation of daily data collection and analysis will free the TPE from the daily grind of data collection and allow them to spend more quality time on evaluation. A note of caution about automation is that the TPE needs to fully understand how the data acquisition system collects, processes, stores the data.

The overall analysis is recommended to progress from a top level down to the details as necessary. For example, in the Figure 5 below.

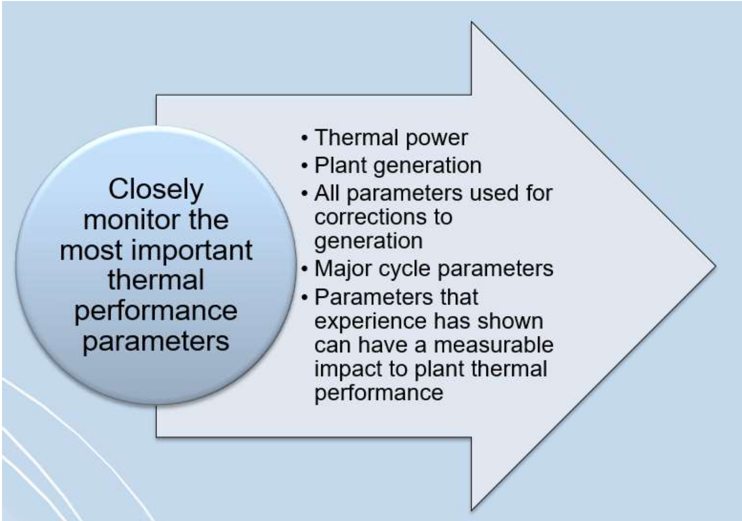

FIG. 5. The overall analysis of the monitoring of the thermal performance parameters

This would also include plant parameters used to correct generation to a baseline condition:

- All thermal power calculation instruments;
- Circulating water temperature;
- Circulating water flow;
- Condenser pressure;
- Condenser hot well temperature;
- Exhaust hood temperature;
- Hydrogen pressure;
- Main steam pressure;
- Main steam temperature (if applicable);
- Reheat temperature and MSR pressure drop.

The progress of the evaluation would proceed to more detail as an issue is identified. The following Figs 6 – 9 show this progress.

(b) Top level evaluation

CANDU Nuclear Power Plant - Unit 1 Thermal Performance Summary				
Current Statistics				**October 20, 2013**
Gross Dependable Capacity			721.5	MWe
Actual Gross Output			711.8	MWe
MW Deficit			-9.7	MWe
Accounted MWs			-8.2	MWe
Unaccounted MWs			-1.5	MWe
Core Thermal Power			100.0%	%
Average Condenser Back Pressure			8.05	kpa
Circulating Water Inlet Temperature			24.3	deg C
Contact True North Consulting	Status	Utility Menu	MW Graphs	MW Accounting

FIG. 6. Progress of the evaluation of thermal performance [1]

(c) Detailed level evaluation

MW Accounting Daily Report	# Alerts →	10	1/7/2010	Main Menu	
Gross Dependable Capacity	MWe	737.9	Steam Turbines	FW Heaters	
Actual Gross Output	MWe	706.6			
MW Deficit	MWe	-31.2	MSRs		
Current Gross Heat Rate	kJ/kW-hr	10504.7	Condensers		
Current Gross Heat Rate Corrected to GDC	kJ/kW-hr	10059.9			
TPI	%	95.8			
ACCOUNTED MW DEVIATIONS (-Loss +Gains)			Trend Graphs		
	Units	**Today**	**Baseline**	**Deviation**	**MWe**
Feedwater Heater 5 TTD	deg C	3.32	2.78	0.54	-0.18
Blowdown Flow	kg/s	6.76	4.00	2.76	-0.03
Reheater TTD	deg C	14.87	20.39	-5.52	1.58
Condenser Efficiency	kpa	3.27	2.68	0.58	0.09
Condenser Sub Cooling	deg C	0.66	0.00	0.66	-0.06
Throttle Steam Pressure	barsa	4554.2	4551.2	2.9	0.05
Core Thermal Power	MWth	2062.0	2062.0	0.0	-0.02
Generator Power Factor	na	1.0000	1.0000	0.0000	0.00
Auxiliary Steam	kg/s	0.00	0.00	0.00	0.00
Miscellaneous (from Manual Input)	MWe				0.00
				Accounted MW Deviations (-Loss +Gains)	**1.44**
				Unaccounted MWe	**-32.68**

FIG. 7. Progress of the evaluation of thermal performance [1]

(d) Component level evaluation

Condenser Daily Report			1/7/2010		MW Accounting	
Performance - All Condensers	**Today**	**Baseline**	**Delta**	**Historical**	**Delta**	**Status**
Subcooling (deg C)	0.7	0.0	0.7	0.3	0.4	Okay
Performance - Condenser A	**Today**	**Baseline**	**Delta**	**Historical**	**Delta**	**Status**
Cleanliness Factor (%)	161.8	85.0	76.8	#NUM!	#NUM!	#NUM!
TTD (deg C)	12.6	4.4	8.2	#NUM!	#NUM!	0.00
Condenser Back Pressure (kpa)	3.29	31.75	-28.5	3.2	0.0	
Performance - Condenser B	**Today**	**Baseline**	**Delta**	**Historical**	**Delta**	**Status**
Cleanliness Factor (%)	163.3	85.0	78.3	#NUM!	#NUM!	#NUM!
TTD (deg C)	12.5	4.4	8.0	4.0	8.4	0.0
Condenser Back Pressure (kpa)	3.26	31.75	-28.5	3.3	0.0	
Performance - Condenser C	**Today**	**Baseline**	**Delta**	**Historical**	**Delta**	**Status**
Cleanliness Factor (%)	163.0	85.0	78.0	#NUM!	#NUM!	#NUM!
TTD (deg C)	12.5	4.4	8.1	3.9	8.7	0.00
Condenser Back Pressure (kpa)	3.26	31.75	-28.49	3.2	0.0	
Supporting Data	**A**	**B**	**C**			
Calculated Circulating Water Flow (m^3/s)	63	63	63			
Circulating Water Flow (ManIn) (m3/s)	46	46	46			
Circulating Water Inlet Temperature (deg C	7.5	7.5	7.5			
Circulating Water Out Temperature (deg C)	13.0	13.0	13.0			
Circulating Water Temperature Rise (deg C	5.5	5.5	5.5			
Hotwell Temperature (deg C)	24.9	24.9	24.9			
LMTD (deg C)	15.2	15.1	15.1			
Total Effective Tube Surface Area (m^2)	1180	1180	1180			
Actual HTC (kJ/s-m2-deg C)	4.9	5.0	5.0			
Ideal HTC (kJ/s-m2-deg C)	4.5	4.5	4.5			

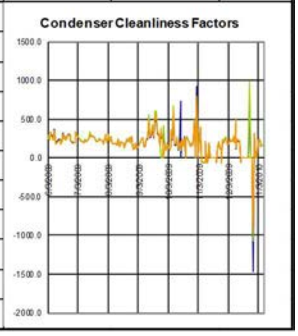

FIG. 8. Progress of the evaluation of thermal performance [1]

Many plants also include a review of the core thermal power health based on data reconciliation or a best estimate calculation.

U1 Thermal Power Predictor							10/20/2013	MW Accounting				
Description	Tag Name	Units	Today	Licensed	Historical Average	Predicted	Uncertainty	Power Health				
REACTOR THERMAL POWER	DT	MWth	2062.3	2062.0	2061.5	2051.6	0.26%	-0.52%				
Power Calculation Independent							.					
Description	Tag Name	Units	Today	Benchmark Data (Manual Input)	Predicted CTP	Systematic Uncertainty	Random Uncertainty	Total Uncertainty	Sensitivity (Manual Input)	1/Uncertainty Squared	Weighting Factor	Weighted Contribution to CTP
First Stage Pressure	averaged	kPa	3663.7	3692.8	2045.8	1.00%	0.19%	1.0186%	1.000000	9637	0.0602	123.2486
HP Turbine Exhaust Pressure	averaged	kPa	1037.9	1044.3	2049.4	1.00%	0.16%	1.0135%	1.000000	9735	0.0609	124.7206
Pressure to Reheater (same as HP Exhaust)	averaged	kPa	1037.9	1044.3	2049.4	1.00%	0.16%	1.0135%	1.000000	9735	0.0609	124.7206
MSR 1 Outlet Steam Pressure	averaged	kPa	1031.0	1037.8	2048.6	1.00%	0.17%	1.0149%	1.000000	9709	0.0607	124.3400
MSR 2 Outlet Steam Pressure	averaged	kPa	1024.2	1028.4	2053.6	1.00%	0.17%	1.0141%	1.000000	9724	0.0608	124.8371
LOW PRESSURE TURBINE "A" INLET STEAM PRESSURE	64112-PT509	kPa	1033.7	1042.5	2044.5	1.00%	0.16%	1.0132%	1.000000	9742	0.0609	124.5090
LOW PRESSURE TURBINE "B" INLET STEAM PRESSURE	64112-PT510	kPa	1006.8	1014.2	2047.1	1.00%	0.17%	1.0147%	1.000000	9713	0.0607	124.2915
LOW PRESSURE TURBINE "C" INLET STEAM PRESSURE	64112-PT511	kPa	1033.3	1040.5	2047.7	1.00%	0.16%	1.0131%	1.000000	9743	0.0609	124.7162
DA PRESS.	64331-PT100	kPa	421.8	426.6	2038.8	1.00%	0.24%	1.0274%	1.000000	9474	0.0592	120.7483
HP5 EXTR. STEAM HEADER PRESS.	64335-PT503	kPa	1265.3	1275.6	2045.4	1.00%	0.17%	1.0141%	1.000000	9723	0.0608	124.3216
HP5 EXTR. STEAM HEADER PRESS.	64335-PT503	kPa	1265.3	1275.6	2045.4	1.00%	0.17%	1.0141%	1.000000	9723	0.0608	124.3216
BOILER 1 STEAM FLOW	63611-FT501	kg/s	263.0	261.0	2077.6	1.00%	0.43%	1.0905%	1.000000	8409	0.0526	109.2162
BOILER 2 STEAM FLOW	63611-FT502	kg/s	261.6	262.6	2053.9	1.00%	0.44%	1.0928%	1.000000	8374	0.0523	107.5172
BOILER 3 STEAM FLOW	63611-FT503	kg/s	257.9	257.8	2062.5	1.00%	0.45%	1.0949%	1.000000	8342	0.0521	107.5495
BOILER 4 STEAM FLOW	63611-FT504	kg/s	262.1	261.2	2069.0	1.00%	0.46%	1.1000%	1.000000	8264	0.0517	106.8920
BOILER 1 FEEDWATER INLET TEMP.	64323-TT-5	deg C	187.5	188.0	2051.5	1.00%	0.06%	1.0019%	1.000000	9963	0.0623	127.7705
BOILER 2 FEEDWATER INLET TEMP.	64323-TT-6	deg C	187.2	187.5	2055.3	1.00%	0.07%	1.0022%	1.000000	9956	0.0622	127.9176
										Sum	Sum	Sum
										159968	1.000000	**2051.6380**

FIG. 9. Progress of the evaluation of thermal performance [1]

2.4.10.3. What do you do when you find a problem?

Search and recovery are processes of returning the unit to its baseline thermal performance in a cost-effective manner once degradation from the established baseline is detected. The process is one of diagnosis to pinpoint the source of the degradation. It is the responsibility of the TPE to identify degradation when it occurs and initiate the actions necessary to correct it. The process needs to be thorough in the search and recovery effort such that any source of degradation is accurately identified. Actions initiated need to be based on sound evidence to ensure the plant organization is not unnecessarily cycled. When possible, the TPE may want to work with the responsible system engineer using various diagnostic tools available to search for the source of degradation. It is the role of the TPE to drive the recovery to completion.

Rapid recovery of unit thermal performance (that is, lost megawatts) depends on the application of structured methods for problem identification and corrective action. The thermal performance programme identifies the critical performance parameters (throttle pressure, backpressure, final feedwater temperature, extraction pressures, heater TTDs and DCAs, and so forth) and evaluates the MWe sensitivity of each. The programme needs to develop and maintain a list of all cycle bypass isolation valves that isolate high energy steam, feedwater, condensate, or drain flows from the condenser during normal, full power operation. These flow paths are also evaluated using the heat balance model or other methods, such as special testing, to quantify their potential contribution to megawatt loss events. Also, a logical problem analysis technique needs to be adopted and applied.

3. TECHNICAL ELEMENTS OF A THERMAL PERFORMANCE PROGRAMME

3.1. DETERMINATION AND BASIS OF KEY PERFORMANCE INDICATORS

For a thermal performance programme to be effective, appropriate KPIs need to be provided. The development of these indicators will be accomplished in conjunction with plant management and be aligned with the overall plant indicators. Often some key business related KPIs, such as plant CF, will be aligned with the thermal performance KPIs. Also, KPIs developed by external organizations such as WANO may be used to develop internal KPIs.

KPIs are a means to refocus on the power plant's principal product; electricity. Since most NPPs are very large and complex, it is often a challenge that the TPE is far removed from the actual equipment. At NPPs a propensity to focus on reactor operation can eclipse the generation of electricity. The TPE's development and utilization of appropriate KPIs can alleviate these issues.

Additionally, it is important for the TPE not to lose focus on what the KPIs are for. Often plant management is so concerned with simply meeting the KPI that there is a desire to change the baseline value of the KPI. It is, however, important to understand what the target is and why.

If the KPI value baseline is too high or not considering uncontrollable conditions, then it will not be realistic. It is essential for the TPE and plant management to be open to ensuring that the values being calculated are relevant to the desired goal. This can be accomplished by considering the following questions:

— What is the expected generation value based on (design and/or installed/operating configuration)?
— What is the actual generation?
— What is the difference?
— Why is there a difference?
— How can the problem be fixed?
— When can the problem be fixed?
— What are the risks associated with either fixing or not fixing the problem?

The Figs 10 – 11 below are essentially asking these same questions.

CANDU Nuclear Power Plant - Unit 1
Thermal Performance Summary

Current Statistics		October 20, 2013
Gross Dependable Capacity	721.5	MWe
Actual Gross Output	711.8	MWe
MW Deficit	-9.7	MWe
Accounted MWs	-8.2	MWe
Unaccounted MWs	-1.5	MWe
Core Thermal Power	100.0%	%
Average Condenser Back Pressure	8.05	kpa
Circulating Water Inlet Temperature	24.3	deg C

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FIG. 10. Progress of the evaluation of thermal performance

Expected generation – What the plant is expected generation for the given atmospheric conditions?

Gross generation – What is the plant generating?

MW deficit – What is the difference between expected generation for the given atmospheric condition and actual plant generation?

Another way to show the relationship between expected generation and actual generation is expressed graphically as below in Figure 11.

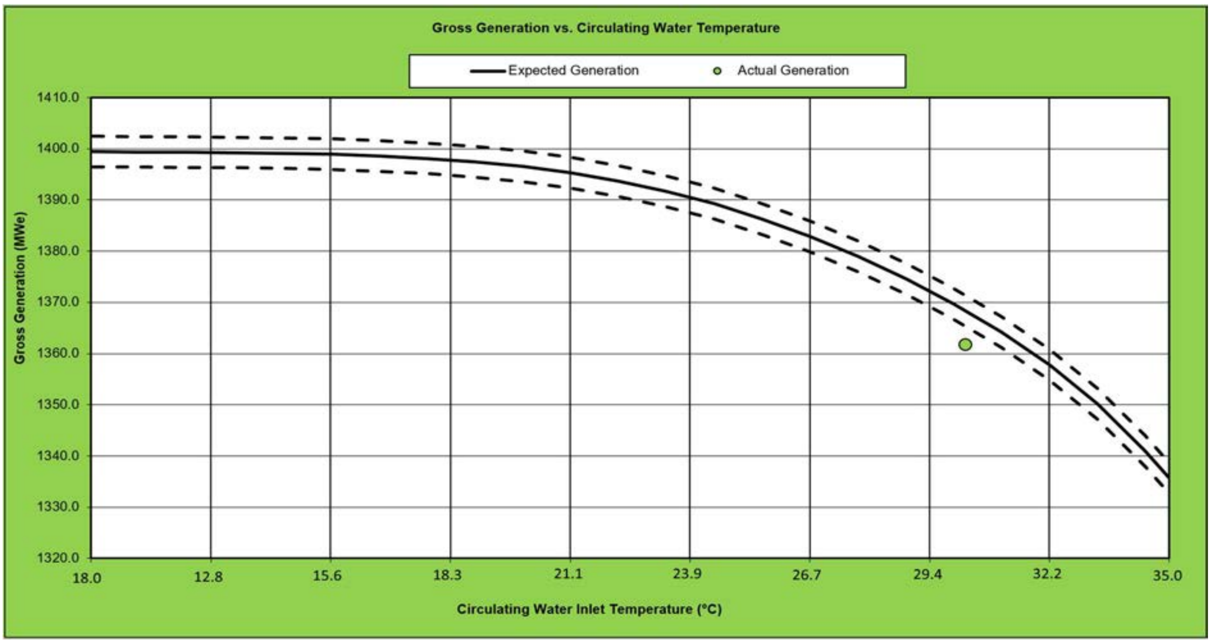


FIG. 11. Examples of KPIs focused on thermal performance

3.1.1. Corrected generation

Corrected generation is determined by measuring plant generation and correcting it to standard conditions. The corrected generation is then compared to the baseline which is already at the standard conditions. The difference would provide an indication of a generation deficit.

If the plant is operating with no losses other than those linked to environmental factors, the corrected generation ought to remain constant. Whereas in the figure above, the target generation is variable based on environmental conditions. Generation can be corrected based on condenser pressure or circulating water inlet temperature.

3.1.2. Capacity factor

Capacity Factor (CF) – The ratio of actual generation over possible generation as shown in the Eq. (1).

$$CF = \frac{Actual\ generation\ in\ period}{Possible\ generation\ in\ period} \quad (1)$$

The CF are calculated from different 'possible generation'.

Net maximal dependable capacity is lowest summer generation on the hottest day of the year from historical data, thus the maximal dependable capacity CF can be above 100% in the winter months.

Depending on how you calculate the expected power, you can have a CF > 100%.

3.1.3. Thermal performance indicator

In the past WANO had established an indicator to provide a measure of plant efficiency. It has since abandoned this indicator, but many plants still use this parameter as a KPI.

The thermal performance indicator (TPI) is the ratio of the corrected plant heat rate (HR) to the plant best achievable heat rate (BAHR). Heat Rate is a ratio of the energy produced in kJ/h (also in terms of calories or joules) to the generation in kilowatts produced (kJ/kWh). The calculation to provide the corrections to plant heat rate may vary from plant to plant but all would include a correction to standard conditions such as condenser cooling water inlet temperature or condenser pressure. Some plants will also add a correction for SG blowdown flow or other plant conditions that are required for normal plant operation and can affect the plant heat rate. The thermal performance is expression is indicated in Eq. (2).

$$TPI = \left(\frac{BAHR}{HR}\right) \cdot 100 \tag{2}$$

Where *TPI* is the thermal performance indicator, *BAHR* is the best achievable heat rate (kJ/kWh) and *HR* is the heat rate (kJ/kWh).

The TPI can be determined by selecting a time period and performing the calculation based on the total reactor power (kJ/h) and total power (kWh) or based on a specific test period. The advantage of the TPI is that is provides an indication of overall plant efficiency. Also, if a plant is operating at a different load than design the TPI can provide a means of comparing the plant efficiency at different loads. The disadvantage from an industry comparison point of view is that the corrections have been inconsistently applied, which led to its abandonment by INPO/WANO.

Data for calculating the thermal performance index are taken from the one day that is most representative of the unit performance for the entire month. the TPI data need to be taken during stable power operation when the plant is greater than 80 percent power, following any single 24-hour period of stable operation at the same power level.

Corrected gross heat rate needs only to be calculated on days when the reactor power averages 99.5% or above and no major BOP components, such as a 2nd stage reheater, are out of service. Also, it is necessary to have a complete day’s data available. If major BOP components are out of service for longer than 30 days or if the unit cannot operate at or above 99.5% reactor power for longer than 30 days, the TPE needs to decide how to perform the calculations. The approach how to calculate daily gross heat rate is shown in Eq. (3)

$$\begin{aligned} DGHR = \sum_{t=1}^{24} \left[CTP \times 0.01 \times LPL + \left(EC \times 24 \frac{h}{day} \right) \times 3600 \ \frac{kJ}{kWh} \right] \\ / \left(DGE \times 1000 \frac{kW}{MW} \right) \end{aligned} \tag{3}$$

Where *CTP* is secondary calorimetric power (MW), *DGE* is daily gross electrical output (MWh), *LPL* is license power limit (MWth) and *EC* is energy credits (MWth).

The best achievable heat rate is determined by the TPE. The daily thermal performance index is obtained by dividing the BAHR (kJ/kWh) by the daily correct unit gross heat rate.

Note: The EC in the equation represents the net energy credits to the primary that are subtracted from the secondary calorimetric used to obtain CTP. This value is the design net energy credits used by General Electric/Westinghouse to obtain the unit design heat rate.

The daily corrected gross heat rate is calculated by dividing the daily gross heat rate by the correction factor obtained from the exhaust pressure correction factors curve, in the thermal kit.

3.1.4. Lost generation

Lost generation is calculated by subtracting the expected generation from the actual generation. The expected generation is determined by adjusting the baseline generation to the standard conditions such as ambient wet-bulb temperature, circulating water inlet temperature or condenser pressure. Other indicators are calculated to account for the effect of lost generation on the overall plant performance.

3.2. MONITORING TIMING

3.2.1. Periodic test

Periodic test can be the best scheme only if the data or information required for performance monitoring are not always available from the plant computer data system. The periodic testing needs to be conducted at scheduled intervals as regular as possible, however, the frequency of the testing can be increased in accordance with a performance authorities' engineering judgment. The testing may range from baselining for indexing expected levels of performance to routine testing under normal operating conditions. As usual in a normal operating condition, the TPE or a dedicated staff member in charge of a plant's efficiency compares current values with reference values of the plant's thermal performance parameters. Each current value becomes representable by arithmetic or time averaging of the plant operating data recorded at regular intervals from an instrument during, for example, 2 hours. It is advised not to obtain the current values during any load change, other scheduled periodic test or facility check within the plant.

The reference values for the comparison with present thermal performance parameters are called benchmark values whose setting time could be chosen either when the plant has optimum thermal performance during the early phase of the current cycle or when the plant's thermal performance is qualified during the beginning of the cycle. The TPE or a dedicated staff member in charge of the plant's efficiency needs to record the thermal performance trend or analyse the cause of performance degradation if the difference in electric power output becomes larger than a specified range between the current values and the benchmark values.

3.2.2. Continuous test (on-line thermal performance monitoring system)

EDF employs an on-line thermal performance monitoring system that can be established to continuously store the plant's thermal performance parameters to a plant computer data system or to separate archives for thermal performance monitoring. The on-line system offers various benefits, but the most appealing advantage is to accumulate operating data over time. As a kind of periodic testing, a performance engineer can implement daily or weekly performance monitoring if the on-line thermal performance monitoring system is available. The continuous performance analysis starts with comparing current values with reference values of the plant's thermal performance parameters.

3.2.3. Pre and post outage test

Typically, the performance test is implemented within 30 days before and after the outage of each cycle in order to overlook the change of thermal performance index and performance history at each component equipment in the turbine cycle. The TPE or a dedicated staff member in charge of plant's efficiency analyses the test result and reports to the plant manager. Then it is registered to plant archives for the purpose of the plant's thermal performance management.

3.2.4. Pre and post modification test

As a part of periodic test, the plant's performance monitoring is conducted prior to and immediately following outages when key component equipment is repaired or modified. Such facilities are SG, HP, LP, MSR, and condenser, which directly affect the output of the turbine cycle. Particularly, this test results need to be reflected on the design factors of plant's on-line thermal performance monitoring system.

3.3. INPUT DATA

3.3.1. Benchmarking (baselining data) acquisition

In order to know as much as possible about the thermal performance of an NPP at a given time, it is recommended to characterize the state of the plant. This data need to be acquired as close as possible to the moment at which one wishes to evaluate plant performance.

This data is used either to define the operating conditions of the plant at a given time (boundary conditions), or to compare with expected parameters at different points of the plant. The measured parameters are pressure (approx. between 0.02 bar and 95 bar), temperature (approx. between 0 °C and 300 °C), flow rate (approx. between 0 and 33 000 kg/s) and electrical power.

The reliability of the data is very important to the process, performance monitoring is nearly impossible without reliable data. Inaccurate or unrepresentative data can lead to errors in diagnosis, discredit the process and undermine efforts made, including the development of reference models or the establishment of a specific organization. Therefore, substantial effort needs to be made regarding determining and acquiring input data.

3.3.2. Data collection

3.3.2.1. Instrumentation (location, type, etc.)

Although the more sensors one has, the more components that can be monitored, not all components have a significant impact on plant efficiency. For example, the performance of the first stage, low pressure secondary feed-water heaters (located after the condenser) has little impact on overall thermal efficiency.

A choice needs to be made to monitor only those components for which the instrumentation is a worthwhile investment. In some cases, instrumentation is based more on reliability than efficiency monitoring.

A greater number of sensors allows for cross-checking between measurements, thereby increasing one's confidence in the reliability of the measurements. Again, an optimum balance needs to be found between the reliability of the monitoring and the cost of installing the sensors. The more reliable the sensors and acquisition systems are, the faster performance drifts can be identified. Thus, the accuracy of the selected sensors needs to be optimized according to the detection threshold needed.

A cost-effective approach is to focus on high energy components: the high-pressure stage of the turbine, the MSR, the condensers, the high-pressure heaters, and the cooling towers. Following this approach, it is possible to achieve gains of 2 to 5 MWe for units producing approx. 1000 MWe.

For these systems, a second level of optimization is possible by reinforcing the instrumentation in order to concentrate on the largest performance losses and instrumenting less the parameters which have a smaller impact. For instance, for high pressure heaters, it is preferable to instrument the flow of each drain line, since the failure of a high-pressure heater can rapidly lead to a loss of several MWe.

Some power plants do not measure the steam pressure in each high-pressure heater. Instead they use the existing pressure gauge on each turbine bled steam line and estimate the shell pressure in the heater. This means, however, that the pressure drop between the measured pressure and the desired information becomes a fixed estimate. The true pressure in the shell may be different due to changes in the pressure drop related to flow and specific volume. There is therefore a risk that the estimated pressure in the heaters is incorrect and over time may lead to unreliability in the efficiency monitoring of the heaters.

With regards to the condenser, instrumentation is focused on the measurement of the vacuum. Between four and six sensors are needed to ensure a good representation of the vacuum. In this case, cross-checking is enabled and ensures a reliable vacuum value.

A final example can be given concerning the monitoring of cooling towers. The air temperature is a key parameter of the tower's performance. EDF test standards prescribe the use of precise instrumentation in the air intake (about 20 sensors). For reliable monitoring, three or four sensors are enough to allow cross-checks of measurements. In Figure 12 below, a thermal model with sensors is shown. This figure serves only illustrative purposes (indicating the level of detail related to data collection).

The sensors and acquisition systems to be used for performance monitoring need to have a reliability comparable to the one required by international standards dedicated to thermal performance. Some examples of these standards are as follow:

NF-EN-60953-2 (Turbine) [6];

NF E38-350 (Condensers) [7];

ISO 16345 (Cooling towers) [8];

ASME Performance Test Code series for all turbine cycle equipment.

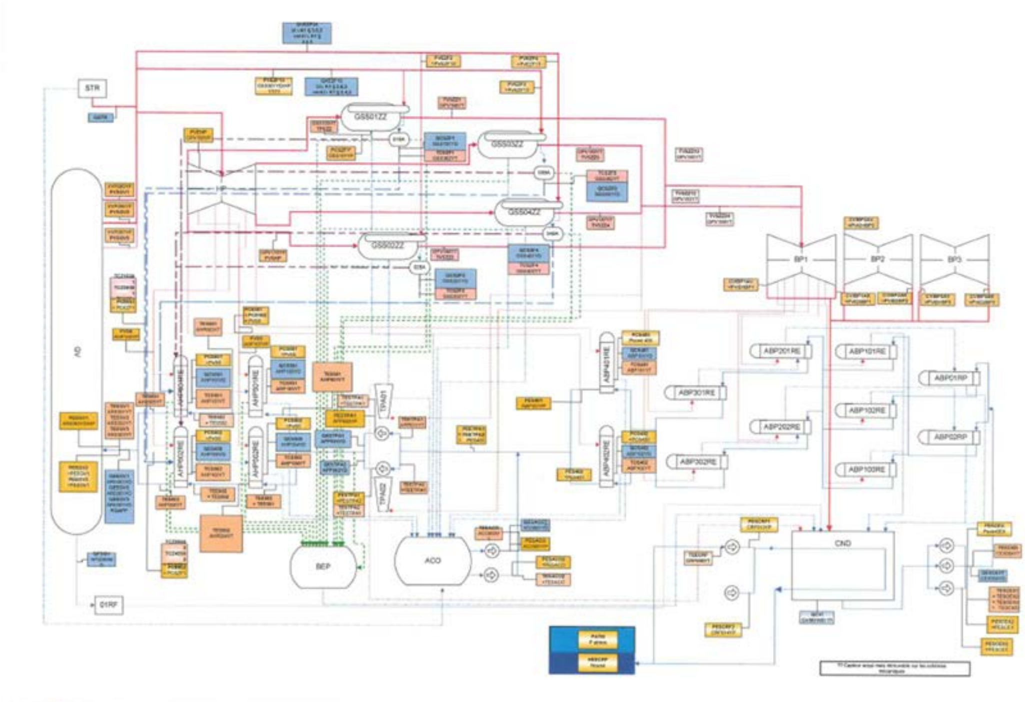


FIG. 12. Thermal model (with sensors)

3.3.2.2. Measurement (calibration, uncertainty, alternate measurement etc.)

As mentioned above, the reliability required is the one defined in international standards dedicated to thermal performance acceptance tests. Since thermal performance needs to be monitored over time, regular calibration is required. Suggested calibration frequencies are provided below:

- Vacuum sensors – yearly;
- Differential pressure sensors for flow – yearly;
- Pressure sensors – two years;
- Temperature sensors – could be as much as three or four years;

Plant installation guidelines for these sensors need to be respected. Special attention needs to be paid to the installation of instrument lines on let-down lines.

3.3.2.3. Data acquisition system (Computer, PI, etc.)

The acquisition system’s accuracy is again based on international standards. For each measured parameter, the overall uncertainty (consider the uncertainty of the sensors) and that of the acquisition system need to remain within the limits imposed by the standards.

Since we need to monitor thermal performance over time, it is necessary to guarantee the reliability of the data acquisition system in the long term. Various interventions are expected in time, for instance:

- Modification of calibration coefficients of sensors;
- Replacement of a faulty sensor;

- Repair of degraded wiring.

In addition, long term performance monitoring requires the ability to guarantee consistent data over time.

For all these reasons, it is best to focus on a permanent acquisition system, connected to a dedicated computer whose reference configuration can only be modified by certain users. This methodology will help guarantee the quality of the input data over time.

In some cases, it may be useful to have a modular acquisition system to which you can easily add new sensors. This makes it possible to progressively complete the instrumentation in place with special instrumentation.

3.3.2.4. Special instrument (ultrasonic, infrared, etc.)

Additional special instrumentation can be put in place to:

- Provide complementary measurements to cross-check existing measurements and improve the precision of measurements.
- Confirm a diagnosis performed with the existing instrumentation.
- Validate hypothetical models used in the performance monitoring tool.
- Carry out hard-to-implement monitoring dedicated to specific components (e.g. valves) and encourage the involvement of operators in performance monitoring.

This special instrumentation is generally set up for a limited period only. Various types of complementary instrumentation are used (sometimes it is the same kind of instrumentation as the permanent instrumentation):

- Ultrasonic flow measurement devices (for instance to check the division of flow between parallel heaters);
- Infrared (IR) thermometers (for instance to monitor valve leaks);
- Flow measurement via dilution rates (for an accurate, independent flow measure).

3.3.2.5. Manual reading (heater level, cooling water pressure drop, etc.)

The instrumentation proposed above is the basis for performance monitoring. When performance drifts are detected from these measurements, it may be useful to have additional information to confirm the diagnosis. This additional information comes from manual records made by an operator in the field.

This information can be:

- The water level in the heaters;
- The temperature of the pipes downstream of the floodgates (obtained via an IR thermometer).

For maximum reliability, it is very important that these manual records be made the same way, week after week. It is recommended, that the manual records are not dependant on the operator who performs them; only then can the records be used to complete the other measures already available.

In order to ensure the best possible reproducibility of these manual records, a procedure describing in detail each step of the 'performance sweep' is to be followed by the operator. It describes all the measures that are to be manually recorded, the exact location of the measurement points, and how to correctly perform the measurement.

3.3.3. Data validation

Errors in input data can be caused by:

- Poor metrological protocol;
- Non-standard operating plant conditions;
- Sensor failure;
- Incorrectly installed sensors;
- Incorrect configuration of the data acquisition system.

Therefore, it is necessary to constantly keep a critical eye on the quality of the input data.

Several approaches are possible to check the quality of the input data used for performance monitoring.

- A basic check is to look at the data and ensure that it remains between a minimum and a maximum value.
- The standard deviation of each numerical value during the acquisition is also a good indicator of the reliability of the data (for example, a zero standard deviation often means a faulty sensor).
- Trend monitoring for each sensor can detect surprising behaviour, especially since the operating conditions of the plant during the performance monitoring test are under control.
- Cross-checking the sensors is a very good way to check the quality of the data. Sometimes there are redundant test sensors (even if this is rather rare since, as has been said above, the number of test sensors is optimized to limit the cost of performance monitoring).

In Figure 13 below an example of a possible cross-comparison is given, in this case with regards to condenser vacuum measures. Condenser vacuum is generally obtained by averaging several vacuum sensors. The data obtained by these vacuum sensors need to be relatively close to each other. If one of the measured vacuums differs significantly from the others, it can be assumed that there is a measurement problem.

When cross-comparison between test sensors is not possible, one can use 'operating sensors'. NPPs have their physical data measured for the needs of everyday plant operation (called herein 'operating sensors'). These sensors are different from those used for performance monitoring (test sensors). However, they can be used to compare measurements when an operating sensor measures the same parameter as the test sensor (on the same location).

Sometimes, an operating sensor and a test sensor are not exactly at the same location but the physical data they measure are connected to each other by a simple physical law (e.g. mass or energy balance). It is then possible to cross-check these two data by relying on the physical law.

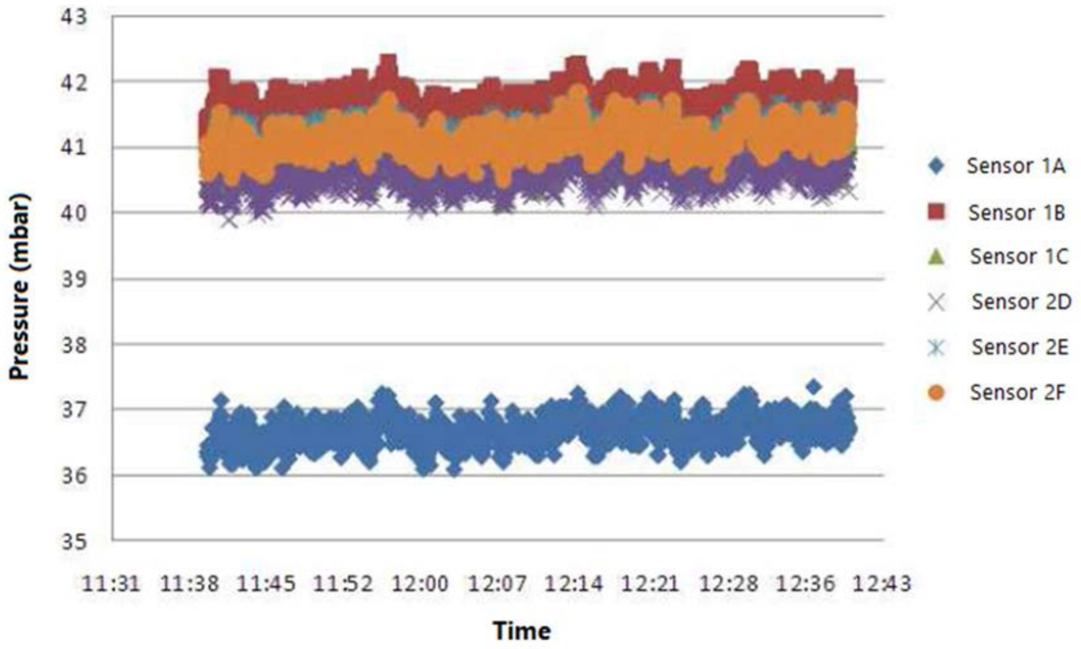


FIG. 13. Example of cross comparison between condenser vacuum sensors

It is necessary to be note that the preferred methods for data validation ought to be simple. Thus, they can be used daily by all those involved in performance monitoring (operators on site, engineering support etc.).

As described elsewhere in this publication, data validation and reconciliation (DVR) can be used to validate the base measurements. This process uses first principles and a statistical comparison to determine the most probable value for each measurement also called the reconciled value. These reconciled values can then be used as input to performance calculations or to identify instrument or component problems that can then be corrected.

3.3.4. Data retention

Performance monitoring relies on the ability to maintain access to old data, be it measured input data or more sophisticated data. Keeping records of the input data allows at any time to compare the situation with any past situation in order to improve diagnosis. As such, the raw data is backed-up, to ensure long term access to past data (covering several decades sometimes).

Most utilities have installed some means to accomplish this requirement. For example, EDF developed a large data storage project called ESPADON: ESPADON (Secure Space for Sharing and Analysing Nuclear Data) is a data storage project (BigData), based on Hadoop technologies.

The goal of any storage system is to have a safe data storage bringing together the various sources of their industrial data and offering business applications, in order to build a flexible service which can respond to the needs of NPPs (reporting, project management, etc.) This data centralization will allow an easy correlation of various information while controlling the safety of the data.

This system is designed to improve overall performance as follows:

- Overcome issues due to the amount of data management which will lead to improvements to the data quality to allow a better productivity of processes.
- Allow for better reactiveness since the data will be pushed continuously from the NPPs and allow faster and better analyses (providing access to the data, and the possibility of cross-checking different data sources).

The goal of the data retention programme is the following:

- To define a data governance.
- To identify needs related to data storage (use cases, return on investment and detailed data perimeter).
- To organize the needed management changes in our NPPs to promote the newly developed tools.

The sensor-based data come from the various sensors installed in NPPs. The corresponding data can be tracked online, or manually recorded in the field or during tests. They can be of analogue or binary type.

A visualization programme needs to provide:

- Access to a graphic interface: graphs, in the form of a logbook and in the form of a summary.
- Possibility to calculate complex indicators from process data.
- Possibility to carry out queries on all the process data; for instance, to measure the running time of a pump, to monitor a temperature gradient also.

Various utilities have implemented or are in the process of implementing data retention and integration programmes which will improve the ability of NPPs to monitor, diagnose and correct thermal performance issues. These programmes include data retention, validation, troubleshooting and reporting features integrated with the NPPs work control processes.

3.4. CYCLE ISOLATION LOSSES

3.4.1. Cycle isolation losses introduction

Cycle isolation loss is the condition that occurs when a high energy valve that is supposed to be closed is actually open and causes fluid leaking to the condenser. This fluid has already had an increase in its energy level by reactor generated energy. Any energy that goes into heating the fluid in the cycle that is not used for generating electricity is lost either to the atmosphere or even worse into the heat sink (condenser). In NPPs there is no way to recover the generation that this fluid would have contributed to. The plant will therefore be generating less electricity than expected, which is a direct impact on revenue and overall plant efficiency.

Often cycle isolation losses are not easy to identify because the leakage is to the condenser and cannot be seen (as shown in Figure 14). If the leakage is to the atmosphere via a vent, relief valve or a drain valve it is visible and therefore can be identified and repaired. Leakage to the condenser can go undetected for a significant amount of time unless methods are implemented to identify and track this leakage.

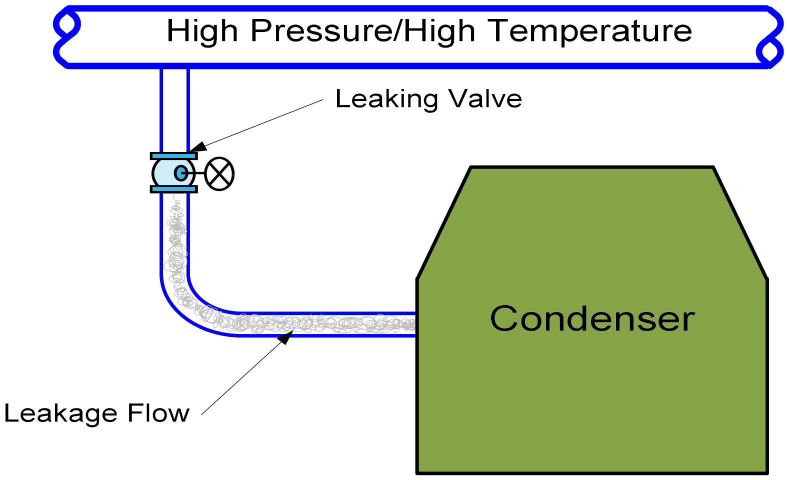


FIG. 14. Cycle isolation condition [1]

Cycle isolation losses are losses that will accumulate over the operating cycle and since leaks typically increase due to valve erosion, it will not be a linear accumulation. The losses can be calculated by using Eq. (4).

$$Loss\left(\frac{currency}{month}\right) = Leakage\ loss\ (MWe)\ \text{x}\ time\ \left(\frac{hours}{month}\right)\text{x}\ revenue\left(\frac{currency}{MWh}\right) \quad (4)$$

Losses caused by leaking valves may vary depending upon where in the turbine cycle the leaks occur as shown in Figure 15. Once the steam has passed through the LP turbine, most of the available energy has been used to generate electricity. A leak at this level in the cycle will not have the impact of a main steam leak upstream of the HP turbine. This is because fluid lost in this level is not available for electricity production. Thermodynamic modelling software can be used to calculate the effect of a leak at specific locations in the cycle. Once the leakage effect at a specific area in the cycle is known it can be used for all the valves at that point in the cycle.

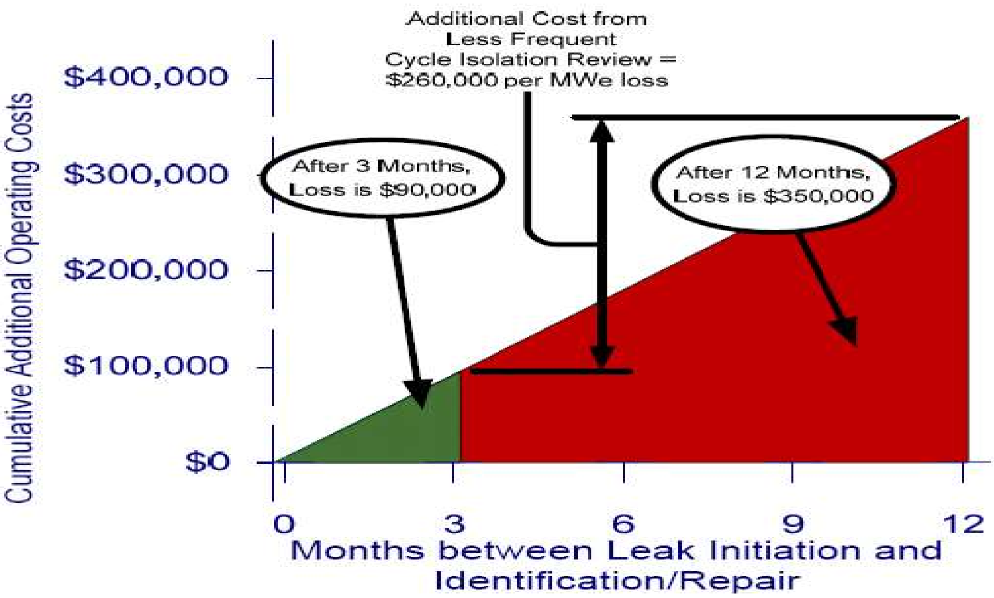


FIG. 15. Cycle isolation losses [1]

3.4.2. Establishing a cycle isolation program

What valves need to be monitored?

There are various leakages to be considered and the focus of the cycle isolation programme needs to be consistent throughout the entire cycle and include internal, external and bypass leakage as shown in Figure 16 on a pressurized water reactor (PWR) example schema.

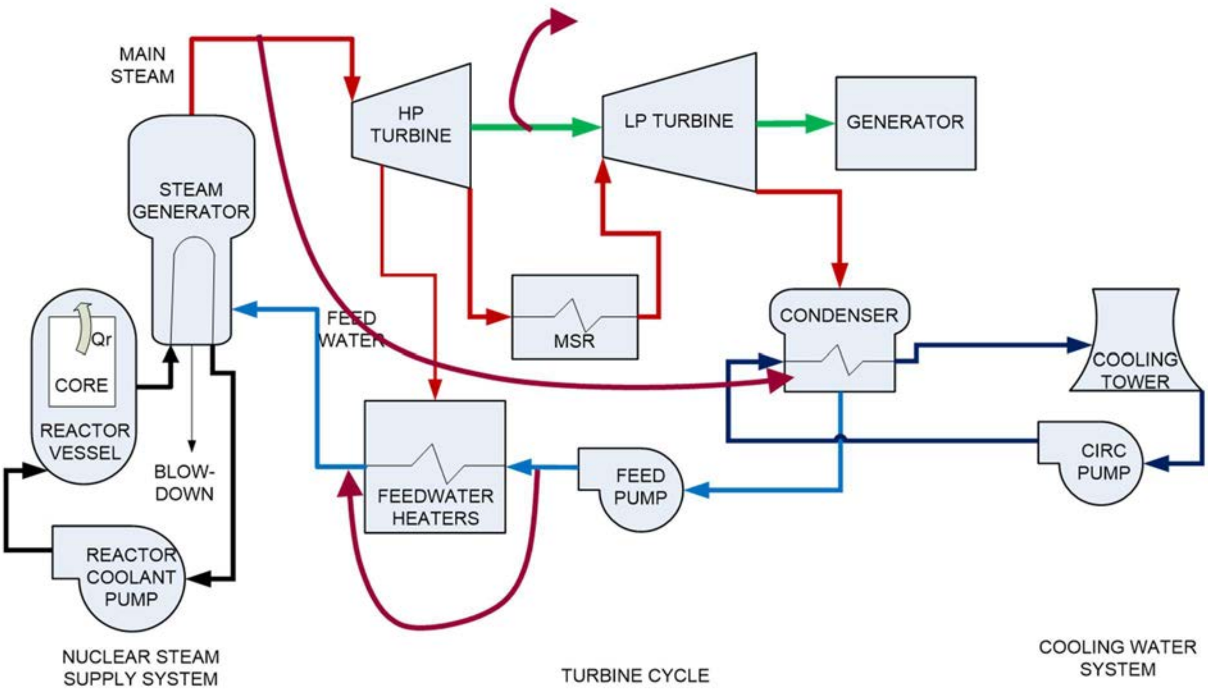


FIG. 16. Cycle isolation leakages for PWR schema [1]

In order to determine the scope of the cycle isolation programme a review of the plant documentation including piping and instrumentation diagrams and thermal kits need to be available.

The system piping and instrumentation diagrams for review include, but is not limited to the following:

- Main steam;
- Reheat steam;
- Auxiliary steam;
- Feedwater;
- Steam sealing;
- Extraction steam;
- Blowdown;
- Condensate;
- Feedwater heater vents and drains;
- Steam generator.

The valves to include in the programme are normally closed and isolate high-energy fluid from a lower energy sink such as a condenser or blowdown tank. It is typically not necessary to monitor valves that vent to atmosphere or floor drains since the leaks can be identified visually or by drain flow measurements for boiling water reactors (BWRs). The same methods described below can be used to calculate flow from vents and drains but the sink would be to atmosphere.

This list is just an example of the types of valves that are advised to monitor.

- Turbine bypass valves;
- Feedwater heater emergency drain valves;
- Main steam line drain valves;
- Gland seal unloader valve;
- Feedwater heater vent valves (if normally closed);
- Gland steam isolation valves;
- Extraction steam line drain valves;
- Heater bypass valves;
- Feed pump recirculation valves;
- Before and after seat drain valves;
- Steam drain line orifices (and orifice bypass valves);
- Relief/safety valves;
- Steam traps/trap bypass valves.

Once the valves have been identified, their location in the cycle need to be assessed (see Figure 17) by reviewing the plant drawings and heat balance diagrams. This review is to evaluate the energy impact these valves have on the cycle and prioritize which valves is best to monitor based on the effect of the leaking valve on plant efficiency. The purpose of following figure is to indicate areas of the cycle that should be monitored for cycle isolations (indicated by blue/red dots).

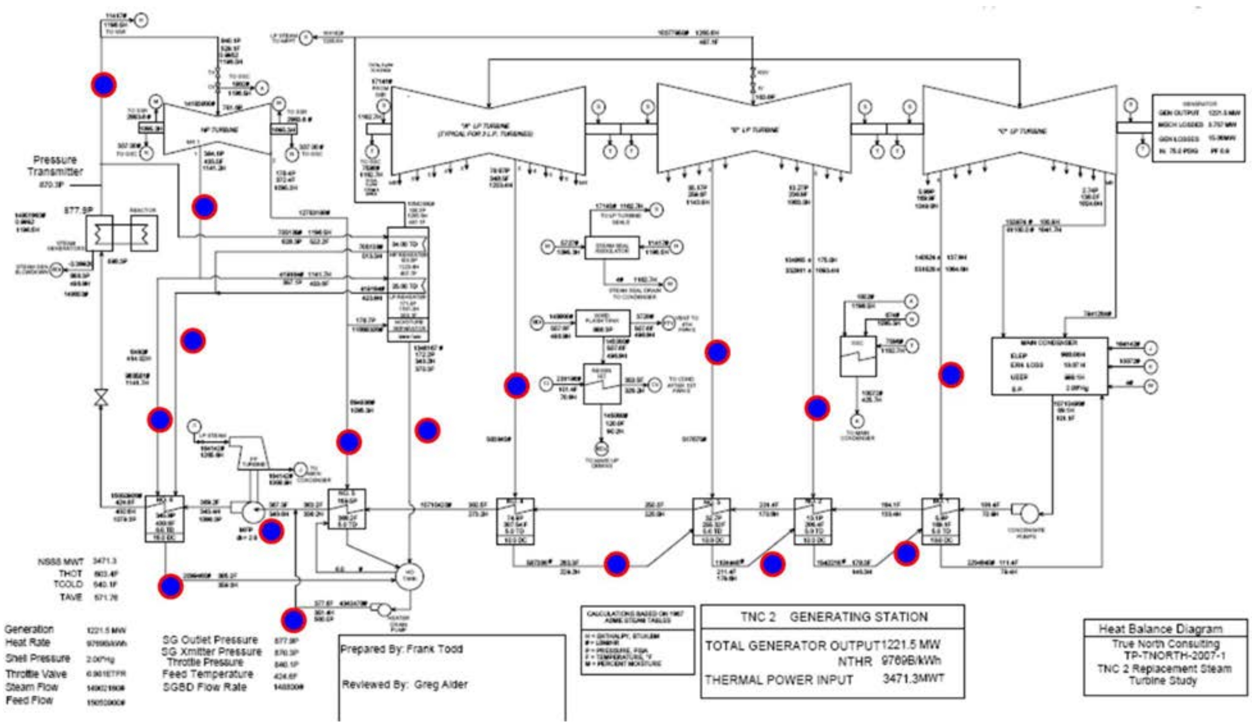


FIG. 17. Cycle isolation leakage valve detection

Once the valves are identified the valve information is collected to aid in determining common valve type leakages and comparisons as shown in Figure 18.

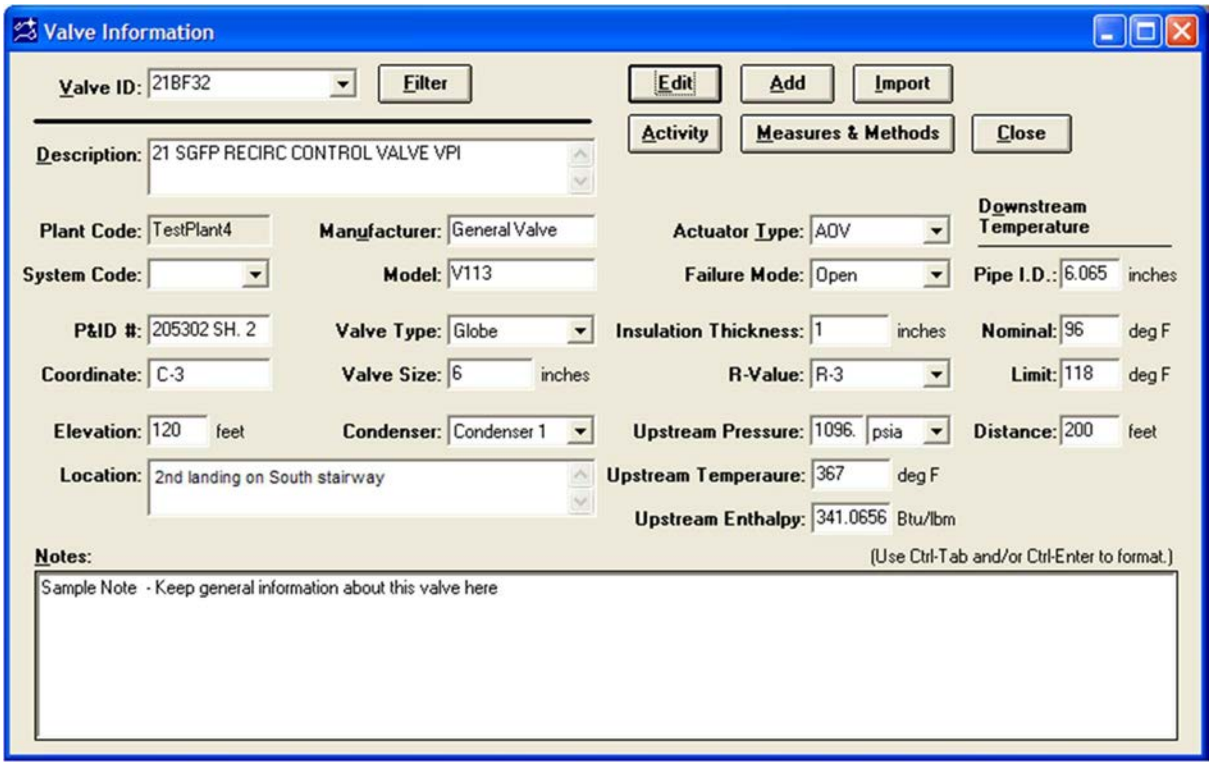


FIG. 18. Cycle isolation leakage data collection

3.4.3. Collection of data

3.4.3.1. Location of temperature measurements

For the data to be reliable to determine leakage, measurement locations need to be well defined, displayed in the plant and consistent. A good practice is to measure the temperature at least 20 pipe diameters downstream of the valve. This is to ensure that the measurement is indicative of the fluid inside the pipe and not the result of conduction through the pipe from the hot side of the valve. This may be difficult to obtain due to the actual configuration in the plant. There may not be 20 pipe diameters between the valve and the condenser.

3.4.3.2. Record temperatures

Temperatures can be measured using permanently installed thermocouples (the best approach), IR thermometers or contact pyrometers. Contact pyrometers are the least recommended approach because it affects the consistency of the measurements based on the relative patience of the one measuring the temperature.

At BWR plants permanently installed thermocouples are recommended since access to most measurement locations is not possible during operation. They are advised to be spring loaded to insure proper contact with the pipe surface. In a PWR if no permanent temperature devices are installed, clearly marked locations need to be identified and holes cut in insulation to allow access. If IR is to be used, the size of the hole is recommended to be such to allow the spot size of the IR thermometer to read only the pipe temperature, not the surrounding locations.

Typically, a 2.5 to 3.75 millimetre diameter hole is enough. When using an IR thermometer, the pipe surface needs to be painted flat black with high temperature resistant paint.

Since IR thermometers use emissivity to determine temperature, ensuring the measurement points have a consistent emissivity is essential. When using the IR thermometers, the operator's technique can influence the results. The best method is to get as close to the pipe as possible and swivel the head of the IR gun. This will allow the thermometer to capture the entire surface under consideration. The maximum temperature reading of the device would be the temperature that is recorded using this method.

It is recommended to observe the surrounding areas for high temperature surfaces. There is a relationship between the diameter of the measurement area and the distance from the IR measuring device. The further away the larger the diameter of the measured area. The lenses of the IR device need to be such that this effect is minimized. If the IR thermometer is not reading directly on the pipe, it can pick up the temperature of the wall behind it and give a false reading as shown in Figure 19.

It is advised to record measurements for all valves defined in the programme at a minimum frequency of quarterly. Post outage measurements need to be taken to verify that the valves opened for a plant shutdown or start-up have been re-seated. In addition to the temperature measurement the distance from the valve to the measurement location, distance away from the pipe when using IR and the distance from the measurement location to the sink needs to be recorded.

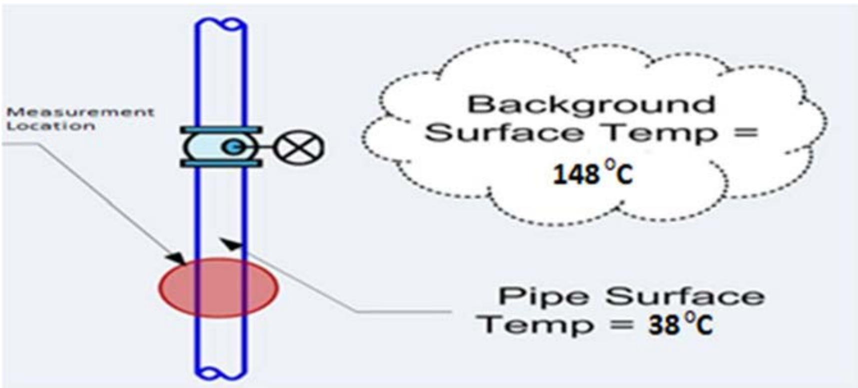


FIG. 19. IR thermometer false reading condition [1]

3.4.4. Methods of analysis

It needs to be stated that all the methods described below are based on various assumptions that need to be validated when performing the calculations. At best these can be an estimated flow. The best method is to isolate the valve while measuring plant parameters, however most of the time the leakage effects are below the ability to accurately determine a loss.

3.4.4.1. Temperature methods

As a valve isolating a high energy liquid leaks the temperature downstream of the valve will increase; the larger the leak the higher the temperature. Due to isenthalpic expansion the temperature downstream of the valve will be lower than the temperature upstream of the valve. However, there is a relationship between the amount of leakage and the downstream temperature of the valve.

Pipe surface temperature as opposed to internal temperature is the most common method of obtaining the reading. If measured manually this is facilitated by a hole cut in the insulation to allow access to the pipe. Some plants have a permanent thermocouple or resistance temperature detectors installed under the insulation. It is true that there will normally be a difference between the pipe surface temperature and the liquid temperature in the pipe. Therefore, the method may sometimes underpredict the actual leakage. Since the methods are not precise, they are considered estimates and are used to prioritize leakage.

The pressure inside the pipe is calculated using the steam tables for the measured temperature at a saturated condition. Most NPPs operate in the saturated region and therefore apart from hot reheat the leakage will not be superheated. Additionally, the superheat is not very high so even the superheated leakage may be close to saturation conditions at the choke point downstream of the valve.

There are many assumptions that go into these calculations and some are very difficult to verify. Engineering judgment along with peer reviews need to be employed when using the results of these calculations. Exelon and other institutions in the United States are starting to use wireless thermocouples on the pipes downstream of high energy valves. Such as the use of collector nodes to capture the signals near the device locations which are then sent to a central collection unit.

The accuracy of the temperature measurement techniques varies depending on the actual configuration of the valve, piping and sink (condenser or flash tank). The typical accuracy is at best 10% (verified during tests on the blowdown pipes). Depending on the actual configuration it is a good practice to reduce the MWe predicted by a factor based on the ability to match the assumptions (length/diameter ratio from measurement point to sink, discharge into header with other valves etc..).

(a) Measuring temperature upstream of leaking valve

Measuring temperature just upstream of a valve can, in some cases, give an indication that the valve is leaking. However, upstream temperature is only useful in cases where the valve is separated from the fluid it isolates by a sufficient length of pipe that functions as a ‘dead leg.’ The pipe length needs to be enough to allow a temperature decrease between the valve and upstream of the valve. However, in the presence of a leak, this temperature just upstream of the valve will rapidly approach the temperature of the upstream conditions.

In the Figure 20 the relationship between leakage and pipe temperatures at various locations along the pipe is shown.

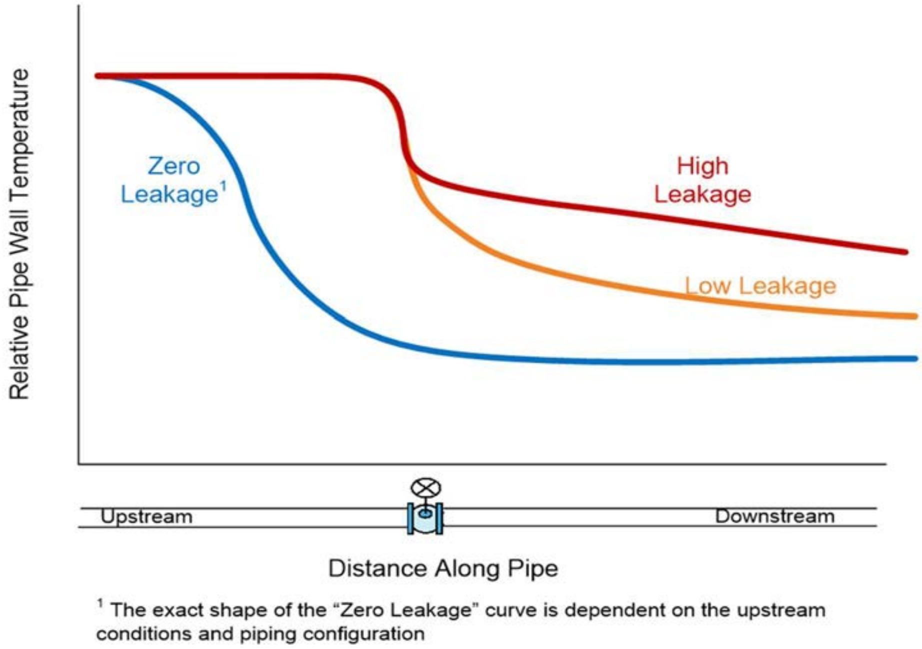


FIG. 20. The relationship between leakage and pipe temperatures

(b) Measuring temperature downstream of a leaking valve

Measuring the temperature upstream of a valve is not enough to determine the magnitude of the leak. Five different methods are described here to determine the magnitude of the leak based on temperature measurement downstream of a valve. All these methods are based on determining the pressure inside the pipe from the downstream temperature. With this pressure and the pressure of the sink the flow can be estimated using the methods indicated below.

Below are examples of some of the calculation methods available for estimating leakage. These all use the measured temperature to obtain the pressure at the measurement location.

Except for the ASME Figure 14[1] of ASME Steam Tables [9], these methods calculate a velocity in the pipe. Equation (5) below is used to determine the mass flow:

$$W = \rho \cdot A \cdot V \tag{5}$$

Where W is mass flow rate (kg/s), ρ is density (kg/m^3), A is a cross section inside area of the pipe (m^2) and V is velocity (m/s).

— Darcy [10]

This calculation method for estimating leakage is shown in Eq. (6):

$$h = K \cdot \left(\frac{v^2}{2g}\right) \tag{6}$$

[1] ASME Figure 14 of ASME Steam Tables [9] is based on English units.

Where h is head loss in the pipe (i.e. pressure change) (m), K is resistance coefficient (-), equal to the Moody friction factor times the length of pipe divided by the diameter (f L/d), v is mean velocity (m/s) and g is acceleration due to gravity (m/s^2).

This derivation does not include an expansion factor to account for the fact that the fluid under consideration is steam, a compressible gas. The expansion factor formula and its application are derived in Fluid Meters [11] or can be read off a table in the Crane manual [12] using the total pipe resistance coefficient (K).

The two previous Eqs (5, 6) can be developed into the following Eq. (7):

$$W = 126350 \cdot Y \cdot d^2 \cdot \sqrt{\frac{\Delta P}{K} \cdot \rho} \tag{7}$$

Where W is flow (kg/h), Y is and expansion factor (-), d is pipe internal diameter (m), ΔP is differential pressure from downstream of valve to sink (kPa), K is resistance coefficient (-), equal to the Moody friction factor times the length of pipe divided by the diameter (f L/d) and ρ is density of fluid downstream of the valve (kg/m^3). The expansion factor Y is inserted as a multiplier to account for compressible flow.

— Sonic flow

The Sonic Equation method assumes there will be choked flow in the pipe. The speed of sound in the fluid is the limiting factor for sonic flow. The equation for choke velocity of a compressible fluid in a pipe is shown as below [13]. This equation assumes isentropic flow of an ideal gas. The expression can be found in Eqs (8, 9):

$$V_{choke} = \sqrt{\frac{k \cdot g \cdot 0.102 \cdot P'}{\rho}} \tag{8}$$

$$P' = (P_1 - \Delta P_{sonic}) \tag{9}$$

Where V_{choke} is choked fluid velocity (m/s), k is ratio of specific heats (-), g is acceleration due to gravity (9.8 m/s^2), ΔP_{sonic} is limiting pressure drop that causes choked flow - this value is determined with Crane table using K (total pipe resistance) (Pa), P_1 is pressure inferred with downstream temperature measurement (Pa) and ρ is density of fluid downstream of valve (kg/m^3).

— Grashof

The Grashof equation is a formula to estimate a discharge flow of saturated steam through a nozzle in terms of a reservoir pressure P and a flow area A. The expression shown in Eq. (10):

$$W = 0.641 \cdot A \cdot P^{0.97} \tag{10}$$

Where is W mass flow rate (kg/h), A is discharge flow area (cm^2) and P is reservoir pressure (kPa or mbar).

Applying the Grashof equation to a leaking valve requires that the pipe be treated as a nozzle from the point where the downstream temperature is taken to its finishing point in the sink.

In reality a length of pipe will have significantly more flow resistance due to friction and flow geometry than a nozzle. The Grashof equation does not consider any flow resistance and a correction is necessary. Derivation of this equation is described in [14].

— Choked flow

The Choke Equation method is similar to the Sonic Equation. It assumes there will be choked flow in the pipe, which means the velocity of the fluid will be limited by the speed of sound in the fluid. The expression is shown in Eq. (11):

$$V_{choke} = \sqrt{k \cdot g \cdot R \cdot T} \tag{11}$$

Where V_{choke} is choked fluid velocity (m/s), k is ratio of specific heats (-), g is acceleration due to gravity (9.8 m/s^2), R is individual gas constant (0.167226 J/kg K for steam) and T is temperature (°K).

Because isentropic flow of an ideal gas is assumed, the following Eqs (12, 13) for the critical pressure and temperature ratios can be applied:

$$\frac{T_0}{T^*} = \left(1 + \frac{k-1}{2}\right) \tag{12}$$

$$\frac{P_0}{P^*} = \left(1 + \frac{k-1}{2}\right)^{\frac{k}{k-1}} \tag{13}$$

Where P_0 is starting pressure (kPa), T_0 is starting temperature (°K), P^* is pressure at choke conditions (kPa), T^* is the temperature at choke conditions (°K), k is ratio of specific heats (1.3 for steam, constant for isentropic flow), P_0/P^* is12.623 and T_0/T^* is 0.638.

The value of T^* can be solved using the measured temperature downstream of the valve and P^* can be solved with the inferred downstream pressure. These values can be used with equation (7) and (9) to determine mass flow rate.

The ASME Figure 14 calculation method (Figs 21 – 22) is based on ASME Figure 14 from the ASME Steam Tables [9]. The figure shows Critical (Choking) Mass Flow Rate for Isentropic Process and Equilibrium Conditions.

Pressure, enthalpy and flow area are used to determine the critical mass flow rate by ASME Figure 14 of ASME Steam Tables [9]. Enthalpy and pressure would be based on the downstream temperature and the saturated pressure for that temperature.

— ASME Figure 14[2] [9]

Flow Function ('w/p) Lookup Table (Figure 14 of ASME Steam Tables)														
Pressure (psia)			0.2	0.5	1.0	2.0	5.0	20.0	50.0	100	200	500	1000	2000
Mass Flow rate (Lbm/HrIn^2Psi)			4	5	6	7	8	9	10	11	12	13	14	15
Inlet Enthalpy														
600	4		82.2	81.7	81.2	80.6	80.6	80.7	80.7	80.7	86	92.9	106	
650	5		79.2	78.4	77.8	77.3	77.3	76.8	76.8	76.8	80.1	85	92.9	
700	6		76.3	75.3	74.7	74.2	73.8	73.2	73.2	73.2	74	78.4	84.1	95
750	7		73.8	72.8	72.2	71.5	70.6	69.8	69.8	69.8	70.7	73	77	84.7
800	8		71.3	70.4	69.7	69	68.2	67.3	66.8	66.8	67	69	71.3	77.3
850	9		69.2	68.2	67.5	66.8	65.8	64.8	64.2	64.2	64.3	65.4	67.5	71.4
900	10		67.2	66.2	65.5	64.8	63.8	62.6	61.8	61.7	61.9	62.3	63.7	66.7
950	11		65.4	64.4	63.7	62.9	61.9	60.6	59.8	59.3	59.4	59.7	60.6	62.9
1000	12		63.7	62.8	62	61.2	60.2	58.8	57.9	57.3	57.3	57.4	57.9	59.8
1050	13		62.2	61.2	60.4	59.6	58.6	57	56.2	55.6	55.2	55.7	55.7	56.9
1100	14		60.8	59.8	58.9	58.2	57.2	55.6	54.6	54	53.7	53.4	53.4	54.5
1150	15		57.5	57.5	57.5	57.7	56.3	54.1	53.2	52.6	52.1	51.7	51.7	52.5
1200	16		53.4	53.4	53.4	53.4	53.4	53.4	52.2	51.3	50.6	50.2	50.2	51.2
1250	17		49.8	49.8	49.8	49.8	49.8	49.8	49.8	49.8	50	50	50	50.2
1300	18		47	47	47	47	47	47.4	47.5	47.5	47.5	47.6	47.6	47.6
1350	19		44.6	44.6	44.6	44.6	44.6	44.9	44.9	44.9	44.9	45.2	45.2	45.2
1400	20		42.5	42.5	42.5	42.5	42.5	42.8	42.8	42.8	42.8	43	43	43
1450	21		40.8	40.8	40.8	40.8	40.8	41.1	41.1	41.1	41.1	41.2	41.2	41.2
1500	22		39.3	39.3	39.3	39.3	39.3	39.5	39.5	39.5	39.5	39.6	39.6	39.6
1550	23		37.8	37.8	37.8	37.8	37.8	38	38	38	38	38	38	38
1600	24		36.6	36.6	36.6	36.6	36.6	36.7	36.7	36.7	36.7	36.7	36.7	36.7
1650	25		35.7	35.7	35.7	35.7	35.7	35.7	35.7	35.7	35.7	35.7	35.7	35.7
1700	26		34.7	34.7	34.7	34.7	34.7	34.7	34.7	34.7	34.7	34.7	34.7	34.7

FIG. 21. Calculation methods available for estimating leakage [1]

Mass flow rate using ASME Figure 14[3] of ASME Steam Tables [9] shown in Eq. (14):

$$W = W_{ASME} \cdot A \cdot P \tag{14}$$

Where W ismass flow rate (lbm/h), W_{ASME} is value taken from ASME Figure 14 (lbm/h)/(in^2 psi), A is flow area (in^2), P is pressure downstream of valve (psia).

[2] ASME Figure 14 of ASME Steam Tables [9] is based on English units.
[3] ASME Figure 14 of ASME Steam Tables [9] is based on English units.

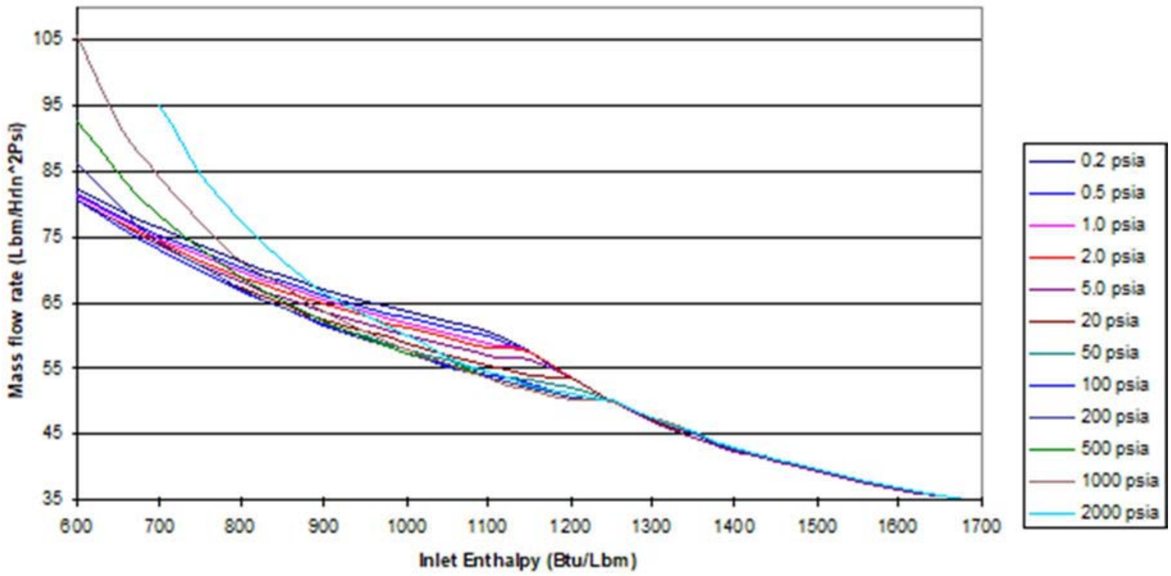


FIG. 22. Calculation methods available for estimating leakage [1]

3.4.4.2. *Corrections to the flow measurements*

— Hydraulic flow resistance

The Bernoulli equation (Eq. (15)) can be used to determine the flow resistance in the pipe which can be used to correct the calculated mass flow.

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 + \sum K \cdot \frac{V_2^2}{2g} \tag{15}$$

$$V_2 = \sqrt{a} \cdot (1 + \frac{K}{2})^{-0.5} \tag{16}$$

$$Flow\ resistance = (1 + \frac{K}{2})^{-0.5} \tag{17}$$

Solving the equation for V_2 an Eq. (16) to be used (where a accounts for the other terms in the Bernoulli equation). The flow resistance is accounted for by the separate equation Eq. (17).

Where P_1 is pressure upstream (kPa), V_1 is velocity upstream (m/s), ρ is density of fluid downstream of valve (kg/m^3), g is acceleration due to gravity (9.8 m/s^2), z_1 is elevation upstream (m), $P2$ is pressure downstream (kPa), V_2 is velocity downstream (m/s), Z_2 is elevation downstream (m), K is resistance coefficient (-).

— Moisture Correction

The five flow equations described in the previous sections only calculate dry steam flow. In most cases there is moisture present that needs to be considered. The relationship between total leakage flow and its steam and liquid constituents can be evaluated with conservation of mass and energy Eqs (18, 19):

$$W_T = W_f + W_g \tag{18}$$

$$W_T \cdot h_T = W_f \cdot h_f + W_g \cdot h_g \tag{19}$$

Where W_T is total leakage flow (kg/h), W_f is fluid component of leakage flow (kg/h), W_g is gas component of leakage flow (kg/h), h_g is downstream gas enthalpy (kJ/kg), h_f is downstream liquid enthalpy (kJ/kg) and h_T is upstream enthalpy (kJ/kg).

Evaluating these two equations yields Eq. (20):

$$W_f = W_g \cdot \frac{h_g - h_f}{h_T - h_f} - W_g \tag{20}$$

The total flow through the valve is the sum of the moisture flow and the gas flow per Eq. (21):

$$W_T = W_f + W_g \tag{21}$$

When steam entering a nozzle is wet, the steam (gas) particles exit the nozzle at different velocities. Therefore, there needs to be a correction.

A correction factor can be developed using the equation for flow of wet steam from [15] as shown in the Eq. (22) below:

$$\frac{v}{v_0} = \frac{1}{\sqrt{x + f^2(1-x)}} \tag{22}$$

Where V is actual steam velocity (m/s), v_0 is total velocity (m/s), x is steam quality, f is ratio of water velocity to steam velocity (-).

Rearranging the previous equation to solve for v_0, substituting 1 minus moisture fraction for quality, and assuming a value of 0.15 for f, a curve for correcting the calculated flow can be developed as shown in Figure 23.

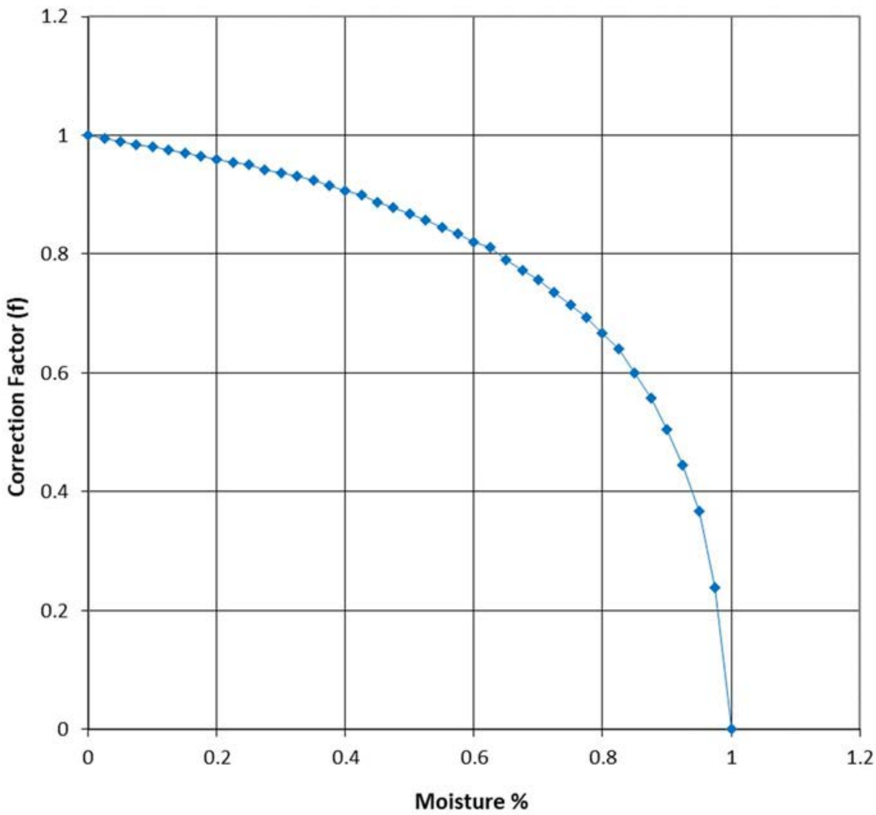


FIG. 23. Curve for correcting the calculated flow [1]

3.4.4.3. Acoustic methods

The basic principle behind the acoustic method of leakage monitoring is the relationship between ultrasonic acoustic signals in a leaking valve and the decay of turbulence resulting from high pressure high velocity fluid flow. The degree of turbulence is predicted by the Reynolds number.

— Signature Comparison Method

The Acoustic Comparison Method uses a single transducer to capture the acoustic signature of the valve. A baseline needs to be obtained for the valve in a leak tight condition. For this to be successful, the valve needs to have the ability to be isolated so that the differential pressure across the valve is removed. In the isolated condition, the baseline acoustic signature is obtained. When a leak is suspected the acoustic signature can be obtained with the valve leaking. Comparing the isolated and the leaking acoustic signatures provides information on the magnitude of the leak.

— Differential Signature Method

This method uses transducers placed upstream of the valve, downstream of the valve and at the valve. The upstream and downstream readings are used to detect ambient background noise. The reading at the valve is compared to the background readings, and a positive difference indicates the presence of a leak.

— A,B,C,D,E Method

As another form of the Differential Signature Method, this method compares the acoustic signatures at five locations along the piping in which the valve is located. There are additional measurement points upstream and downstream of the valve which provides a basis for eliminating background noise.

— Direct Comparison Method

The Direct Comparison Method compares identical valves to each other without a comparison to a reference or background. The principles for this method are the same as those for the other acoustic methods mentioned, but only signals within the group of identical valves are compared. The assumption in this method is that the background noise for all valves in the group are identical. Therefore, it is important to verify this assumption when using this method. typical analysis methods

3.4.5. Estimation of plant impact

All leaking valves do not affect the plant in the same way. Once the location in the cycle is established, the impact of a leak needs to be evaluated. A factor that converts a leakage flow rate to a MW electric value is a method to assess the overall impact of the leak. The process for developing this loss factor is listed below:

- Simulate a leak in a thermal model. Using plant thermodynamic modelling software, create a small (4500 kg/h or 1% of the flow in that line) flow from the location in the cycle to the sink where the leak would go. Run the model simulation and record the generation with the leak.

- Calculate the power of the leak. Multiply the upstream enthalpy times the modelled flow rate to calculate the energy lost from the leak (the effect of the sink enthalpy will be encompassed by the model results).
- Calculate the loss factor. Divide the difference in baseline generation and generation with the leak by the power from the leak. This is the loss factor that can be multiplied by the product of the calculated flow and the enthalpy at the sink to determine the loss in generation.

Based on these methods the heat rate effects can also be determined. An example output based on these calculations is shown below in Figure 24.

Valve ID	Description	Type	Size (inches)	Flags[1,2]	Temperature Limit (deg C)	Downstream Temperature (deg C)	Average Leakage (kg/hr)	Average Loss (MWe)
2BD-11A	Primary II Superheater Outlet Header Drain - West Side		1.25		65.6	250.6	436.5	0.1
2BD-11B	Primary II Superheater Outlet Header Drain - East Side		1.25		65.6	249.4	426.2	0.1
2BFW-450	Heater 4/5 Feedwater Bypass Valve		12		999.0	166.1	***	***
2BFW-610	Heater 6 Feedwater Bypass Valve		12		999.0	153.9	***	***

FIG. 24. Results of process of developing loss factor

3.5. TYPICAL ANALYSIS METHODS FOR MONITORING PLANT PERFORMANCE

3.5.1. Performance test code method

3.5.1.1. Test methodology

One of the most common engineering tools used for performance analysis of the overall NPPs and their key components is conducting performance tests according to the ASME PTC. The basic concept of the code test is measuring a performance parameter of a test target and then correcting it for affecting variables external to the test boundary. In other words, the object of the code tests is to determine the expected performance parameters when the external affecting variables are operated at the base reference conditions.

For example, if the overall turbine cycle performance is the test target, no correction for turbine cycle internal components is required and the measured generator power output is corrected for the variables entering and leaving the turbine cycle. If the test target is scaled down to the steam turbine, the test boundary includes the turbine proper and its subsystems, such as the HP and LP turbine sections with the steam admission valves. The measured generator output is supposed to be additionally corrected for performance of the feedwater heating system and MSR. This is because if the performance level of these component deviates from the base reference conditions, turbine extraction steam flow will be affected and turbine shaft power will be changed in the long run.

(a) Full scale test and alternative test in ASME PTC 6 [3]

The ASME PTC 6 [3] is the most widely used technical guidelines for performance testing of steam turbines and their cycle. This code presents two types of test methods, full-scale test and alternative test.

The full-scale test method requires extensive thermal cycle measurements and heat balance calculation. This is because the measured generator power output is corrected for all affecting variables external to the steam turbine test boundary.

Performance parameters of all the turbine cycle components need to be known, and the measured generator power output is corrected for their differences from the base reference conditions. As a reward for these efforts, the full-scale test allows performance analysis of every turbine cycle components, such as turbine sections, MSR, feedwater heaters and feedwater pumping system. Accurate performance testing of the condenser is also possible with additional measurement of cooling water temperature. The test results also provide engineering input for the turbine cycle heat balance modelling.

The alternative test method relies on much less measurements because performance correction for the affecting variables is selective and limited depending on their impacts on the test result. This test method relies on test measurements less than the test cycle heat balance calculation and as such makes greater use of correction curves for cycle adjustment (correction) with resultant cost savings over the full-scale test.

(b) Limitations of the ASME PTC 6 [3] full scale test

In case of the nuclear turbine cycle, the steam turbines are predominantly operated at the wet steam region, which is the most significant challenge to conducting the ASME full-scale test. As moisture contents needs to be known to calculate the turbine extraction steam enthalpy, the test cycle heat balance calculation and the component basis performance analysis are not easy to conduct. The ASME PTC 6 [3] suggests several methods to conduct the full-scale test in NPP, such as steam sampling and analysis with tracer technique or heat balance with feedwater heater drain flow measurement. However, these methods are still costly and even impractical from the cost-benefit point of view.

Under these conditions, NPP operators experience difficulties monitoring and trending performance of turbine cycle components and their impact on the electric power output. As a result, they cannot cope effectively with plant anomalies related to the turbine cycle performance.

(c) KNHP's performance diagnostic test method

Under this background, Korea Hydraulic and Nuclear Power (KHNP) developed a performance diagnostic testing programme which allows consistent performance monitoring and trending of the nuclear turbine cycle.

Test measurements basically refer to technical guidelines in the ASME PTC 6 [3] Full Scale test method, but this test programme does not require effort for tracer technique or heater drain flow measurement. Instead this method employs less intrusive data acquisition for moisture content through minimum assumptions. These are based on well-established principles of thermodynamics and steam turbine performance characteristics.

In addition to ASME PTC 6 [3], following ASME Performance Test Codes are additionally used for performance analysis of the turbine cycle components;

- ASME PTC 12.1 Closed feedwater heaters [4];
- ASME PTC 12.2 Steam surface condenser [16];
- ASME PTC 12.4 Moisture separator reheaters [17].

This approach allows the plant performance engineers to calculate the test cycle heat balance and monitor the turbine cycle performance on a component basis. Test results also provide engineering inputs to build and tune the AS-IS BASIS turbine cycle heat balance modelling.

3.5.1.2. Test calculations

The KHNP's performance diagnostic programme provides extensive information about the turbine cycle performance and more practical to conduct relative to the ASME PTC 6 [3] Full Scale test with minimal sacrifice of test uncertainty. Performance parameters of the overall turbine cycles and their key components can be evaluated with the following steps (lettered from (a) to (g)):

(a) Test measurements and test cycle heat balance calculation

The following assumptions are made to determine the turbine extraction steam enthalpy;

- Assume design efficiency level or expansion slope ($\Delta h/\Delta s$) of the HP turbine. If the MSR drain flows are measured and look reasonable, heat balance calculation around the MSR can be used to determine the HP turbine exhaust steam enthalpy.
- Use expansion line curve K&E 1864-31 (dry region) and 127 cm radius curve (wet region) to estimate the turbine extraction stage steam enthalpy as shown in Figure 25.
- Use design moisture removal effectiveness of the LP turbine moisture removal stage to estimate the moisture blowdown from steam path.
- Assume isentropic process for the turbine extraction to the feedwater heaters and the reheaters.

In Figures 26, 27 examples of test measurements and test cycles heat balance calculation from a performance diagnostic test for Korean Standard NPP (OPR1000) are shown.

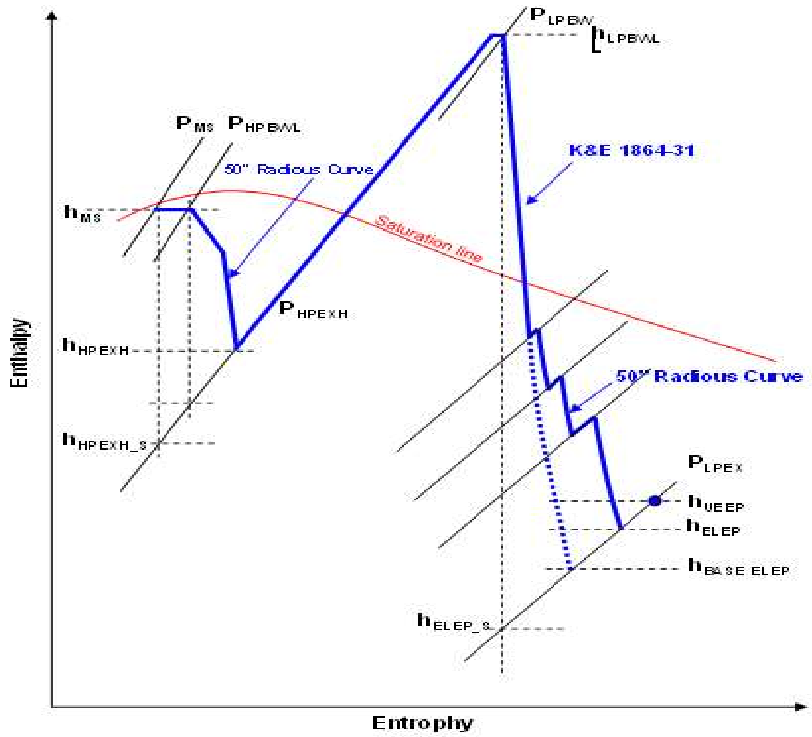


FIG. 25. Typical HP-LP turbine steam expansion line

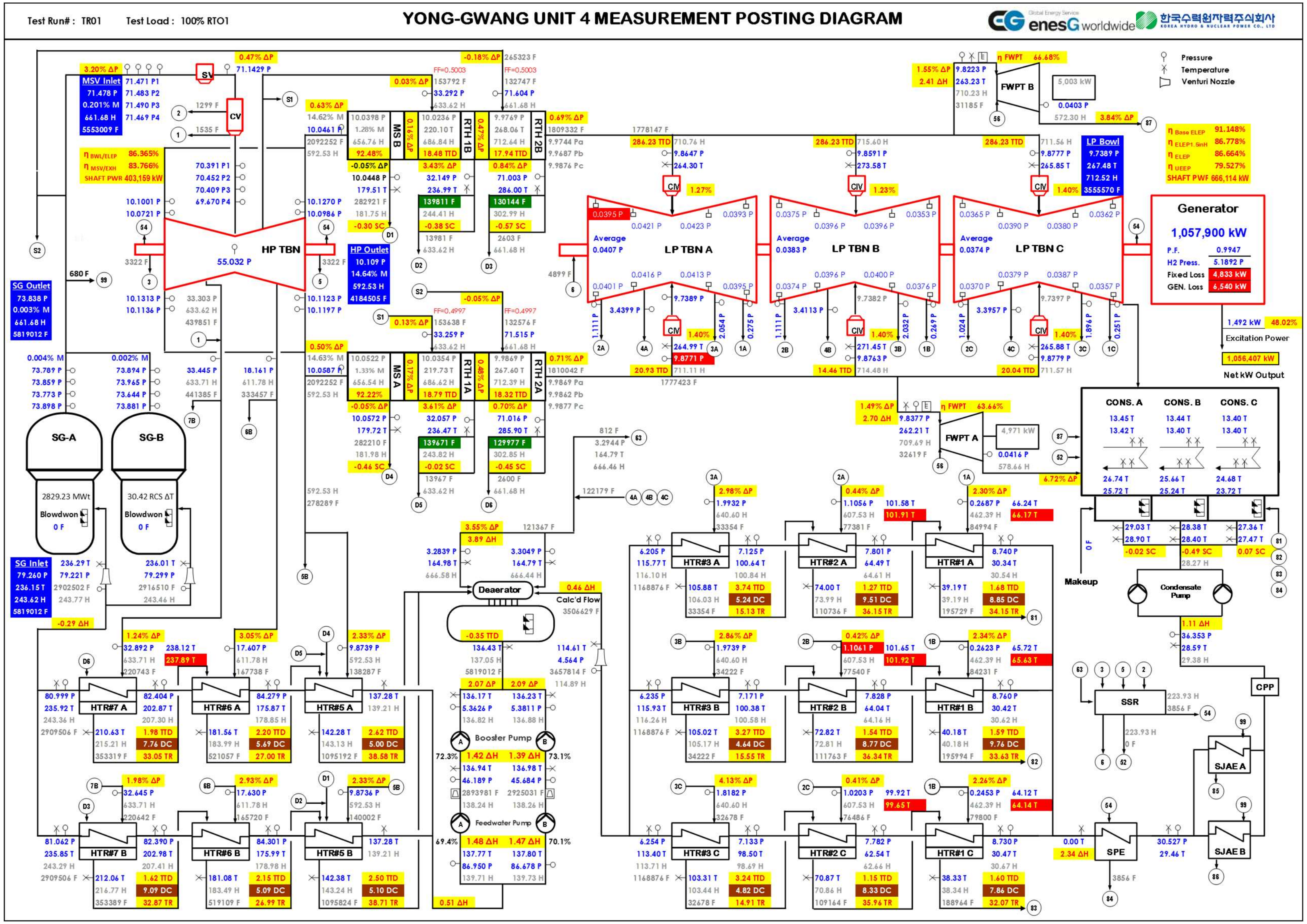


FIG. 26. Test measurements of the performance diagnostic test

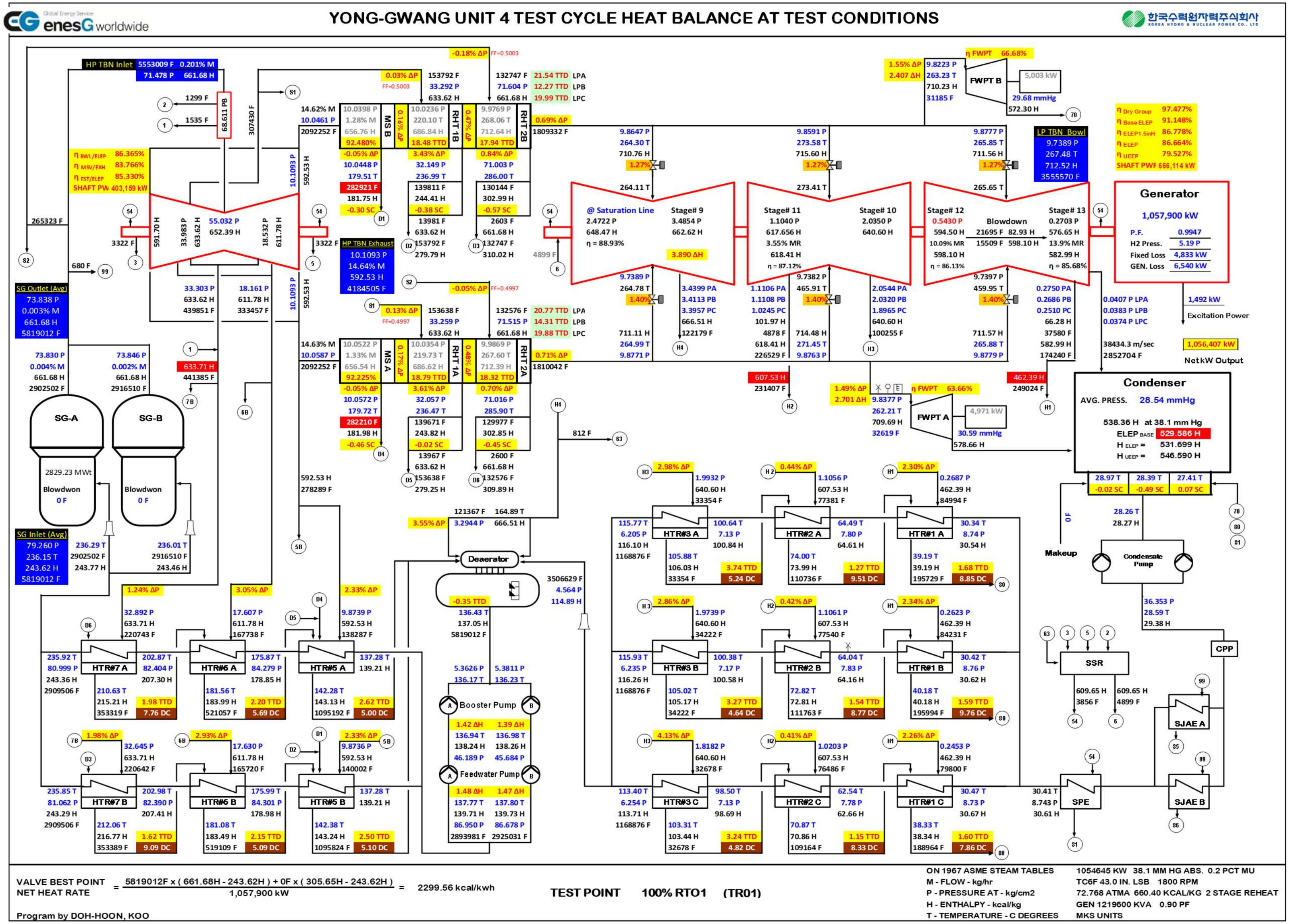


FIG. 27. Test cyle heat balance calculation of the performance diagnostic test

(b) Overall turbine cycle performance analysis

In order to analyse the overall turbine cycle performance, the corrected turbine cycle power output, as explained in Section 5.3, is evaluated using the turbine manufacturer's correction curves.

(c) Steam turbine performance

Following steam turbine performance parameters, explained in Section 5.4, are evaluated from the test cycle heat balance calculation and the ASME PTC 6 [3].

- Overall performance of the turbine (corrected turbine-generator power output);
- HP and LP turbine section efficiencies;
- Steam expansion ratio (pressure ratio);
- Steam flow passing capacity (flow factor).

Most of the nuclear turbine cycle use the ASME PTC 6 [3] alternative test method for steam turbine testing due to restrictions relating to measurement of moisture contents of the cycle steam. This test programme uses the ASME full-scale test method conducting heat balance calculation for the group 1 corrections. This approach is possible through the assumptions explained in (a). Despite these assumptions, the corrected turbine-generator power output is still valid because the same assumptions are applied for both test cycle heat balance calculation and group corrections.

(d) Moisture separator reheater performance

The following MSR performance parameters, explained in Section 5.5, are evaluated from the test cycle heat balance and ASME PTC 12.4 [17].

- Moisture separator effectiveness;
- Reheater TTD;
- Cycle steam pressure drop through MSR.

(e) Feedwater heater performance

Following feedwater heater performance parameters, explained in Section 5.8, are evaluated from the test cycle heat balance and the ASME PTC 12.1 [4].

- Feedwater heater TTD;
- Feedwater heater DCA.

Tube side pressure drop and shell side pressure drop are also one of feedwater heater performance parameters. However, their impact on generator power output is negligible and as such not evaluated in this test programme.

(f) Steam surface condenser performance

The following feedwater heater performance parameters, explained in Section 5.6, are evaluated from the test cycle heat balance and the ASME PTC 12.2 [16].

- Cleanliness factor (CLF);
- Corrected condenser pressure.

(g) Feedwater pumping system performance

Feedwater pumping system performance parameter, that is, feedwater pump turbine (FWPT) driving steam flow adjusted for turbine exhaust pressure, explained in Section 5.9, is evaluated from the test cycle heat balance together with feedwater pump discharge pressure vs. rpm.

3.5.1.3. Thermal performance modelling and verification of test results

To build the thermal performance modelling, performance parameters of turbine cycle components determined from the test cycle heat balance calculation, together with the turbine manufacturer's design information, are used as input data.

The followings are typical performance parameters internal to the turbine cycle;

- Turbine performance parameters
  - HP turbine expansion line efficiency (use design efficiency level);
  - LP turbine expansion line efficiency;
  - Steam flow passing capacity (flow factor) at turbine inlet and extraction stage;
  - Shaft packing and valve stem leakage flows (use design value if not measured);
  - LP Turbine last stage blade annulus area and exhaust losses (use design value);
  - Turbine-generator mechanical losses (use design value);
  - Generator electrical losses (use design value);
  - Cycle steam pressure drop though turbine admission valves.

- Turbine cycle BOP performance parameters
  - Moisture separator effectiveness;
  - Reheater TTD;
  - Cycle steam pressure drops though MSR;
  - Feedwater heater TTD and DCA;
  - Feedwater heater tube side pressure drop;
  - FWPT driving steam flow adjusted for turbine exhaust pressure or efficiency;
  - Feedwater pump and condensate pump enthalpy rise or efficiency;
  - Feedwater pump and condensate pump discharge pressure;
  - Condenser subcooling;
  - Condenser CLF (if the condenser performance is modelled);
  - Turbine extraction steam line steam pressure drop to feedwater heater and FWPT.

(a) Calculation of the corrected turbine cycle power output

The corrected turbine cycle power output, as determined in Section 3.5.1.2. Step (a) using correction curves, can be also determined from recalculation of heat balance using this modelling. In this case the following turbine cycle external variables, for which the measured generator power output was corrected, needs to be the reference (design) heat balance conditions;

- Throttle steam pressure;
- Throttle steam moisture content;
- LP turbine exhaust pressure or condenser cooling water inlet temperature;
- Generator power factor;
- Steam generator thermal power (MW).

(b) Calculation of the turbine-generator power output

Once the thermal performance modelling is built, the heat balance calculation for the group 1 correction according to the ASME PTC 6 [3] Full Scale test method is also possible. This is done by just changing the affecting variables external to the turbine test boundary (turbine cycle BOP performance parameters) from the measured values to the reference (design) heat balance values. This can be done using the measured values for the turbine internal variables (turbine performance parameters) and then recalculating the heat balance.

(c) Modelling verification

The corrected turbine cycle power output determined using correction curves is cross-checked with the expected value from the heat balance calculation using the thermal performance modelling. Table 4 shows a sample comparison between these two methods. It is found that the corrected turbine cycle power output determined from the modelling is different by 0.06% maximum from the test result using the correction curves. Through this process, the turbine manufacture's correction curves for the turbine cycle external variables are verified and confirmed simultaneously.

For the corrected turbine-generator power output according to the AMSE PTC 6, test results from the full-scale test (heat balance calculation) and alternative test (correction curves) cannot be directly compared. The full-scale test corrects the measured generator power output for the whole turbine cycle BOP performance parameters. Correction of the alternative test is selective depending on its sensitivity to the generator power output. So, each group 1 correction curve is verified and confirmed separately.

TABLE 4. COMPARISON OF TEST RESULT FOR 1000 MW KOREAN NPP'S PERFORMANCE DIAGNOSTIC TESTS

Corrected turbine cycle power output		**From correction curves (kW) (a)**	**From performance modelling (kW) (b)**	**Difference**	
				(a)-(b)	**[(a)-(b)]/(a)**
HANBIT #3	TR01	1 048 359	1 047 843	-516	-0.05%
HANBIT #4	TR01	1 053 009	1 053 124	115	0.01%
HANBIT #5	TR01	1 043 511	1 043 551	40	0.004%
	TR02	1 043 522	1 043 524	2	0.0002%
HANBIT #6	TR03	1 045 424	1 044 817	-607	0.06%
	TR04	1 045 855	1 045 294	-561	0.05%
HANUL #3	TR01	1 040 770	1 041 244	474	0.05%
	TR02	1 041 051	1 041 515	464	0.04%
HANUL #4	TR01	1 048 146	1 047 882	264	0.03%
	TR02	1 047 832	1 047 729	103	0.01%

TABLE 4. COMPARISON OF TEST RESULT FOR 1000 MW KOREAN NPP'S PERFORMANCE DIAGNOSTIC TESTS (cont.)

Corrected turbine cycle power output		From correction curves (kW) (a)	From performance modelling (kW) (b)	Difference	
				(a)-(b)	[(a)-(b)]/(a)
HANUL #5	TR01	1 043 282	1 043 485	203	0.02%
	TR02	1 043 039	1 043 217	119	0.02%
HANUL #6	TR01	1 042 423	1 042 863	439	0.04%
	TR02	1 042 160	1 042 435	275	0.03%

3.5.1.4. Conclusion

Nuclear steam turbines are operated in the wet steam region. Thus, the moisture content of turbine extraction steam to the feedwater heaters or reheaters needs to be known for the turbine cycle heat balance calculation. The ASME PTC 6 [3] suggests several methods to measure the moisture contents, such as steam sampling and analysis with a tracer technique or heat balance with feedwater heater drain flow. However, these methods are less practical to conduct.

KHNP's performance diagnostic testing programme basically refers to test measurements for the ASME PTC 6 [3] full scale test except some assumptions made to best estimate moisture content of the turbine extraction steam.

This approach allows full calculation of the test cycle heat balance in an easier and more practical way. Relative to the ASME PTC 6 [3] full scale test, performance analysis of the key turbine cycle components is possible through this test programme. Additional test uncertainty caused by using the assumed turbine performance data is inevitable. However, test results are still valid for performance monitoring and trending purpose because of stringent steam turbine performance characteristics used for these assumptions. The thermal performance modelling obtained from this test programme allows the plant performance engineer to more accurately simulate changes in the generator power output. Also, at off-design operating conditions and turbine cycle modification like equipment replacement or unit uprating.

KHNP has been conducting this performance diagnostic testing programme periodically, via a six-year-cycle, for every NPP unit under commercial operation. This is was done to monitor and accurately trend the performance level of the overall turbine cycle and its components. The purpose was also to update the turbine cycle heat balance modelling. Test results are also used to identify and recapture the performance losses cause by non-optimized operation and maintenance work.

3.5.2. Data reconciliation

3.5.2.1. Description of data reconciliation and validation process

Even with the use of precise measurements, operating parameters such as pressures, flow rates and temperatures exhibit deviations from the expected plant values. These deviations may be caused by an accumulation of permissible individual tolerances along the measurement loop or by process related factors. An example of this is the hot leg temperature measurements at some PWR NPPs. Due to the location of the instruments and the stratified flow stream exiting the reactor a deviation between the actual bulk temperature and the measured temperature exists. With the use of the data reconciliation process the measured values can be corrected to the expected process values with suitable means. This is to determine the uncertainty and acceptability of the corrected measurements. By this method the measurement drift can be determined.

In the case described above, whenever the plant is started up from the cold condition a cross calibration of the instruments is performed. As the plant achieves full power level and the process measurements move to their steady state condition influenced by the temperature stratified flow exiting the reactor; deviations of individual measured variables from the normal values can be determined by the data reconciliation method. Data reconciliation methodology employs a combination of statistical procedures and first principle analysis. This combination provides a qualified representation of the measurement system which relies on the determination of estimated values along with their uncertainties from measured variables. It also relies on a system model representing plausible assumptions. The plausible assumptions are based on unknown systematic deviations in measurements arising from instrument installation issues. This results in a large set of redundant variables based on actual measurements and derived from physical properties. The use of these redundant variables and the application of random matrix theory produces an acceptable method of determining the uncertainty of complex systems.

This method can be also used to resolve measurement error problems. One method used is in the [18] which describes the theoretical and practical calculation methods to assure the quality of measurements and evaluate their results for energy conversion within NPPs. Measurement uncertainties are considered by representing material and system variables as measured variables. The true value of the measured variable is superimposed by the sum of random and independent influences along with the sum of the unknown systematic deviations. By application of the central limit theorem, a sum of independent random variable converts into a normal distribution. These variables are combined in a measured variable vector to form an n-dimensional random variable. A key element of this process is to estimate the covariances of the measured variables to enable their stochastic dependencies to be considered. Derived variables from first principals are equated with the measured variables and combined in the vector of the measured variables. The combined uncertainty of the measured values is expressed in an empirical covariance matrix.

The application of the central limit theorem applies due to the treatment of all the variables as random. This results in the application of the statistical certainty of 95% and given at 95% confidence interval. ASME 19.1 A.2 describes a methodology for determining a weighted overall uncertainty whenever a value of a parameter is approximated by several different measuring methods. This methodology includes systematic, random errors, and parameter uncertainties.

The goal of the DVR process is to correct any instrument biases and minimize the random error. Therefore, the other parameter measurements will be corrected by the process. As such, the ability to monitor core thermal power is robust and not as susceptible to single point failures.

Often it has been the role of the TPE to interpret both plant data and thermodynamic modelling results to aid in the evaluation of core thermal power. Changes in personnel and reduced staffing over the years has resulted in an overall decreased ability to perform this traditional analysis. These require a well-maintained, finely tuned thermodynamic model and an engineer with extensive experience in an operating plant. With the current economic climate in electricity production, not all utilities have the resources to support an engineer to gain the necessary experience. Even if the skill is available at a particular plant there is still a disadvantage related to the availability of the engineer to troubleshoot problems. Also, the current modelling/monitoring programmes in use at many plants do not have the statistical algorithms to adjust the plant data correctly.

The measurement values are improved by means of a correction calculation, which applies conditions based on mass flow balances and energy balances. In the case of a splitter with two outputs to the flow measurement, a relationship between the measured variables exists. That will never quite fulfil the physical law of conservation of mass.

This methodology relies on the principle of redundant measurements of which there are two types: functional and hardware. Functional redundancy results from applying the physical properties and relationships between various measurements. Even if there were only two actual flow measurements functional redundancy can be achieved by solving for the third measurement. In Figure 28 the two types of redundancy are shown.

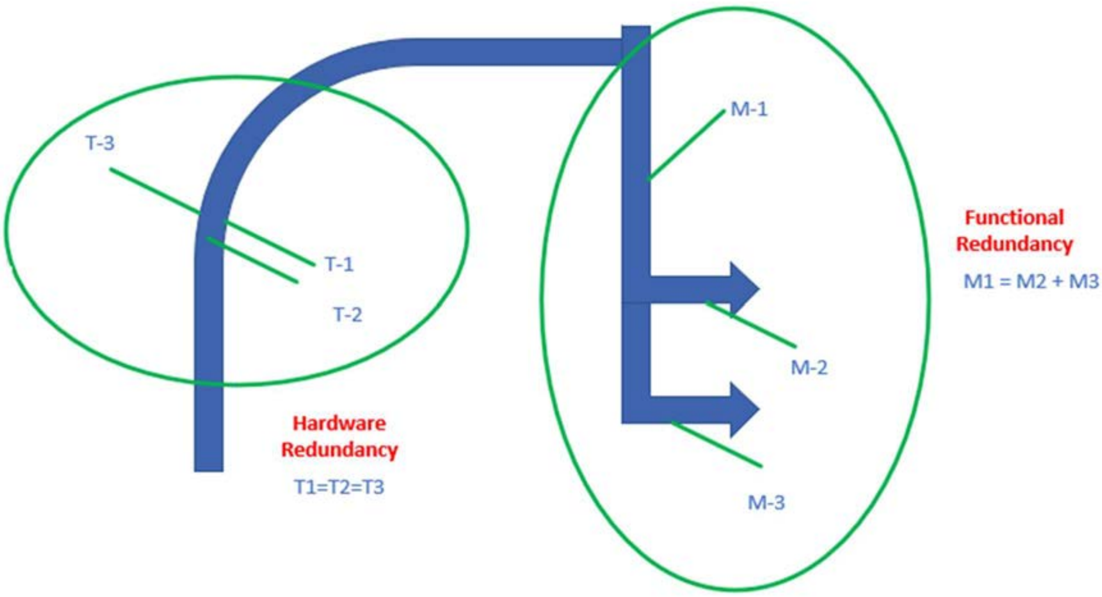


FIG. 28. Two types of measurement redundancies

The measured values of $\dot{m}_2$ and $\dot{m}_3$ (from Figure 28) will not always be balanced due to measurement errors. Therefore, corrections are determined to bring balance to the variables. Accounting for the covariance based on corrections to the data can yield a method to provide a statistically sound correction factor. That can be used to determine a reconciled value and the covariance matrix yielding the minimum measurement uncertainties.

(a) Fitting of the data and quality controls

When the heat balance calculations are performed, contradictory measured values are converted to unequivocal 'true' values for the measured variables. These are to obtain closed mass, energy and materials balances. Differences between the measured values and the 'true' values may be captured with a covariance matrix. The covariance matrix provides a measure of how the measured values and the 'true' values are correlated. The covariance matrix may then be adjusted with an empirical covariance matrix that accounts for the measurement uncertainties and the plausible assumptions made when completing the heat balance calculations (auxiliary conditions per [18]).

With the covariance matrix calculations, the uncertainties of the measurements and the plausible assumptions may be considered using standard statistical methods. In statistical theory, the central limit theorem states that for a large sample size, the sample mean will start to look like population mean. Sampling the distribution of mean will result in a Gaussian distribution. Figure 29 illustrates the relationship between the standard deviation σ and the 95% confidence level of the plant measured values.

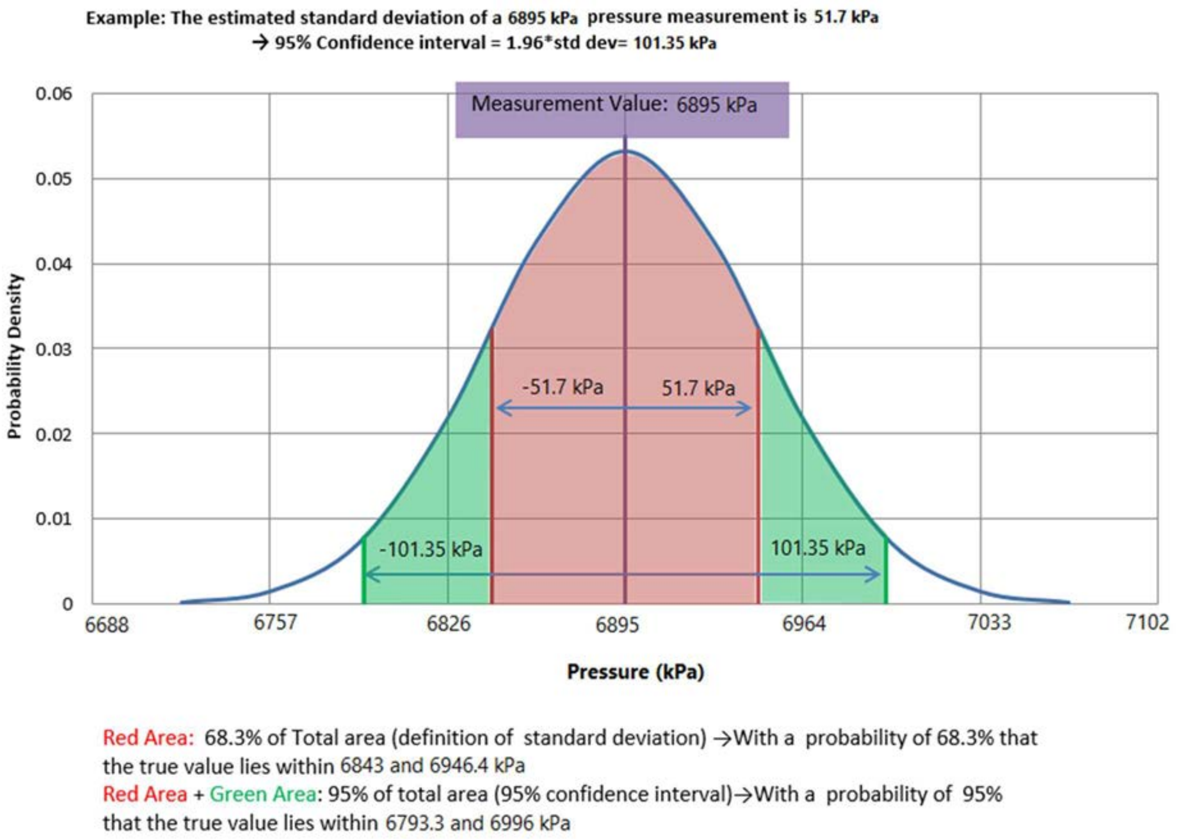


FIG. 29. The relationship between the standard deviation σ and the 95% confidence level of the plant measured values

(Note: 1.96 shown above is dependent on the number of samples and t-distribution. This factor is usually rounded off to 2.95% of the area under the normal distribution lies within 1.96 standard deviations of the mean. 1.96 is the approximate value of the 97.5 percentile endpoint of the normal distribution.)

A Gaussian correction principle is used to remove the inconsistencies, or contradictions between the heat balance estimates, the measured data, and the auxiliary conditions. In simple terms, the Gaussian correction method is very similar to a least squares regression to solve the covariance matrix. The empirical covariance matrix that accounts for the random errors of measurement uncertainties and the plausible assumptions is also considered during the solution.

(b) Overall process quality measurement

One of the advantages of using the data reconciliation process is the incorporation of an objective method to determine the quality of the overall process. Thus, the confidence that can be placed in the results.

The Objective function is defined in Eq. (23). An Objective function is used to determine the overall quality of the system of variables.

$$\text{Objective function} = \sum \left\{ \frac{measured\ value - reconciled\ value}{standard\ deviation} \right\}^2 \rightarrow minimum \tag{23}$$

The CHI-SQUARED which is defined in Eq. (24) is a statistical value for model redundancies. The value of the quality needs to be smaller than 1.

$$Quality = \frac{Objective\ function}{95\%\ quantile\ of\ Chi^2} < 1 \tag{24}$$

If this criterion is not fulfilled, the following errors can be responsible:

- Idealized model is not correct;
- Uncertainties are set as too small;
- Existing suspected measurements.

(c) Quality control of the data reconciliation process

The reconciliation quality is an indicator of the quality of the reconciliation process. The quality is defined as the ratio of the Objective Function and $\chi 2$ as shown in Eq. (25).

$$Quality = \frac{Objective\ function}{\chi^2} \tag{25}$$

(d) Individual parameter quality measurement

Quality control is provided to detect serious errors to evaluate if the measured or estimated values are incorrect and not suitable for reliable results. This measure is defined as a Single Penalty in Eq. (26). The single penalty is a KPI for each measurement.

$$\left| \frac{v_i}{\sqrt{max\left(s_{v,ii}, \frac{s_{x,ii}}{10}\right)}} \right| \leq 1.96 \tag{26}$$

Where v_i is measurement correction (measured value – reconciled value), s_{vii} is variance of correction applied to measurement and $s_{x,ii}$ is variance of measured value. For application in the DVR software the Single Penalty equation is rewritten into an equivalent form.

Both sides of Eq. (27, 28)[4] are squared and the variance terms are converted to uncertainty values.

$$Single\ Penalty = \frac{(measured\ value - reconciled\ value)^2}{\max\left(correction\ uncertainty^2, \frac{measurement\ uncertainty^2}{10}\right)} \cdot (1.96^2) : \leq 3.84 \tag{27}$$

$$correction\ uncertainty^2 = measured\ uncertainty^2 - \ reconciled\ value \tag{28}$$

If any value exceeds the limits imposed by the single penalty equation it is considered suspect and can be removed from the calculation. If a suspected value occurs either the measurement itself is erroneous or the uncertainty assigned to this measurement is too small. By this methodology, any errors introduced by a faulty instrument can be detected and removed. The removal of the instrument will reduce the number of redundancies and thus increase the overall uncertainty. However, a quantifiable basis for removing the instrument is achieved and the overall result is improved.

3.5.3. Data driven methodology

The data driven methodology is a method that incorporates the use of sophisticated computer software using data reconciliation techniques. This is to achieve a high probability that the measurements have been corrected to the most likely value. This information is then used as an input to models which produce performance indices. These can be evaluated to determine plant efficiency and the likely cause of reduced plant efficiency.

3.5.3.1. *Component based key performance indicators*

Process values alone usually give no information about the efficiency of the process quality at a plant. Performance indicators are determined by using multiple process values and give more general information about state or health of cycle and/or equipment.

The best way for the process and equipment monitoring and diagnostics is a combination of the three-level monitoring using methodology from top down and bottom up:

- The most representative thermal cycle parameter;
- The most representative equipment performance parameter; see Section 3.5.3.3;
- Additional information derived from detail analysis using another measurement; it depends on type of equipment, see next sections.

3.5.3.2. Calculation of performance deviation

Nuclear power plant operators maintain efficiency under continuously changing conditions. Key performance parameters serve as general information about the deviation of the process from its optimum and provide a higher level of transparency into the state of equipment health.

To express the deviation of the actual process from its optimum, and to express the actual state of equipment health, two variants of KPI calculation are used as shown in Eqs (29, 30):

---

[4] See Figure 28 for explanation of the 1.96 value.

Variant I

$$KPI = \frac{\eta_{act}}{\eta_{exp}} \tag{29}$$

Variant II

$$KPI = \eta_{act} - \eta_{exp} \tag{30}$$

Actual efficiency and/or effectiveness are calculated according to standards. Regarding the expected efficiency and/or effectiveness there are several possibilities that may be applied, based on:

— Characteristic reference curves usually provided by equipment vendors;
— Historical data; acceptance testing, periodical testing;
— Physical models;
— Data driven model using validated historical data (reconciled data).

The method based on data driven models is the latest approach of expected value calculation. The models replace traditional characteristic reference curve and reflect the multi-dimension KPI dependence on affected quantities.

Note: A very important step in a data driven model building is a causal analysis. A general workflow of a KPI calculation, see Figure 30 below.

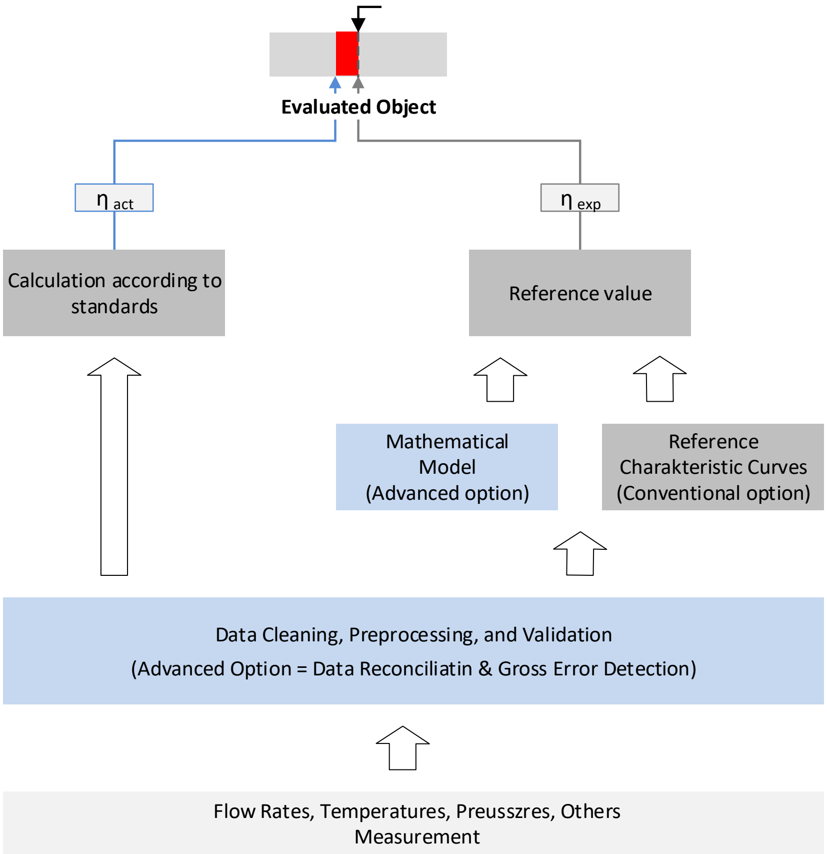


FIG. 30. Calculation of performance deviation

3.5.3.3. Component-based performance analysis

Component analysis enables performance engineers to calculate and trend the equipment health and performance of plant over time. Identifying the origin of any performance deviation. A hierarchical structure of KPI can be used for identifying the origin of performance deviation. The based idea is expressed in a picture below.

Figure 31 shows structure of three level:

— Total plant level;
— Plant area level;
— Component level.

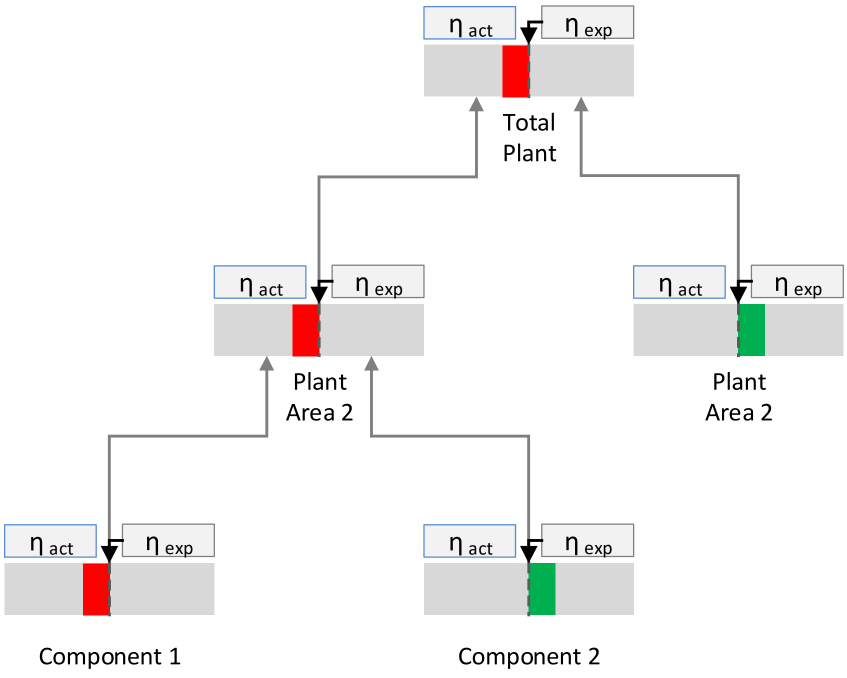


FIG. 31. Identifying the origin of performance deviation

Colours are used to express whether KPI meets an expectation or not:

— Green colour means the process or equipment health is in the expected range;
— Red colour means the process performance or equipment health is not in the expected range.

Applicability of this approach on steam turbine cycle is described in a simplified way on the picture below. The structure of the steam turbine cycle is divided to three levels:

— Unit level;
— Steam turbine, condenser, and cooling tower level;
— Heaters, and pump level.

Each item of the structure is described by its own KPI as shown in Figure 32. The structure can show the origin of expected performance deviation.

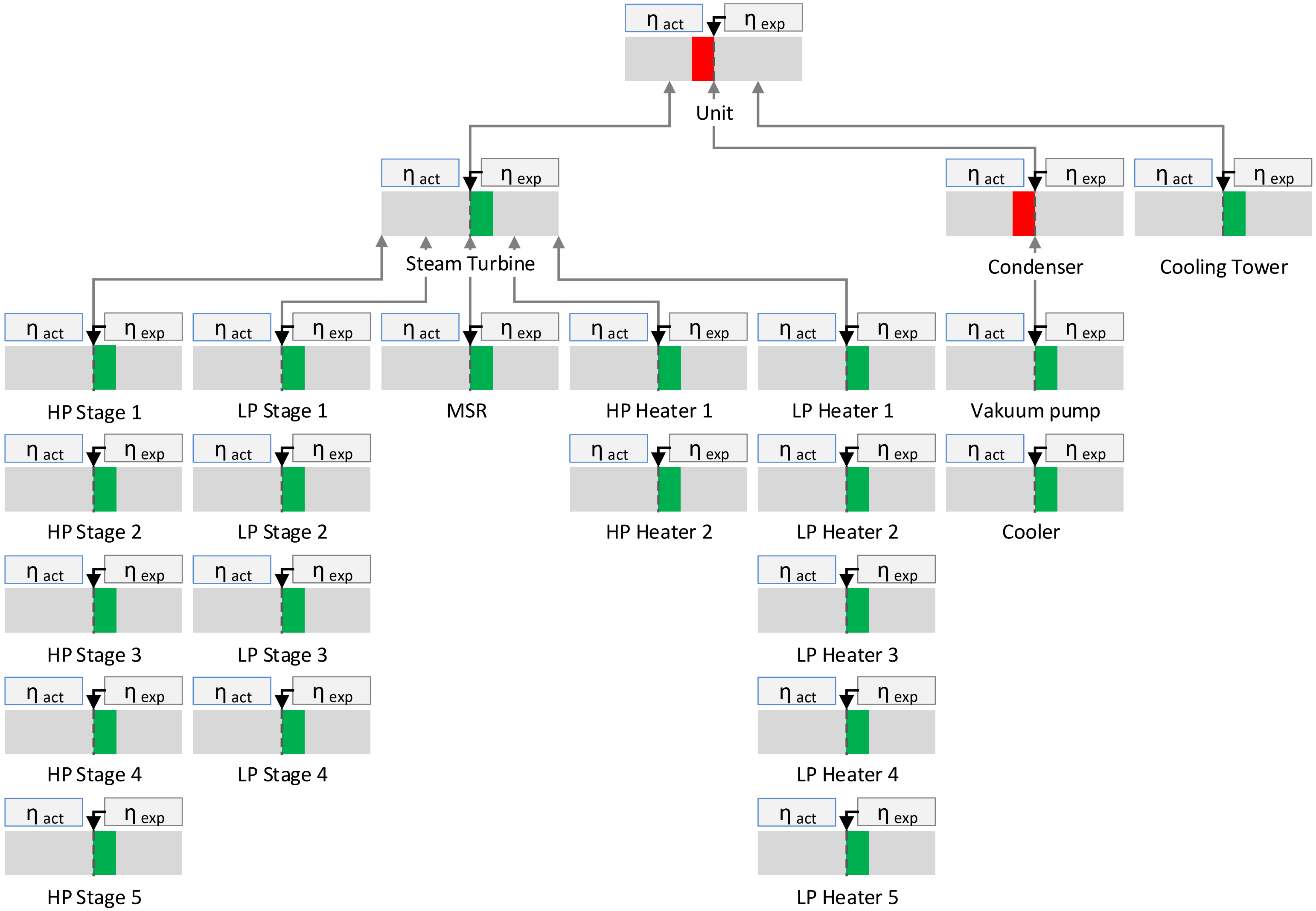


FIG. 32. Identifying the origin of performance deviation – simplified NPP

3.5.4. Comparison with components and plant model

One of the available methods to assess conventional plant performance is to estimate the thermal power that the so-called ‘secondary circuit’ turns into electrical energy. To achieve such a balance with a reliable degree of confidence, the use of a model as a ‘numerical twin’ of the plant brings systematic application of formulas and efficient data treatment. Once correctly tuned, computed values are expected to provide a good representation of main physical quantities and, finally, an image of secondary circuit performance.

Measurements with low uncertainty are therefore needed, to provide the model with accurate values. Data reconciliation can be used to provide these validated measurements however a reliable model with sufficient redundancy is necessary which may require significant resources. The method presented here aims to take benefit from both set of accurate measures and an adjusted model in order to perform a thermal performance evaluation.

The first Section 3.5.4.1 introduces the main principles of this method and its benefits. The second Section 3.5.4.2. describes precisely how to implement the method: what are the requirements in order to use this method, how to gather data and how to carry out the calculation. The third Section 3.5.4.3 explains what kind of analysis and results are expected from this method. Some examples are given to illustrate the theory.

3.5.4.1. Method principles and advantages

(a) Use of simulation (modelling)

The secondary circuit is a complex system: it is rather difficult to get a detailed performance assessment only by using hand calculations. Thermodynamic simulation enables calculation of the state of water everywhere in the circuit. It is thus possible to define performance indicators for each equipment, and then for the whole system. These indicators are calculated automatically once the correct input data is given, assuming that constant input parameters have been beforehand correctly tuned (electric generator efficiency coefficients for instance). Using thermodynamic simulation helps assessing performance.

The main issue is to choose the best physical description level to adopt. It is necessary to adopt a compromise between different sets of precision in modelling, in term of details that the model will be able to provide about estimated physic quantities.

A very-detailed model presents two main drawbacks:

- The more detailed the model, the more input data is required (to tune and to provide input to the model). It can be difficult for the end-user to gather the required inputs, to tune properly and then to compute the model. Especially if it is a periodic task.
- Physics based modelling enables representation of a perfect system. But when it comes to real physic behaviour, there are many possibilities for the system to operate differently (blow down opened, equipment clogging, pumping phenomenon, etc.). It is very difficult to take them all into account in the model. Differences between simulation and reality always remain, producing in proportion differences between measured values (coming from real sensors) and expected values (computed by the model).

To improve the representativeness of the model when compared to the real operation profile, this method suggests using a model which represents the thermodynamic state under specific stabilized operating conditions of the secondary circuit.

These conditions are compared to a reference state. In this approach, the number of limit conditions is optimized and reduces the input data required by the model.

(b) Use of reference state

Reference state refers to the best-known set of conditions in which the circuit is expected to operate. This state will not necessarily have been observed during commissioning. The conditions during commissioning are expected to be close to the reference conditions but are often different. The operating conditions of the reference state includes internal conditions, set by the operator, and external conditions, determined by the environment.

— Internal conditions

To reach the peak plant performance (i.e. the highest gross continuous electrical power with no time limit) the plant needs to operate close to its design. These conditions need to be listed before the application of the method.

Example: When the internal conditions are listed, those in Table 5 appear.

TABLE 5. INTERNAL CONDITIONS FOR THE SIMULATION

Internal conditions example	**Reference values example**
Heat power in steam generator	100% Nominal power
Steam generator blowdown flow rate	1% Nominal feedwater flow rate
Secondary circuit	Under normal operation (no isolated line, auxiliary boiler shut down, no blow down opened, etc.).

— External conditions

External conditions correspond to all the conditions determined by the environment which have an impact on the overall performance of the plant. Since it is impossible to force these conditions, they need to be listed, accounted for and reference values need to be fixed.

Example: The following external conditions in Table 6 can be identified, and reference conditions relevant with design studies are set.

TABLE 6. EXTERNAL CONDITIONS FOR THE SIMULATION

External conditions example	**Reference values example**
Cooling water temperature	13°C
Cooling water flow rate	50 m^3/s
Air dry-bulb and wet-bulb temperature	15°C and 11°C (when humidity rate = 60%)
Secondary circuit	Under normal operation (blow down closed, etc.)
Power factor	$\cos\phi = 0.9$

— Reference state use

Once a reference state is defined, while the plant is operating under stabilized conditions (closest as possible to the internal conditions mentioned above), the comparison can be made between the reference state and the operating plant state. For that purpose, the same limit operating conditions are set in the model. **The result of reference and real plant state computation enables, among others, to identify electric power production differences**. They are divided into three groups:

- The external losses: due to the external conditions of the secondary circuit (thermal power production, power factor, cooling water temperature, etc.).
- The internal losses: due to internal conditions (condenser defect, abnormal SG blowdown flow rate, back-up, blow down opens, etc.).
- The unknown losses: remaining electric power production loss once external and internal losses are removed.

To get more details, please refer to Section 3.5.4.3 (b).

(c) Analysis results and advantages

The comparison between the plant and the reference state needs to be conducted on a regular basis: a weekly basis is recommended to grant at least one representative comparison a month.

Regular comparison between the plant under stable operating conditions and the reference state is a performance assessment method which presents the following advantages:

— Each test provided major component diagnosis, identifies electric production power overall losses and allocates them between the different parts of the secondary circuit.
— In the long-term, regular tests help overcome model imprecision. Trends can be followed between and during cycles and more detailed performance indicators can be calculated. The plant performance overview is thus improved, and more actions can be identified to recover electric power losses.
— The model is based on data gathered during performance tests carried out during plant commissioning for instance. It can also be modified with additional data collected throughout operation: it is not based on calculations, which often require a significant number of tests to be completed. Therefore, **it does not need long specific tests**.

Warning about the use of results:

Like all the other performance methods described in this publication, a TPE needs to bear in mind that this method is not self-sufficient: it helps the performance engineer carry out the analysis, but it cannot replace human knowledge.

And of course, because expected values result from a model fed with data, any error or drift during the elaboration of input data leads to potential mistakes into performance evaluation (either sensors issues, offset management, acquisition failure or erroneous numerical conversion, as part of a non-exhaustive list of possible deviations). Nevertheless, because this method is not exclusively data-driven, inspection of parts of the model computational behaviour can help identify consistency anomalies within input data.

3.5.4.2. Physical plant and computation model comparisons – method requirements

Physical plant and computation model comparison method provides a detailed overview of the secondary circuit performance. To do so, some requirements need to be fulfilled so that the method gives the expected results.

The requirements can be grouped as follows: instrumentation, staff organization and modelling requirements.

(a) Instrumentation requirements

To compute the thermodynamic model, it is necessary to have the required **set of data to calculate the mass and heat balance in every significant point of the secondary circuit**. The data can be either assumptions or measured values; the latter will of course give more precise results. Except for data that is difficult to measure (for instance: steam extraction humidity, SG vapor humidity), **it is advised to choose measurements for all the parts of the secondary circuit that have a strong impact on performance**.

For example, within the feedwater system, it is important to collect measurements on the high-pressure heaters. On the contrary, the low-pressure heaters have a less significant impact on performance and can thus be left without complete instrumentation. In the same way, it is important to correctly instrument the turbine, the MSR and the condenser. In the NPPs in which this method has be implemented, there are at least 80 sensors (flow rate, temperature and pressure) dedicated to this method.

In order to improve the accuracy of the diagnosis, the measurements are required to be as reliable as possible. It implies that:

- Sensors need to be calibrated to minimize the measure uncertainty.
- The maintenance programme needs to integrate specific calibration frequencies to correct sensor drift.

In plants in which this method has been implemented, the choice is typically made to use dedicated sensors. It is thus easy to choose the best technical characteristics and to elaborate a specific maintenance programme. However, it is recommended to consider early during the design. For operating plants, studies can determine whether it is appropriate to use existing sensors to implement this method.

Furthermore, these sensors are not required for plant operation. They can be replaced with operating sensors and maintained independently. Additionally, comparisons between operating and test sensor values, when locations are similar, provides information that can be used in calibration and online performance monitoring processes.

To get more information about measurement, please refer to Section 3.3.2.

(b) Organization requirement

The method requires compliance with specific operating conditions during testing and to have operational equipment available on site. It is thus mandatory to define organizational responsibilities related to:

- Measurement device management (maintenance, inspection and sensor calibration).
- Test operation procedure gathers the steps to be followed during the test, defines the contributors involved and their responsibilities and outlines the analysis and the validation process. For instance, it lists all actions to be taken by operators and the test technician to ensure relevant measurements are trustworthy.
- The search for non-identified defects.
- The action plan to solve these defects and to recover performance.
- Results checking and communication.

The greater part of this organization is normally on site, but a part can be handled elsewhere, in a business unit or an engineering unit (for instance outside the plant but whose role involves support, training, in-depth analysis, and the collection of feedback and lessons learned).

(c) Simulation requirement

The use of simulation implies thermodynamic model development at the beginning. It means that **human and financial resources need to be dedicated at first** to model specification, data gathering, model computation development, and model calibration. This effort is often relevant if the method is to be applied in the long run, or if there is more than one plant which can benefit from the developments (standardized plant series for instance).

Once the hardware and software are operational, continued human and financial resources maintain system functionality. For example, the process to modify the model (equipment modification with new design characteristics), needs to be clearly identified and defined to maintain the relevant information and technical skills.

In conclusion, this method provides a detailed overview of the secondary circuit performance provided that some requirements are fulfilled. These are easier to establish if the method is to be applied over the long-term, or if there is more than one plant which can benefit from the resource investment.

3.5.4.3. Method application

(a) Input data definition and acquisition

— Input data definition

Before beginning regular testing, it is necessary to define the physical quantities to be measured. As mentioned before, the chosen data need to enable the determination of mass and heat balances in parts of the secondary circuit which significantly impact performance (see Section 3.5.4.2 (a)).

Depending on the situation, it is not necessary to gather every possible measurement. For instance, if two heat exchangers are in line with sensors that provide the heat and mass balance, it is possible to calculate the feedwater output temperature.

However, some quantities are difficult to obtain. In this case, if the quantity is expected to be stable, data from periodic tests or commissioning performance tests can be used. For example, the humidity in steam extraction lines and in the turbine steam path is difficult to measure. It is possible to define this value when the turbine is commissioned. Normally, this is not expected to change unless significant modifications are performed on the turbine steam path. The humidity values can be extracted from these test results and used during the model tuning phase.

The study therefore is developed from sensors and values to be measured each time a test is completed (temperature, flow rate, pressure) and the set of fixed parameters for the model.

At NPPs in which this method is applied, typically on the order of 80 sensors (flow rate, temperature and pressure) provide measured input. These sensors are divided as shown in Table 7. In Figure 33 below, a thermal model with sensors is shown. This figure serves only illustrative purposes (indicating the level of detail related to data collection).

TABLE 7. SENSORS CATEGORIES

High pressure turbine	~ 5 sensors
MSR	~ 20 sensors
Low pressure turbine + low pressure heaters	~ 8 sensors
Condenser and water intake	~ 15 sensors
Turbine-driven feedwater pump	~ 5 sensors
High pressure heaters	~ 15 sensors
General: electric power, steam generator feed-in water, steam generator vapor	~ 15 sensors

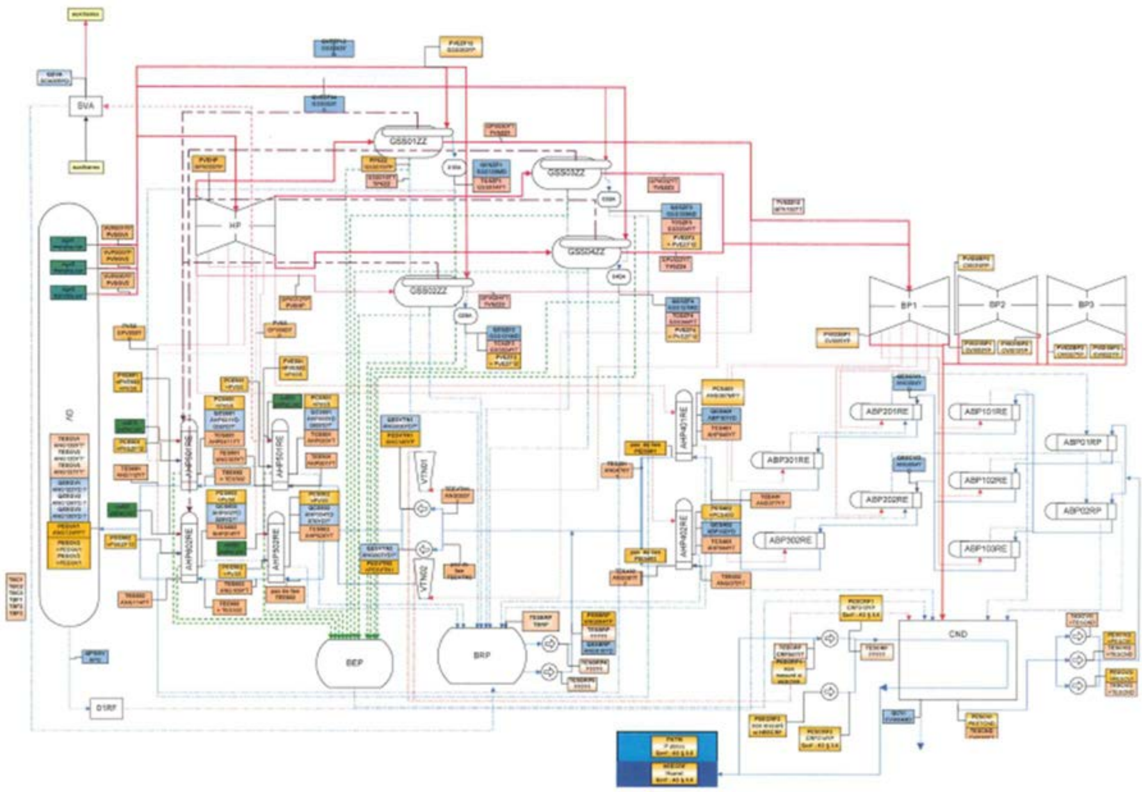


FIG. 33. Thermal model (with sensors)

There are typically about 15 fixed parameters: steam humidity out of the SG, pump characteristic and performance curves, electric generator efficiency, tide height (in the case of a NPP located near the sea). To help identifying parts of the secondary circuit to focus on, and the mass balance and heat balance equations, please refer to Section 5.

— Input data acquisition and collection

The data acquisition is carried out according to the operation organization procedures and guides (see Section 3.5.4.2 (a)):

(i) The operator configures the plant as close as possible to the **reference state**. In the example given in Section 3.5.4.1. (b), the operator gets close to 100% nominal power, shuts down the auxiliary boiler if possible, checks that there are no isolated lines, etc.

(ii) The operator maintains the reactor / primary systems in steady state operation until the secondary circuit is **stabilized**. It can take a few hours (moreover, this 'stability window' needs to be scheduled or otherwise coordinated with the grid dispatch operator)

(iii) The field-technician, according to the organization, authorizes or initiates the preacquisition check. The sensors are checked and drained, the plant state is checked, etc.

(iv) The data acquisition is launched. Even if the plant is supposed to be in a stabilized state, it is recommended to choose an acquisition period long enough to catch instabilities (pumping, abnormal measure variations): between 15 to 30 minutes for example. The stabilized state needs to be checked throughout the duration of the test. A typical 5 second acquisition time step is typically enough to ensure each acquired quantity provides about 180 – 360 time stamped values.

(v) Once the acquisition is complete, a first check on mean values and scattering for each value helps to validate the set of data.

(vi) The checked results are stored and sent to feed the thermodynamic model.

For each test, the collected data is used as input in the thermodynamic model and computation system. The following describes how to proceed.

(b) Thermodynamic model use

— Goal.
Apply thermodynamic computation to assist in performance monitoring as well as the correction of identified losses.

— Assessing specifically the performance of each secondary systems.
As the model simulates each system of the secondary circuit, with mass and heat balance, the water physical state is known almost everywhere in the circuit. Key performance indicators being defined, they are calculated for each system.

— Identifying and quantifying the electric power production losses.
The thermodynamic model is computed with data collected during the test. This model is compared to the reference state to identify where the source of electric power production discrepancies as shown in Figure 34. Three groups are typically categorized as:

- External losses: due to the external conditions of the secondary circuit (thermal power production, power factor, cooling water temperature, etc.).
- Internal losses: due to internal conditions (condenser defect, abnormal SG blowdown flow rate, back-up, blow down opens, etc.).
- Unknown losses: remaining electric power production loss once external and internal losses are removed.

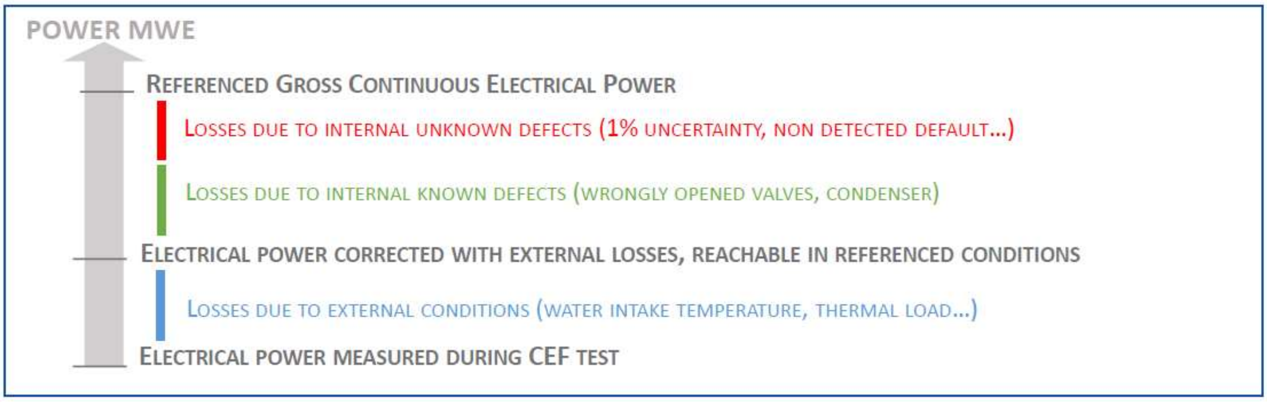


FIG. 34. Losses identification

(c) Thermodynamic model content

Input data of the model consists of:

- Data collected during the test (measured values): pressure, flow rates, temperatures, electric power, etc.
- Input parameters, which are considered stable from one test to another. It can be for example: steam humidity from the SG, feed in water distribution between two lines of re-heaters. These values often come from previous tests, which are carried from time to time but cannot be done on a regular basis (because there are expensive, or because they need specific operation conditions).

(d) How to carry the comparison (shown in Figure 35) between model and reference data

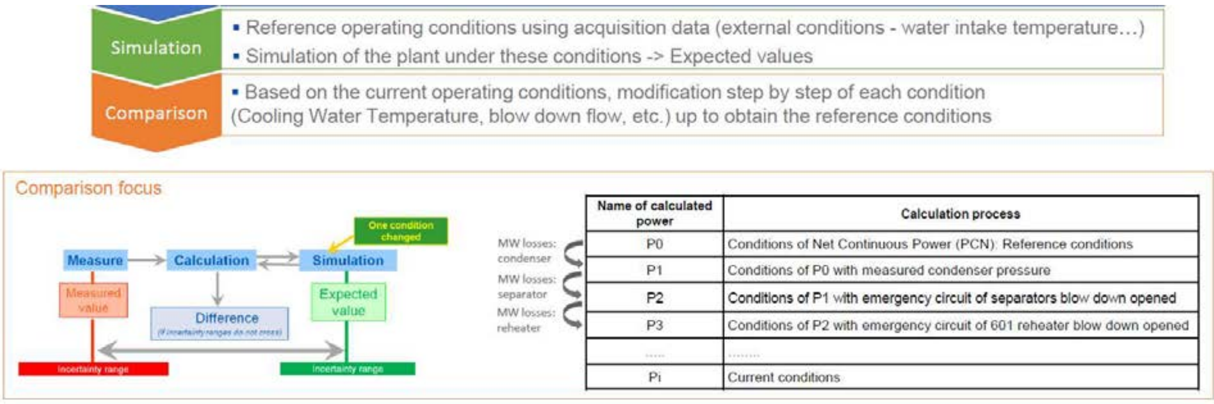


FIG. 35. Comparison between model and reference data

Steps to be followed (lettered from (i) to (iv)):

(i) Step 1: Reference conditions definition.

Apply the model in the reference state to obtain the reference overall electric production.

(ii) Step 2: Calculation of external conditions difference between test conditions and reference conditions.

For each external condition difference:

- Apply the model with all reference state conditions, except the external condition that are already dealt with.
- Determine the difference between reference electric production, and the electric production computed.

(iii) Step 3: Calculation of the difference between test conditions and reference conditions for internal conditions.

For each internal condition difference:

- Apply the model calculations with all reference state conditions, except the external condition that are already dealt with.
- Make the difference between reference electric production, and the electric production computed.

(iv) Step 4: Residual differences analysis

Once external and internal power losses difference are determined from the reference state production, the remaining differences come from unexplained defects.

3.5.4.4. Analysis and expected results

The results from the comparison between the measurements and models can be used and monitored over time thanks to different tools:

- A view of the thermodynamic balance for each component and for the overall plant as shown in Figure 36 (this figure serves only illustrative purposes). This is a great way to improve knowledge of the plant's behaviour.

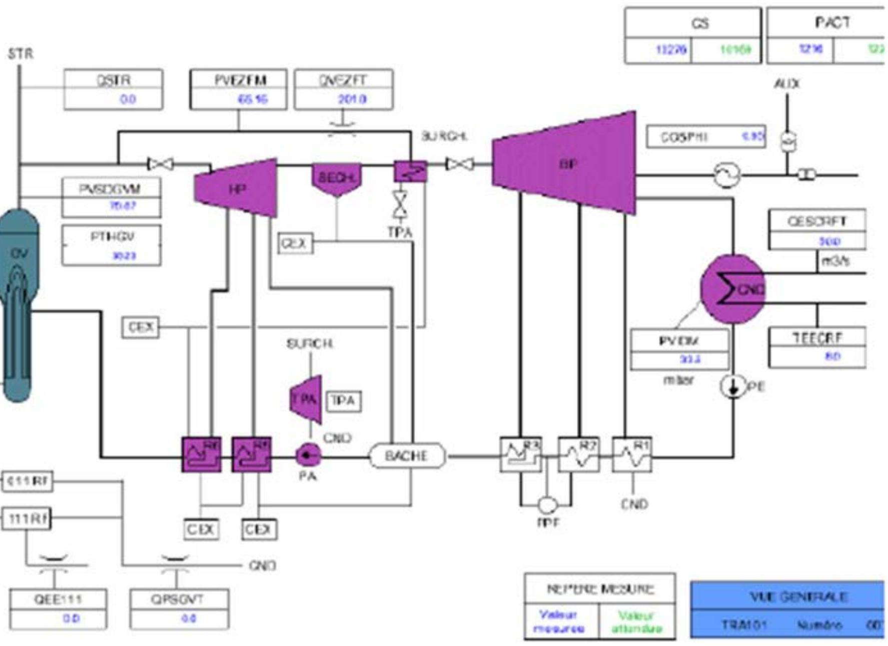


FIG. 36. Thermodynamic balance results

— A ‘resulting’ spreadsheet listing the differences between measured and expected behaviour is shown in Figure 37. Hereunder are listed the external losses (blue rectangle), internal losses (green rectangle), and resulting ‘unexplained’ or ‘unidentified’ losses (red rectangle). This figure is only for illustrative purposes (to show the layout of ‘resulting’ spreadsheet).

EDF Electricité de France

RESULTATS DU CEF

(acquisition)

SAPEC

EDF

Centrale : CHOOZ B	Tranche: CHZ1	CEF No 015	Essai Patern 199
Section : ESSAIS	Cycle : 6 / 5	Date Rapport 20/01/05 16:40:54	
Réalisateur deux	Visa	Date Acquisition 20/01/05 16:23:16	1min 9s

ECARTS DE PUISSANCE				PCB = 1557 MW	
	Désignation	**Unité**	**Valeur mesurée**	**Valeur à la PCN**	**Ecart MW**
Ecarts externes	Charge thermique GV	MWth	4261	4270	-3.5
	Temp eau CRF (hors aéro)	°C	21.4	23.4	12.1
	Débit eau CRF entrée condenseur	m3/s	42.63	50.10	-12.3
	Cosinus phi		1.00	0.90	4.0
	Débit STR	kg/s	0.00	0.00	0.0
Ecarts internes dus à la configuration	Débit des purges GV	kg/s	22.00	24.0	0.0
	Orientation des purges GV		CEX	->CEX	
	Secours purges sécheur 001zz	%	0.0	0.0	0.0
	Secours purges sécheur 002zz	%	0.0	0.0	0.0
	Secours condens surch. 11+13	%	0.0	0.0	0.0
	Secours condens surch. 12+14	%	0.0	0.0	0.0
	Secours condensats. AHP601RE	%	0.0	0.0	0.0
	Secours condensats. AHP602RE	%	0.0	0.0	0.0
	Secours condens AHP500RE	%	11.3	0.0	-5.9
	Désignation	**Unité**	**Valeur mesurée**	**Valeur attendue**	**Ecart MW**
Ecarts internes dus au materiel	Condenseur	mbar abs	66.4	70.8	7.0
	Aéroréfrigérants	°C	22.4	21.4	-6.1
Ecarts	Total par rapport à la PCB	MW	1		
	Externes	MW	0		
	Internes	MW	-5		
	Inexpliqués	MW	5		
Puissances	Brute mesurée	MW	1558		
	Corrigée des écarts externes	MW	1557		
	MAX pendant l'essai	MW	1558		
	MIN pendant l'essai	MW	1558		

FIG. 37. Resulting spreadsheet of losses

— The engineer in charge of monitoring the plant uses tools to get a view of the trends and to detect early drifts, during one cycle or over years.

— The collected data is used whenever needed for in-depth analysis (intra or infra-cycle). It can for instance be used to understand the impact of equipment changes or modifications (i.e. a heat exchanger) on the overall performance of the plant. It can also be useful when looking for the ‘unidentified losses’, for example to understand where to apply a more rigorous diagnosis.

3.5.5. Generic performance calculation (empirical relationship)

For an overall evaluation of plant performance some plants use a methodology based on empirical relationships between plant parameters and plant output. Based on these relationships, the amount of generation can be compared to a ‘target’ value derived from corrections supplied by these empirical relationships. Corrections can be applied to the measured plant generation to account for known effects. Performing this ‘accounting’ can prevent an engineer from ‘chasing their tail’ or evaluating a problem that is not real. The following methodologies are available to determine these relationships.

(a) Vendor curves
- Supplied with plant thermal kit.
- Not always accurate due to difference between designed and installed conditions.
- Sometimes they do not consider the whole plant (e.g. blowdown flow, source of cooling water, actual feedwater heater configuration.

(b) Plant thermodynamic modelling curves
- Based on a thermodynamic plant model;
- Uses the heat balance as a baseline;
- Can be tuned to actual plant data;
- Can be used to verify vendor curves.

Cautions that need be observed when using a thermodynamic model to develop correction curves:

- Know what is in your model. Especially if the model was developed by another engineer it is important to understand all the inputs and configuration of the model.
- Check that the plant model to be used for generating correction curves ‘floats’ with core thermal power as an input.
- Verify that there are no hard values (e.g. fixed pressures/flows etc.).
- Check that the model agrees with the heat balance and plant operation.

Steps to develop a curve (from (i) to (iii)):

(i) Step1: Vary the parameter of interest and plot the change in the parameter versus the change in gross generation.
   - May need to use ‘design-mode’ components.
   - *Check redundant indications and references* - Look at more than parameter of interest to verify you are not being fooled.
   - *Check redundant indications and references* - Compare to other curves either in thermal kit or other plant curves.

(ii) Step 2: Use Excel or a similar tool to graph the results and determine the equation for the change in generation with respect to the change in the parameter.

(iii) Step 3: Use the trend line from the plot for correcting gross generation.

Various example curves are provided below in Figs 38 – 43. Some of the curves compare well with the thermal kit curve. However, some are not so good. Any discrepancies between the two curves ought to be dispositioned.

Final Feedwater Heater (6) TTD				
Performance Summary				
	TTDs (deg F)	MWe	Delta MWe	
	4.00	945.319	0.280	
	5.00	945.204	0.166	
Baseline	**6.50**	**945.038**	**0.000**	
	6.00	945.093	0.055	
	7.00	944.983	-0.055	
	8.00	944.874	-0.164	
	9.00	944.7636	-0.274	
	10.00	944.653	-0.385	
	11.00	944.5418	-0.496	
Polynomial (for delta MWe)	(MWe) = -0.1106 * (TTD)			
Factor	**-0.1106**			

FIG. 38. Performance summary of feedwater heater

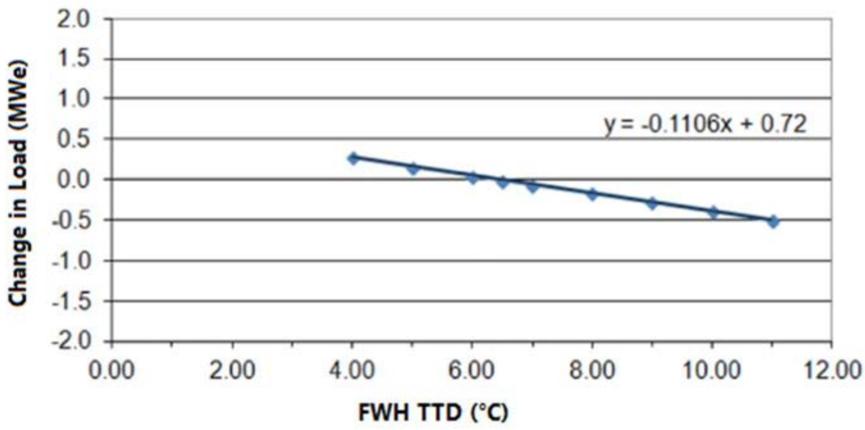


FIG. 39. Resulting curve of relationship between load and feedwater heater

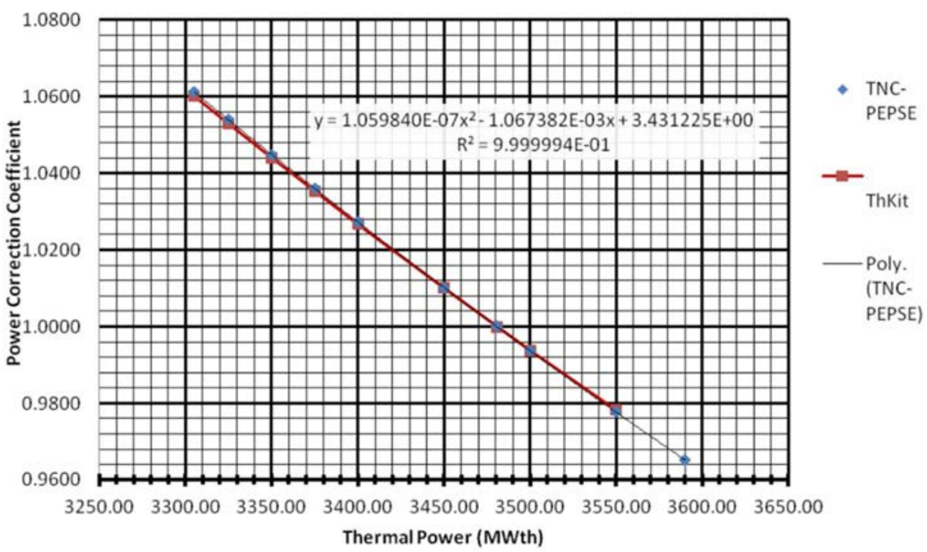


FIG. 40. Resulting curve of relationship between power correction and thermal power

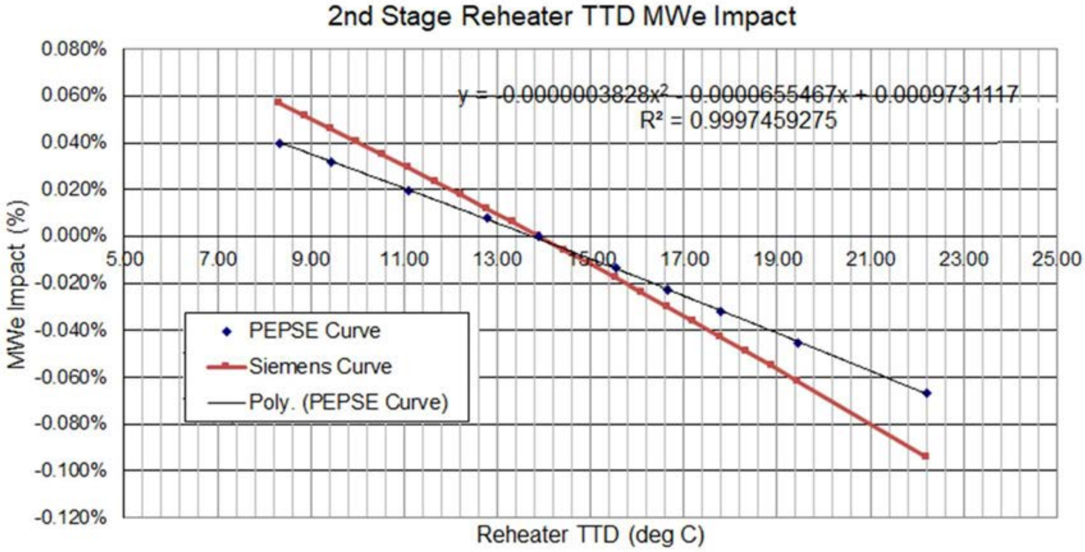


FIG. 41. Resulting curve of relationship between power correction and average MSR TTD

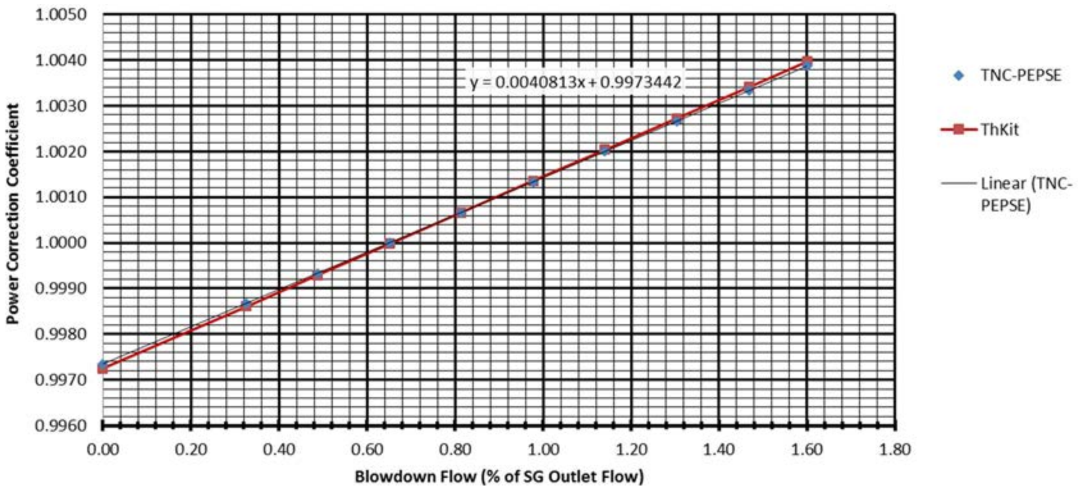


FIG. 42. Resulting curve of relationship between power correction and blowdown flow

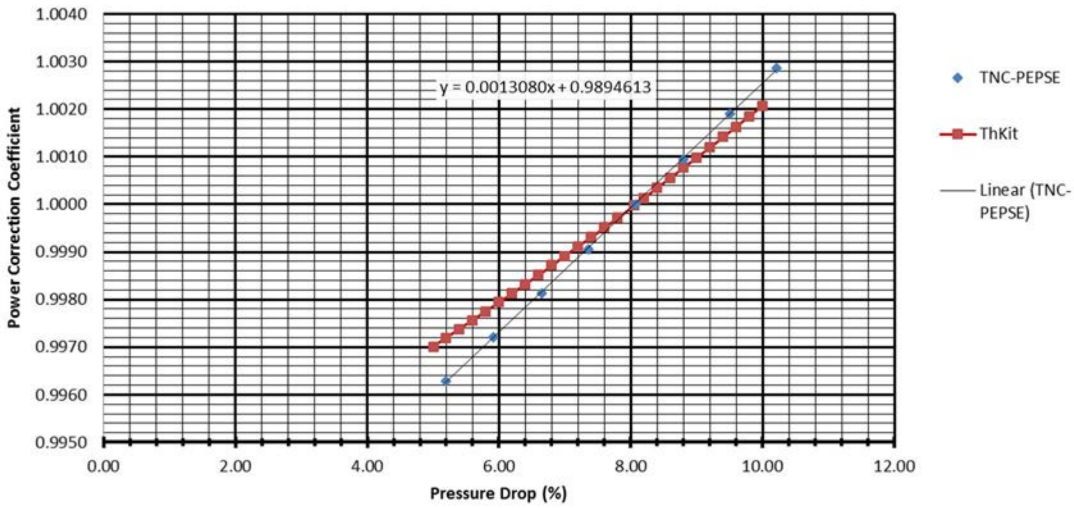


FIG. 43. Resulting curve of relationship between power correction and pressure drop

(c) Empirical curves

Empirical curves are based on historical plant performance. Common curves developed include condenser pressure to generation correction curves and circulating water inlet to condenser pressure correction curves or circulating water inlet to generation curves.

The starting point for an empirical curve is plant data, but it will look like the Figure 44.

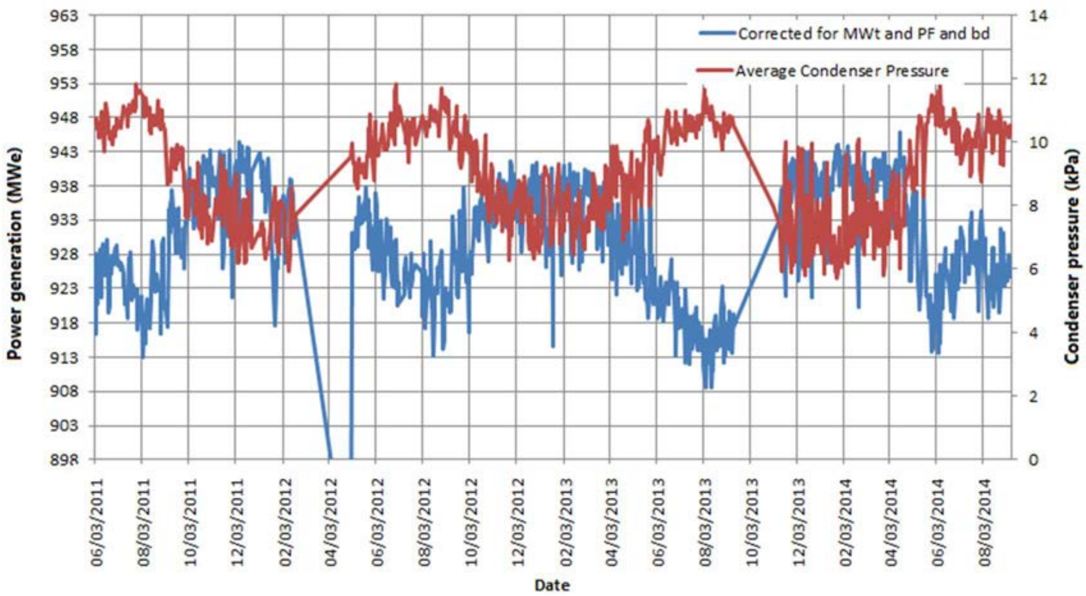


FIG. 44. Power corrected generation

The goal is to convert the data (process shown in Figure 45) in the graph above to a reliable curve to correct generation over time based on condenser pressure and circulating water inlet temperature.

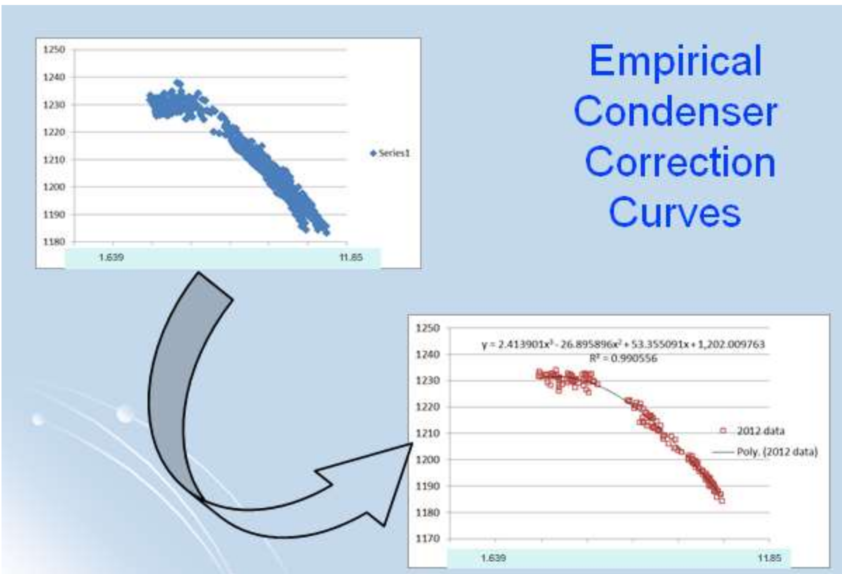


FIG. 45. Empirical condenser correction curves [1]

Steps (lettered from (i) to (vi)) to develop curves:

(i) Step 1: Check for recent turbine test data which were able to adjust condenser pressure during the test. This is the preferred method over using historical plant data as the conditions are controlled. However, the results would be invalid if they cannot be corrected to current plant conditions. If test instruments are used for the curve, the data are advised to be corrected to plant instruments to make the correction curves valid for normal plant operating conditions. If there have been any changes in plant operation or configuration since testing was performed, these need to be evaluated to verify the applicability of the curves to the new conditions. The test also needs to cover the entire range of condenser pressures and circulating water inlet temperatures seen in operation, summer to winter.

(ii) Step 2: If no test data are available, the historical plant data ought to be used to determine the corrections. The following steps are required to be followed to develop the curves.

(iii) Step3: Filtering data can help deal with issues described in the following Figure 46.

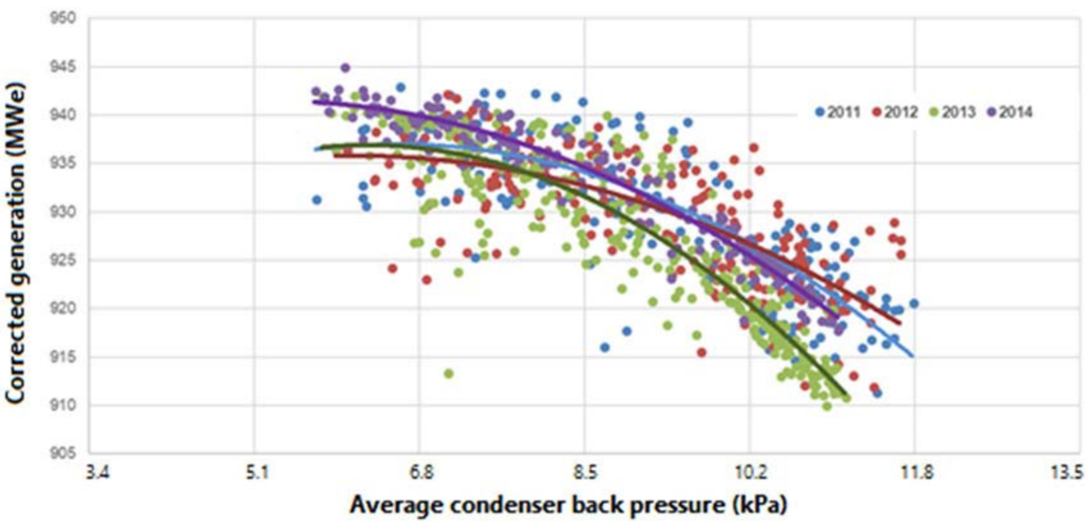


FIG. 46. Condenser back pressure vs gross generation

(iv) Step 4: It is clear from Figure 46 that there are different curves based on what year is chosen. Filter data for:

- 99.5% reactor power (as shown in Figure 47);

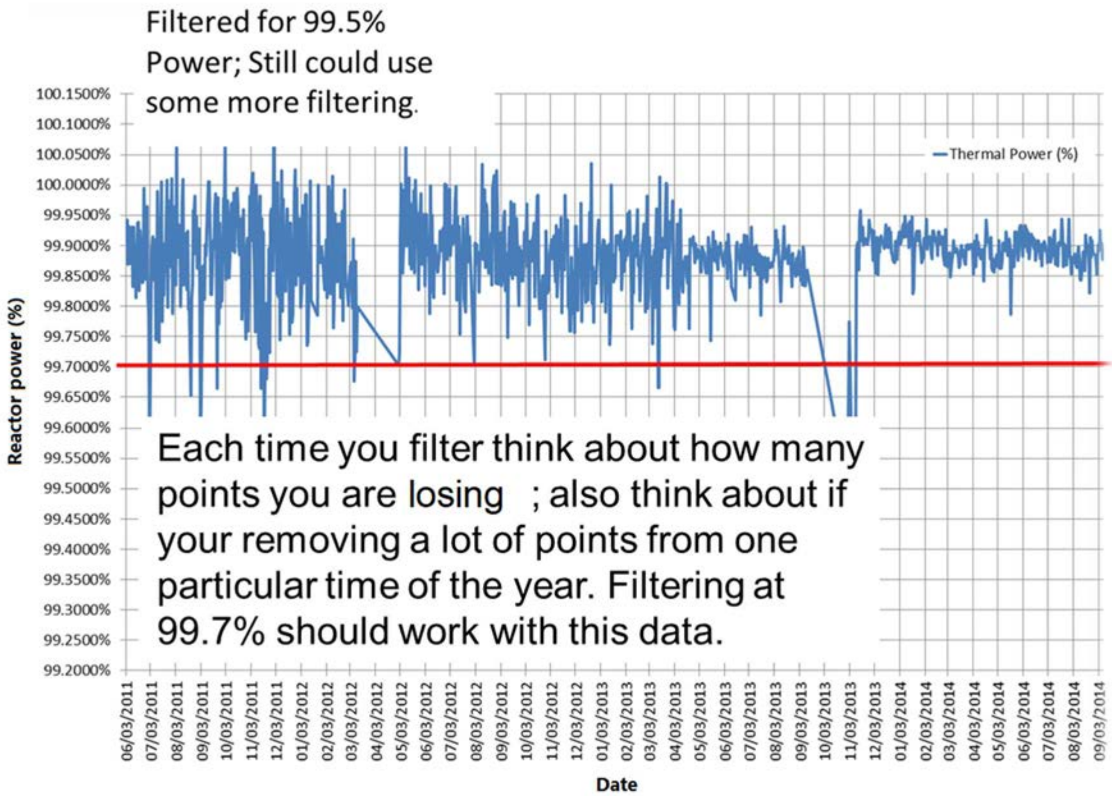


FIG. 47. Filtered data for power vs time

- Average first stage pressure – determine which instrument to use (as shown in Figure 48).

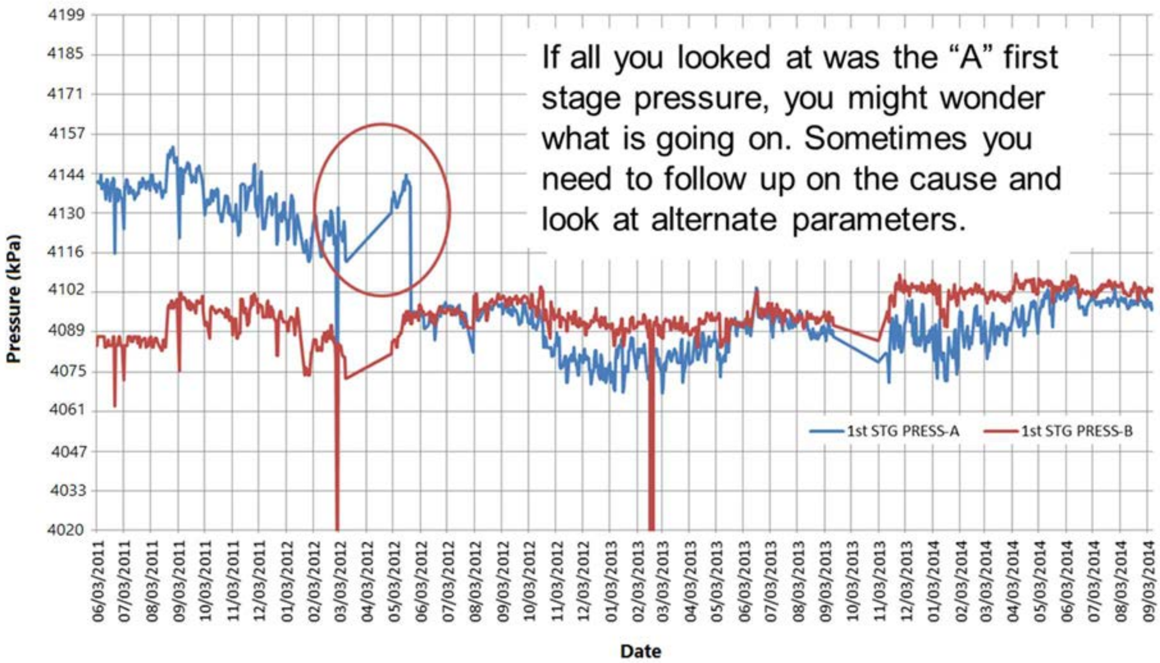


FIG. 48. Average first stage pressure

- Average circulating water temperature rise (as shown in Figure 49).

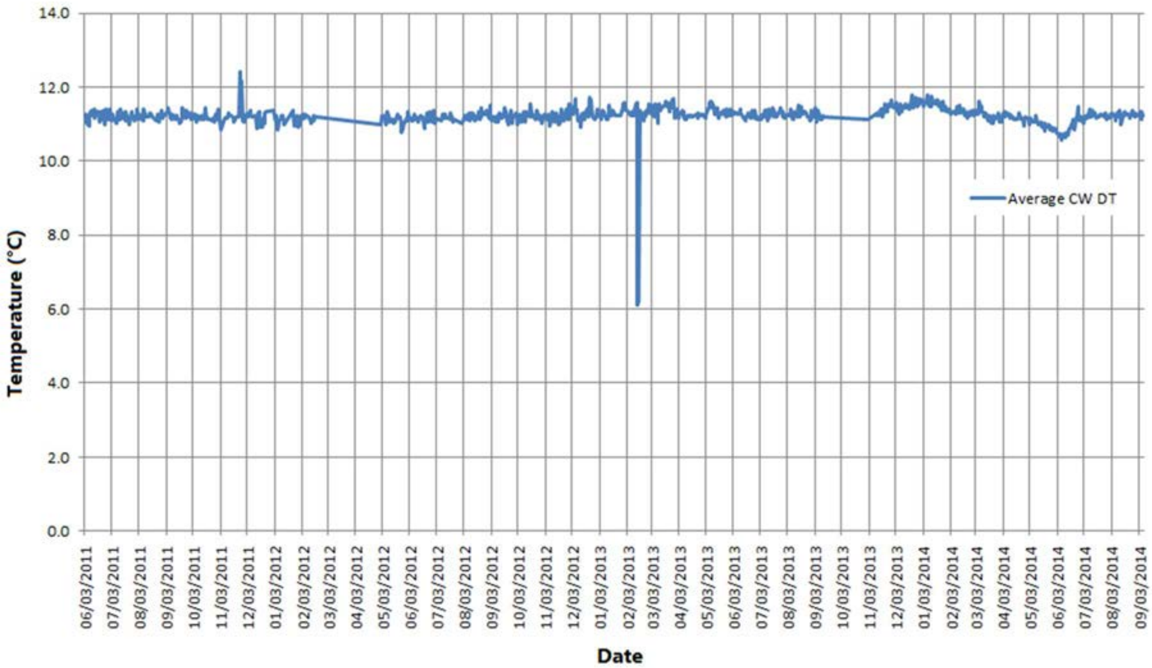


FIG. 49. Average circulating water temperature

- Outages;
- Anything that indicates 100% power.

(v) Step 5: Correct gross generation for reactor power (see Figure 50).

Note: Can also correct to a baseline power factor and/or blowdown flow (PWR) if desired.

— Plot average condenser pressure versus corrected generation.
— Filter for obvious statistical outliers.
— Create trend line.

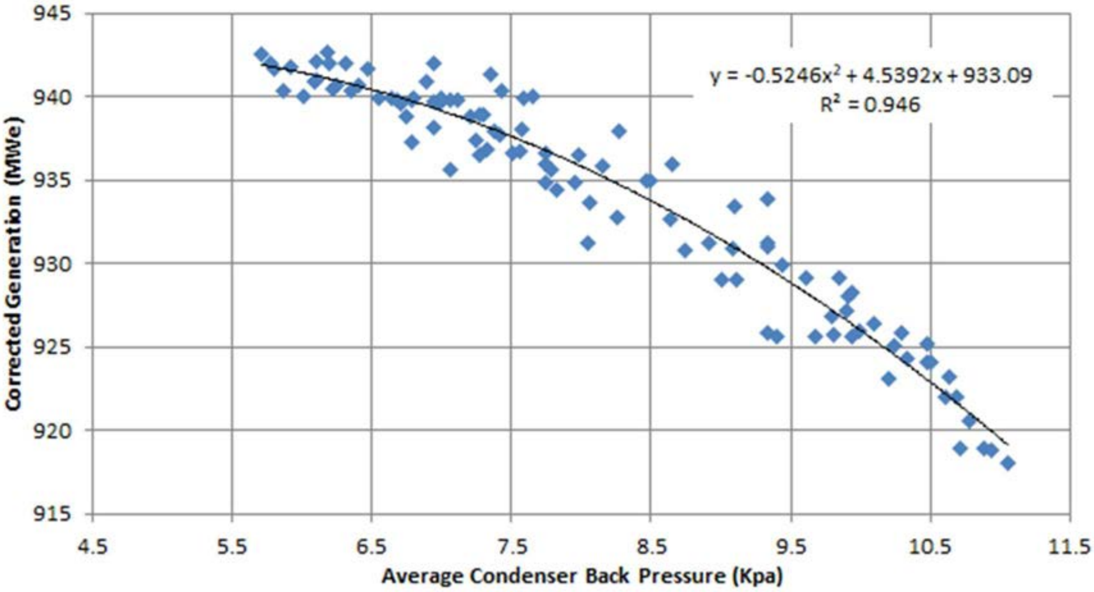


FIG. 50. Correct gross generation for reactor power

- Filtered data still covers the entire range of back pressures.
- R2 > 0.9 (correlation).
- Check equation against condenser pressures across the range.

(vi) Step 6: Calculate Baseline Generation.
- Correct the test baseline generation from test condenser pressure to design condenser pressure.
- Use trend line to find generation at the design back pressure.

A similar process is followed to determine the expected condenser pressure curve for condenser cooling water inlet temperature as shown in Figure 51.

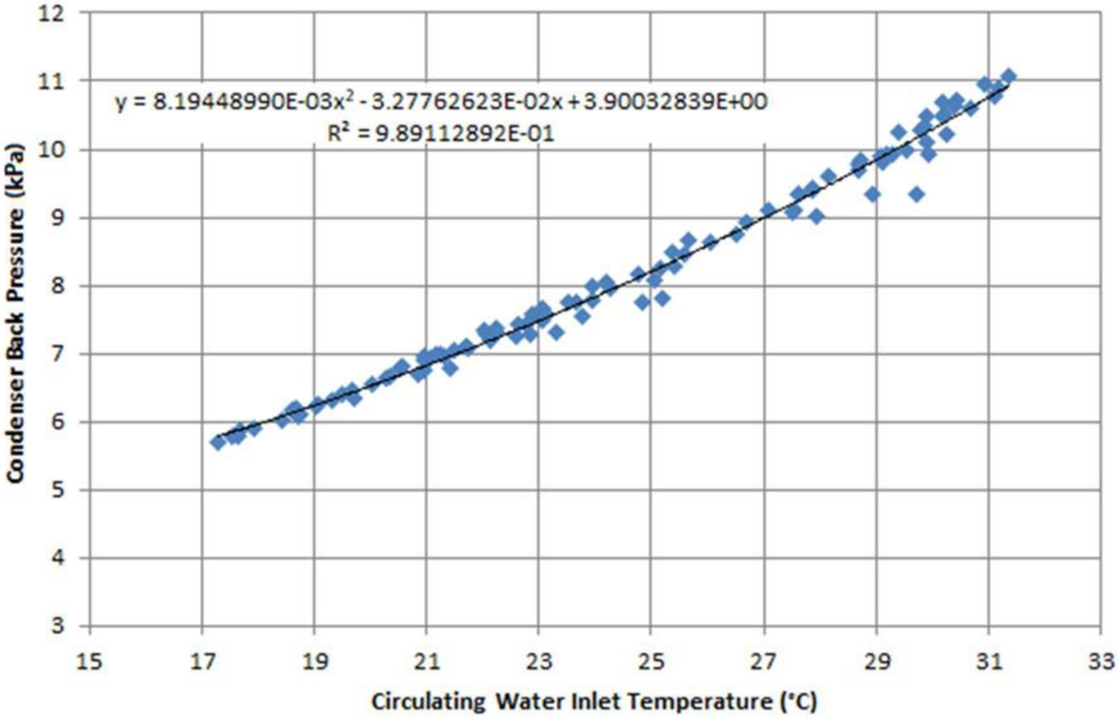


FIG. 51. Expected condenser pressure curve for condenser cooling water inlet temperature

3.5.6. Advanced pattern reconciliation

The monitoring of NPP efficiency and more generally its performance can be traditionally set up in three different ways:

- Establish a maintenance and diagnostic centre.
- Apply an advanced pattern-recognition solution based on existing plant process data.
- Apply a thermal performance modelling solution based on existing plant process data.

These different approaches reduce costs by identifying plant defects before they become catastrophic failures, by reducing system engineer working time and by reducing NPPs losses.

(a) Principle of approach

In the Advanced Pattern Recognition approach (APR), no mathematical model of the process is needed. Knowledge about the system is assumed to consist exclusively of a learned set of measurement vectors and associated operating conditions. This data is used to build a mapping of the measurement space onto a decision space, in such a way that the probability of misclassification (or assignment to a wrong state) is minimized. The pattern recognition methodology is usually divided in two stages: feature extraction and classification.

Feature extraction consists of finding a limited but informative representation of the process, based on raw measurements. Usually, a large set of candidate features is first computed using signal or image processing techniques, and multivariate statistical procedures are used for selecting a subset of these features or combinations thereof.

Once a suitable representation space has been defined, this space is partitioned into decision regions corresponding to the assignment to each of the known states, or pattern rejection.

(b) Implementation on NPP

The implementation of this approach on NPP takes place in several phases:

- Identifying components whose problems can lead to plant downtime, load reduction or even reactor trip.
- Identification of the sensors available to monitor these components.
- Identification of possible failure modes of the component. This approach allows selection of the most important sensors for tracking the equipment's health.
- Definition of reference periods where the equipment performed optimally.
- Definition of admissible limits of drift in terms of the possible risks. If the threshold is too restrictive, the model will generate too many alarms. On the contrary, if the threshold is too high the model will not detect certain levels of dysfunctional behaviour.
- Establishment of an organization responsible for the management of alarms and for the prompt analysis of any identified drift.

This APR approach can be implemented in two ways:

- A centralized mode: Each NPP is independent in terms of the creation and management of alarms. This approach is justified by the staff's detailed local knowledge of the equipment and operating modes.
- A decentralized mode: In this case, another group is responsible for the entire nuclear fleet for the creation and monitoring of the APR model. This approach makes it possible to pool skills and share similar operating problems more effectively.

(c) To go further

This traditional approach of on-line monitoring has proven its efficiency in reducing maintenance costs and engineers' working time. The next step in an online monitoring programme to optimize costs, based on two studies:

- Increasing the number of sensors in the NPP by using new technology, which enables early detection of a more comprehensive set of failure mechanisms. The use of wireless sensors can avoid the problem of heavy cabling.

- Applying advanced equipment diagnostics and prognostics. The APR approach gives a warning about the onset of parameter drift once a threshold has been crossed but does not give information on the origin of this problem. The next step is to determine the origin by root cause analysis and predicting the evolution of the dysfunction.

4. KEY COMPONENT PERFORMANCE – NUCLEAR STEAM SUPPLY SYSTEM

Nuclear power plants use the fission process to generate heat. This heat is used to convert water into steam. The primary heat source as defined in this publication includes the reactor and/or SG. In a BWR plant, saturated steam is generated in the reactor core, which is then passed through two stages of moisture removal elements before it is directed to the HP turbine. In a PWR plant, hot water from the reactor is circulated through the SG tubes. The heat is transferred to the shell side of the SG via the tubes causing steam production. Depending on the design of the SG the steam produced is either high quality saturated steam or superheated steam. CANDU reactors circulate heavy water through the SGs, while in a high temperature gas reactor, hot gas is circulated through the SGs which may produce both superheated main steam and reheat steam.

Various designs of SGs exist. Feeding U-tube type of SGs in which the feedwater enters a downcomer region and flows up past the tubes. High quality (typically greater than a quality of 0.9975) saturated steam exits from the top of the SG after passing through two moisture separator sections in the steam dome. The moisture that is separated from the saturated steam recirculates with the feedwater. The moisture return path is through the downcomer region between the tube wrapper and the outer SG shell. The SGs have feedwater entering in a lower baffled section of the T_{cold} region of the tubes, which uses the primary coolant at T_{cold} to preheat the feedwater.

Another design is the once-through SG. The once-through SG produces superheated steam. This is achieved by passing the primary coolant from the reactor which enters to the top of the SG and flows in a counterflow direction to the feedwater. The bottom portion of the tubes serves as an economizer region where the incoming feedwater is preheated to saturation conditions. The feedwater boils and saturated steam is produced in the middle section of the tube region and the steam is then superheated by up to 10 °C in the top tube region as it passes over the hot incoming primary coolant tubes.

The SGs used in PWR plants are heat exchangers with reactor primary cooling water on the tube side entering at T_{hot} typically between 321~332 °C and exiting at a T_{cold} between 282 ~301°C. The average of these temperatures is referred to as T_{avg}. Steam generator outlet pressures and temperatures vary with load but may reach as high as 78.26 bar and 317.2 °C. The boiling water reactor does not use an intermediary heat exchanger (SG) to transfer the heat to the turbine cycle. The steam is generated directly in the reactor and passes through the moisture removal devices to produce high quality steam. The moisture is recirculated back to the boiling region of the reactor.

4.1. REACTOR THERMAL POWER MEASUREMENT

Reactor thermal power (RTP) is typically calculated using secondary side steam properties instead of directly observing the nuclear instruments due to their relatively high inaccuracy. The control volume is drawn around the thermal interface between the primary and secondary systems to establish all the energy sources that cross the boundaries. This interface differs between plant types.

There is a distinction between the amount of energy provided by the reactor and the amount of energy available to the turbine cycle so called Secondary Thermal Power (STP). This difference is the summation of the energy credits and losses to the primary system. The most significant energy credit to the reactor is the heat added by reactor coolant pumps or recirculation pumps. Different equations are used for each general type of plant, as examples outlined below.

4.1.1. Pressurized Water Reactor (PWR) power calculation

The PWR thermal power is calculated using a control volume around the SGs. Typical thermal power calculations for a PWR are as follows in Eqs (31, 32):

$$\mathrm{RTP} = m_{fw} \cdot (h_{ms} - h_{fw}) + m_{bd} \cdot (h_{bd} - h_{fw}) + Q_{misc} - Q_p - Q_{pzr} \tag{31}$$

$$\text{STP} = m_{fw} \cdot (h_{ms} - h_{fw}) + m_{bd} \cdot (h_{bd} - h_{fw}) \tag{32}$$

Where Q_{pzr} is heat added by the pressurizer (kJ/h), Q_p is heat added by the reactor coolant pumps (kJ/h), Q_{misc} is ambient losses (kJ/h), m_{fw} is final feedwater flow rate (kg/h), h_{ms} is main steam enthalpy (kJ/kg), h_{fw} is final feedwater enthalpy (kJ/kg), m_{bd} is blowdown mass flow rate (kg/h), h_{bd} is enthalpy of blowdown (kJ/h).

Note: Blowdown energy is included in the Secondary Power equation, but it is not necessarily available to the turbine cycle unless there is a heat recovery system.

4.1.2. Boiling Water Reactor (BWR) power calculation

In BWP plants, the pressure vessel is part of the steam cycle, so the control volume is drawn around the reactor and its supporting systems as determined in Eq. (33).

$$\begin{aligned}\text{RTP} = m_{fw} \cdot \left(h_{ms} - h_{fw}\right) + m_{crd} \cdot (h_{ms} - h_{crd}) + m_{rwcu} \\ \cdot \left(h_{rwcu_{in}} - h_{rwcu_{out}}\right) + Q_{misc} - Q_p\end{aligned} \tag{33}$$

Where, Q_{pzr} is heat added by the pressurizer (kJ/h), Q_p is heat added by the reactor coolant pumps (kJ/h), Q_{misc} is ambient losses (kJ/h), m_{fw} is final feedwater flow rate (kg/h), h_{ms} is main steam enthalpy (kJ/kg), h_{fw} is final feedwater enthalpy (kJ/kg), m_{crd} is mass flow rate to control rod system (kg/h), h_{crd} is control rod system inlet enthalpy (kJ/kg), m_{rwcu} is mass flow rate to reactor water clean-up system (kg/h), $h_{rwcu,in}$ is reactor water clean-up enthalpy from the reactor (kJ/kg) and $h_{rwcu,out}$ is reactor water clean-up enthalpy to the reactor (kJ/kg).

4.1.3. CANDU power calculation

CANDU plants are similar to PWRs on the secondary side with the addition of a drain line from the main steam reheaters entering the SG. Like PWRs, secondary power is used as a baseline instead of primary power. A typical power calculation for a CANDU reactor is determined in Eqs (34, 35):

$$\begin{aligned}\text{RTP} = m_{fw} \cdot (h_{ms} - h_{fw}) + m_{rh} \cdot (h_{ms} - h_{rh}) + m_{bd} \cdot (h_{bd} - h_{fw}) \\ + Q_{misc} - Q_p\end{aligned} \tag{34}$$

$$\text{STP} = m_{fw} \cdot \left(h_{ms} - h_{fw}\right) + m_{rh} \cdot (h_{ms} - h_{rh}) + m_{bd} \cdot (h_{bd} - h_{fw}) \tag{35}$$

Where Q_{misc} is ambient losses (kJ/h), Q_p is heat added by the reactor coolant pumps (kJ/h), m_{fw} is final feedwater flow (kg/h), h_{ms} is main steam enthalpy (kJ/kg), h_{fw} is final feedwater enthalpy (kJ/kg), m_{bd} is blowdown mass flow rate (kg/h), h_{bd} is enthalpy of blowdown (kJ/kg), m_{rh} is reheater drain flow to the reactor (kg/h) and h_{rh} is reheater drain enthalpy (kJ/kg).

Thermal power calculations use several redundant layers of instrumentation with varying degrees of accuracy to ensure that licensed limits are observed. Tables 8 and 9 list the main measurements for BWR and PWR plants with their sensitivities of 1 % errors to the RTP, respectively. These data are the results of sensitivity analysis for a specific BWR unit. CANDU plants measure the same parameters as PWRs with the addition of the reheater drain line temperature and flow.

TABLE 8. BWR THERMAL POWER INPUT SENSITIVITY

BWR Thermal power calculation parameter error sensitivity, 1% error	
Parameter	**Sensitivity**
Final feedwater flow	1.0006 %
Final feedwater temperature	-0.5886 %
Reactor pressure	-0.0501 %
Final feedwater pressure	-0.0014 %
Main steam quality	-0.8196 %
Reactor water clean-up inlet temperature	0.0077 %
Reactor water clean-up outlet temperature	-0.0058 %
Reactor water clean-up flow	0.0011 %
Recirculation pump power	-0.0017 %

TABLE 9. PWR THERMAL POWER INPUT SENSITIVITY

PWR Thermal power calculation parameter error sensitivity, 1% error	
Parameter	**Sensitivity**
Final feedwater flow	0.9968 %
Final feedwater temperature	-0.6221 %
Main steam pressure	-0.0296 %
Final feedwater pressure	-0.0010 %
Main steam quality	-0.8847 %
Steam generator blowdown flow	0.0032 %
Steam generator blowdown temperature	-0.0008 %
Steam generator blowdown quality	-0.0002 %

From Tables 8 and 9 we can recognize that final feedwater flow, final feedwater temperature and main steam quality are significant contributors to measurement error on the RTP. Field experience of performance monitoring shows that the most common and serious cause of overestimation of the RTP is fouling problems on the differential pressure flow meters such as venturis or orifice plates. Typically, venturi fouling leads to overestimation of the RTP, while orifice fouling causes underestimation of the RTP.

Another common cause of the RTP measurement drift is errors on final feedwater temperature measurement. For example, 1 °C error on the final feedwater temperature will result in around 0.3% error on the RPT and resultantly the generator power output will be increased or decreased by 0.32% depending on direction of its drift.

In case of the main steam quality a constant input value from the previous moisture carryover test is used for calculation of the RTP. It is not a common occurrence, but once a failure of the moisture removal devices inside the SG (PWR) or the reactor (BWR) occurs, it may introduce errors to the calculation of RTP.

4.2. DETERMINATION OF UNCERTAINTY

Calculation of the reactor power uncertainty typically uses a method to evaluate the uncertainty of a calculated result that has multiple input parameters. This methodology will account for the error propagation between inputs and results. This section discusses a methodology to evaluate the uncertainty as well as a general discussion of the uncertainty of RTP with an example calculation of BWR plants.

This calculation example uses the method outlined in Section 7 of ASME PTC 19.1 [19] to evaluate the uncertainty of RTP. Section 7 of ASME PTC 19.1 [19] states that for a result R that has a set of n independent inputs [z_1, z_2,...,z_n], the uncertainty of R can be evaluated as shown in Eq. (36).

$$\varepsilon_R = \sqrt{\left(\frac{\partial R}{\partial z_1} \cdot \frac{z_1}{R} \cdot \varepsilon_{z_1}\right)^2 + \left(\frac{\partial R}{\partial z_2} \cdot \frac{z_2}{R} \cdot \varepsilon_{z_2}\right)^2 + \ldots + \left(\frac{\partial R}{\partial z_n} \cdot \frac{z_n}{R} \cdot \varepsilon_{z_n}\right)^2} \tag{36}$$

Where ε_R is the uncertainty of result R (% R), R is the numerical value of result R, z_n is the numerical value of the input z_n, ε_{Zn} is the uncertainty of input z_n (% z_n), $(\partial R/\partial z_n)*(z_n/R)$ is the relative sensitivity factor for input z_n.

The uncertainty propagation method shown in Eq. (36) applies only to random uncertainties. If any of the input uncertainties being applied to Eq. (36) has bias components, these biases are supposed to be assessed separately. This calculation uses a formula presented in ISA RP67.04.02-2000 [20].

Given a result R that has an independent set of n inputs [z_1, z_2,…,z_n], let each input have an uncertainty such that $\varepsilon_{Zn,Total}$= [$\varepsilon_{Zn} \pm \alpha_{Zn}$], where $\pm\alpha_{Zn}$ is a potential one-sided bias uncertainty term. The total impact of these individual bias terms on the result R can be evaluated as shown in Eq. (37).

$$\alpha_R = \left(\frac{\partial R}{\partial z_1} \cdot \frac{z_1}{R} \cdot \alpha_{z_1}\right) + \left(\frac{\partial R}{\partial z_2} \cdot \frac{z_2}{R} \cdot \alpha_{z_2}\right) + \ldots + \left(\frac{\partial R}{\partial z_n} \cdot \frac{z_n}{R} \cdot \alpha_{z_n}\right) \tag{37}$$

Where α_R is the total bias uncertainty for result R (% R), R is the numerical value of result R, z_n is the numerical value of the input z_n, α_{zn} is the bias uncertainty for input z_n (% z_n), $(\partial R/\partial z_n)*(z_n/R)$ is the relative sensitivity factor for input z_n.

Note: Positive and negative bias terms need to be assessed separately.

4.2.1. Technical inputs

The inputs to the uncertainty calculation need to be defined and substantiated for example:

- The engineering standards manual that provides plant specific guidance on the implementation of the Reactor Supplier guidelines and methodology.
- ASME PTC 19.1-2003 [19] 'Test Uncertainty'. ASME PTC 19.1 [19] describes a method for evaluating the uncertainty of a calculated result is the source for Equation 1.
- Software Specification that provides the software description of the Reactor Power Uncertainty.
- The calculations evaluating all the instruments used as inputs to the Reactor Power Uncertainty calculation.

4.2.2. Assumptions

All assumptions need to be identified, and the basis justified. See the following examples

- This calculation assumes a value of 0.9995 for the quality of the steam exiting the reactor. This is the value recommended by the Software Requirement Specification (SRS) document. An uncertainty of ±0.05% is assumed for this value based on its deviation from 1.0000.
- The recirculation pump has an assumed efficiency of 0.9. An uncertainty of ±5.00% is assumed for this value.
- The radiative heat loss from the reactor to the surrounding environment has an assumed value of 2532.134 kJ/h based on the value used by the SRS. An uncertainty of ±20.00% is assumed for this value.
- This calculation will separately assess both the positive and negative potential bias uncertainty terms as shown by Eq. (33). However, in order to simplify the final RTP uncertainty, this calculation will assume that the larger one-sided bias uncertainty term is symmetrical. This will allow the bias term to be added directly to the total random uncertainty term calculated with Eq. (32) and the final uncertainty can be written as a single '±' value. This simplification is conservative because it replaces the smaller bias term with a larger value.

4.2.3. Input definition and determination of input error terms

The equations used by the plant computer to determine the RTP are defined in the SRS. For the purposes of this calculation several of the separate calculation steps from the SRS are combined. These combined calculations are shown in Eq. (38).

$$MWth_{reac} = \begin{pmatrix} \begin{bmatrix} CF_{Flow} \cdot m_{FW} \cdot (h_{MS} - h_{FFW}) + m_{CRD} \cdot (h_{MS} - h_{CRD}) \\ + m_{RWCU} \cdot (h_{RWCUin} - h_{RWCUout}) + Q_{rad} \end{bmatrix} \\ -\left[\eta_{pump} \cdot MW_{pump}\right] \end{pmatrix} \tag{38}$$

Where $MWth_{reac}$ is the reactor thermal power (MWth), m_{FW} is the final feedwater flow (kg/h), m_{CRD} is the control rod drive flow (kg/h), m_{RWCU} is the reactor water clean-up flow (kg/h), CF_{Flow} is the ultrasonic flow meter correction factor for venturis, h_{MS} is the main steam enthalpy (kJ/kg) based on X_{MS} and $P_{RctrPress}$ (see the inputs description below), h_{FFW} is the final feedwater enthalpy (kJ/kg) based on T_{FFW} and P_{FFW} (see the inputs description below), h_{CRD} is the control rod drive enthalpy (kJ/kg) based on T_{CRD} and $P_{RctrPress}$ (see the inputs description below), h_{RWCUin} is the RWCU enthalpy into reactor (kJ/kg) based on T_{RWCUin} and P_{RWCU} (see the inputs description below), $h_{RWCUout}$ is the RWCU enthalpy out of reactor (kJ/kg) based on $T_{RWCUout}$ and P_{RWCU} (see the inputs description below), Q_{Rad} is the radiation heat loss to environment (kJ/h), MW_{Pump} is the power used by recirc pump (Mwe), η_{Pump} is the efficiency of recirculation pump, $P_{RctrPress}$ is the reactor pressure (bar,a).

A further description of the inputs is provided below:

m_{FW} *(final feedwater flow)* – Final feedwater flow is measured by two venturis on two separate final feedwater loops leading to the reactor. The flows of both venturis are summed using plant computer point to determine the final feedwater flow. The full uncertainty analysis of the final feedwater flow at can be found in a separate calculation. The total uncertainty for feedwater flow is ±18.221 kg/h + 23.859 kg/h. The potential bias term 23.859 kg/h has two components. There is +2.830 kg/h potential bias caused by the plant computer input uncertainty. A value of

+21.029 kg/h potential bias is due to the effects of venturi fouling. This fouling bias term can be eliminated if the correction factor (CF_{Flow}) is used. If CF_{Flow} is used, then the feedwater flow uncertainty becomes ±18.221 kg/h +2.830 kg/h.

m_{CRD} *(control rod drive flow)* – Control rod drive (CRD) flow is measured by a venturi flow meter and the calculated flow value is taken from plant computer point. The full uncertainty analysis of the control rod drive flow is in a separate calculation. The final uncertainty includes a potential bias of +31.8 kg/h from the plant computer input error and a symmetrical uncertainty of ±565.2 kg/h. The total uncertainty for control rod drive flow is ±565.2 kg/h + 31.8 kg/h. The CRD flow uncertainty calculation states the total uncertainty for control rod drive flow: ±565.2 kg/h + 31.8 kg/h. The potential bias of +31.8 kg/h is from the plant computer input error.

m_{RWCU} *(reactor water clean-up flow)* – The reactor water clean-up (RWCU) flow is measured by orifice flow meters and the calculated flow values are taken from plant computer points. The full uncertainty analysis for the RWCU flows is in another calculation. The total uncertainty for RWCU flow is ±596.9 kg/h, +39.9 kg/h, -61.7 kg/h.

CF_{Flow} *(ultrasonic flow meter correction factor for venturis)* – The correction factor is used to remove the effects of fouling from the final feedwater flow venturi measurements. CF_{Flow} is the ratio of the measured flow to the venturi flow. Since the meter was calibrated to the venturi in its un-fouled state, CF_{Flow} corrects the venturi to this un-fouled state. The uncertainty of CF_{Flow} is evaluated in in a separate calculation. The calculate uncertainty is ±0.3709% flow for Feedwater Loop A and ±0.3585% flow for Feedwater Loop B.

Q_{rad} *(radiative heat loss to environment)* – As discussed in Section 4.2.2 the radiative heat loss from the reactor to the surrounding environment has an assumed value of 2532.134 kJ/h based on the value used by the SRS document. An uncertainty of ±20% is assigned to this value.

MW_{Pump} *(power used by recirculation pump)* – The power used by the recirculation pump motor is the total measured by megawatt transducers on two separate pump units. The pump power is taken from plant computer points. The full uncertainty analysis for the pump is in a separate calculation. The total uncertainty for pump power is ±128.24 kW + 7.84 kW. The potential bias of +7.84 kW is from the plant computer input error calculation.

η_{Pump} *(efficiency of recirculation pump)* – The recirculation pump has an efficiency of 0.9. This is the value used by the SRS. An uncertainty of ±5.00% is assumed for this value.

$P_{RctrPress}$ *(reactor pressure)* – The reactor pressure in bar is taken from plant computer point and is adjusted to absolute pressure (bar) using the barometric pressure reading from plant computer point from the meteorological or 'met' tower. The full uncertainty analysis for reactor pressure and the barometric pressure are in a separate calculation. The total reactor pressure uncertainty is ±5.341% (+0.169-0.291)% and barometric pressure uncertainty is ±0.00576 bar + 0.00085 bar.

T_{FFW} *(final feedwater temperature)* – +/-The final feedwater temperature is a calculated value for each feedwater loop and is taken from plant computer points. The full uncertainty analysis of the final feedwater temperatures taken from the computer points is in a separate calculation. The final feedwater temperature uncertainty is ±1.5°C. The alternate feedwater temperature uncertainty is ±0.54°C for Loop A and ±0.54°C for Loop B.

P_{FFW} *(final feedwater pressure)* – The full uncertainty analysis of final feedwater pressure is in a separate calculation. The final feedwater pressure uncertainty is ±1.602 bar + 0.197 bar.

T_{CRD} *(control rod drive temperature)* – The total control rod drive pump discharge temperature uncertainty of ±3.28 °C.

T_{RWCUin} *(reactor water clean-up temperature at inlet)* – The total RWCU inlet temperature uncertainty of ±3.28 °C.

$T_{RWCUout}$ *(reactor water clean-up temperature at outlet)* – The total RWCU outlet temperature uncertainty of ±3.28 °C.

P_{RWCU} *(reactor water clean-up pressure)* – The total RWCU pressure uncertainty of ±5.516 bar about a nominal value of 75.842 bar, a.

X_{MS} *(quality of main steam)* – The Quality of Main Steam has an assumed value of 0.9995 with an uncertainty of ±0.05% based on its deviation from 1.0000.

4.2.4. Determination of loop/channel uncertainty value

4.2.4.1. Channel random uncertainty

The general uncertainty formula in Eq. (32) can be applied to the RTP formula shown in Eq. (34) with a few modifications.

The first modification is to the inputs list from Eq. (34). These inputs include several enthalpy terms. Each of these enthalpy terms has its own pair of inputs used on a steam table to look up the enthalpy value. The inputs list for Eq. (32) is altered to include these enthalpy inputs and not the actual enthalpies. This publication assumes that any uncertainty introduced by the steam tables is negligible. Thus, the enthalpy uncertainties are affected only by the uncertainties of their inputs and leaving the enthalpies out of the uncertainty evaluation is justified.

The relative sensitivity factors (see Eq. (32)) for the RTP uncertainty are approximated with a sensitivity study. The sensitivity study determines the impact that a small change in each input has on the result R (RTP). The term $\partial R/\partial z_n$ in Eq. (32) effectively becomes $\Delta R/\Delta z_n$ where Δz_n is a small change in the input z_n and ΔR is the resulting change in RTP.

A summary of the sensitivity study can be seen in Figure 52. The values of all the inputs used in the sensitivity study and the equations used are based on those shown in SRS document.

Eq, (38) can be re-written for the RTP calculation with the afore-mentioned modifications. The result is shown in Eq. (39).

Input	Input Description	Baseline Thermal Power (Mwth)	Thermal Power With 1% Input Increase	Sensitivity (% Change in Thermal Power)
P_{RWCU}	Reactor Water Clean up pressure	1772.432	1772.431183	-0.0000203
$P_{RctrPress}$	RX PRESS NARROW RANGE		1771.635499	-0.0449125
$T_{RWCU\ in}$	RWCU INLET TEMP		1772.559333	0.0072099
$T_{RWCU\ out}$	RWCU OUTLET TEMP		1772.333512	-0.0055309
P_{atm}	BAROMETRIC PRESSURE		1772.420885	-0.0006013
P_{FFW}	FINAL FW PRESS		1772.401402	-0.0017005
$W_{FWTotal}$	TOTAL FW FLOW		1790.063876	0.9948104
$T_{CRD\ Temp}$	CRD DISCH TEMP		1772.422522	-0.0005089
W_{CRD}	CRD FLOW		1772.523282	0.0051759
MW_{Pump}	RECIRC PUMP POWER A		1772.418689	-0.0007252
	RECIRC PUMP POWER B		1772.418736	-0.0007226
W_{RWCU}	RWCU FLOW A		1772.448059	0.0009319
	RWCU FLOW B		1772.448025	0.0009299
η_{pump}	Pump Efficiency Constant		1772.405882	-0.0014478
Q_{rad}	Reactor Thermal Loss		1772.438576	0.0003968
X_{MS}	MAIN STEAM QUALITY		1758.858875	-0.7657654
T_{FFW}	FW NOZ TEMP A		1767.009701	-0.3058985
	FW NOZ TEMP B		1767.025711	-0.3049952
CF_{XFlow}	Venturi Correction Factor (Loop A)		1781.277993	0.4991138
	Venturi Correction Factor (Loop B)		1781.217425	0.4956966

FIG. 52. Example summary of sensitivity analysis

$$\varepsilon_{MWth} = \sqrt{\begin{aligned}&\left(\theta_{CF_{Flow}}\cdot\varepsilon_{CF_{Flow}}\right)^2+\left(\theta_{m_{FWtotal}}\cdot\varepsilon_{m_{FFWTotal}}\right)^2+\left(\theta_{X_{MS}}\cdot\varepsilon_{X_{MS}}\right)^2+\left(\theta_{T_{FFW}}\cdot\varepsilon_{T_{FFW}}\right)^2+\left(\theta_{P_{FFW}}\cdot\varepsilon_{P_{FFW}}\right)^2\\&+\left(\theta_{P_{atm}}\cdot\varepsilon_{P_{atm}}\right)^2+\left(\theta_{P_{RctrPress}}\cdot\varepsilon_{P_{RctrPress}}\right)^2+\left(\theta_{m_{CRD}}\cdot\varepsilon_{m_{CRD}}\right)^2+\left(\theta_{T_{CRDTemp}}\cdot\varepsilon_{T_{CRDTemp}}\right)^2\\&+\left(\theta_{m_{RWCU}}\cdot\varepsilon_{m_{RWCU}}\right)^2+\left(\theta_{T_{RWCU\,in}}\cdot\varepsilon_{T_{RWCUin}}\right)^2+\left(\theta_{P_{RWCU}}\cdot\varepsilon_{P_{RWCU}}\right)^2+\left(\theta_{T_{RWCUOut}}\cdot\varepsilon_{T_{RWCUout}}\right)^2\\&+\left(\theta_{Qrad}\cdot\varepsilon_{Qrad}\right)^2+\left(\theta_{MW_{pump}}\cdot\varepsilon_{MW_{pump}}\right)^2+\left(\theta_{\eta_{pump}}\cdot\varepsilon_{\eta_{pump}}\right)^2\end{aligned}} \quad (39)$$

$$\Theta_z = \frac{\Delta MW_{th}}{\Delta z}\cdot\frac{z}{MW_{th}} \quad (40)$$

Eq. (40) express Θ_z *which* is the sensitivity factor for input *z* where *z* label corresponds to one of the uncertainty terms (%MWth /%z).

Where ε_{MWth} is the total random uncertainty of reactor thermal power (% MWth), ε_{CFflow} is the uncertainty of ultrasonic flow meter correction factor (% CF_{flow}), $\varepsilon_{mFW\ Total}$ is the uncertainty of total feedwater flow (% $m_{FW\ Total}$), ε_{XMS} is the uncertainty of main steam quality (% X_{MS}), ε_{TFFW} is the uncertainty of final feedwater temperature (% T_{FFW}), ε_{PFFW} is the uncertainty of final feedwater pressure (% P_{FFW}), ε_{Patm} is the uncertainty of barometric pressure (% P_{atm}), $\varepsilon_{PRctr\ Press}$ is the uncertainty of reactor steam pressure (% $P_{Rctr\ Press}$), ε_{mCRD} is the uncertainty of CRD flow (% m_{CRD}), ε_{TCRD} is the uncertainty of CRD pump discharge temperature (% T_{CRD}), ε_{mRWCU} is the uncertainty of reactor water clean-up flow (% m_{RWCU}), $\varepsilon_{TRWCU\ in}$ is the uncertainty of reactor water clean-up inlet temperature (% $T_{RWCU\ in}$), ε_{PRWCU} is the uncertainty of reactor water clean-up pressure (% P_{RWCU}), $\varepsilon_{TRWCU\ out}$ is the uncertainty of reactor water clean-up outlet temp (% $T_{RWCU\ out}$),ε_{Qrad} is the uncertainty of radiation heat loss (% Q_{rad}), ε_{MWpump} is the uncertainty of power used by recirculation pump (% MW_{pump}), $\varepsilon_{\eta pump}$ is the uncertainty of recirculation pump efficiency (% η_{pump}).

4.2.4.2. Channel bias uncertainty

The general bias uncertainty formula in Eq. (37) can be applied to thermal power formula from Eq. (38). Only the inputs that have bias terms, as shown in Eq. (41) need to be included.

$$\begin{aligned}\alpha_{MWth} = \left(\theta_{m_{FWtotal}} \cdot \alpha_{m_{FWTotal}}\right) + \left(\theta_{P_{FFW}} \cdot \alpha_{P_{FFW}}\right) + \left(\theta_{P_{atm}} \cdot \alpha_{P_{atm}}\right) \\ + \left(\theta_{P_{Rctr\text{Press}}} \cdot \alpha_{P_{Rctr\text{Press}}}\right) \\ + \left(\theta_{m_{CRD}} \cdot \alpha_{m_{CRD}}\right) + \left(\theta_{m_{RWCU}} \cdot \alpha_{m_{RWCU}}\right) + \left(\theta_{MW_{pump}} \cdot \alpha_{MW_{pump}}\right)\end{aligned} \tag{41}$$

Where α_{MWth} is the total bias uncertainty of reactor thermal power (% MWth), $\alpha_{mFW\,Total}$ is the potential bias of total feedwater flow (% $m_{FW\,Total}$), α_{PFFW} is the potential bias of final feedwater pressure (% P_{FFW}), α_{Patm} is the potential bias of barometric pressure (% P_{atm}), $\alpha_{PRctr\,Press}$ is the potential bias of reactor steam pressure (% $P_{Rctr\,Press}$), α_{mCRD} is the potential bias of CRD Flow (% m_{CRD}), α_{mRWCU} is the potential bias of reactor water clean-up flow (% m_{RWCU}), α_{MWpump} is the potential bias of power used by recirculation pump (% MW_{pump}) and Θ_z is the sensitivity factor for input z where z label corresponds to one of the uncertainty terms (%MWth /%z).

Note: Equation (41) needs to be evaluated at the largest absolute value for each uncertainty term. The terms with positive biases will be summed independently from the terms with negative biases. The largest absolute value of these two values will be used as a '±' uncertainty for this calculation.

4.2.4.3. Total channel uncertainty

The total RTP uncertainty is found by combining the results of Eq. (39) and Eq. (41). The larger of the two bias uncertainty terms calculated with Eq. (41) is treated as a symmetrical '±' uncertainty term and the smaller bias term is ignored. The total random uncertainty and total bias uncertainty are combined as shown in Eq. (42).

$$\varepsilon_{MWthTotal} = \pm(\varepsilon_{MWth} + \alpha_{MWth}) \tag{42}$$

Where $\varepsilon_{MWthTotal}$ is the total uncertainty of reactor thermal power (% Mwth), ε_{MWth} is the total random uncertainty of reactor thermal power (% MWth) and α_{MWth} is the total bias uncertainty of reactor thermal power (% MWth).

4.2.4.4. Reactor thermal power uncertainty computation

There are four separate evaluations based on whether the ultrasonic flow and temperature measurements are activated. The results are shown in the Table 10 below.

TABLE 10. FIVE EXAMPLE RESULTS OF THE RTP UNCERTAINTY CALCULATION

	Random uncertainty of total feedwater flow (%)	**Bias uncertainty of total feedwater flow (%)**	**Total uncertainty of thermal power (%)**
UFM disabled, UTM disabled	0.5021	0.6580	±1.3032%
UFM disabled, UTM enabled	0.5021	0.6580	±1.2354%
UFM enabled, UTM disabled	0.5021	0.0780	±0.7760%
UFM enabled, UTM enabled	0.5021	0.0780	±0.7138%

4.3. MEASUREMENT IMPROVEMENT (REGULATORY APPROVAL)

The overseeing regulatory bodies for NPPs have an interest in the maintaining of acceptable margins for the operation of NPPs. Historically, these margins have been discreet based on some overall estimated uncertainty for the calculation of core thermal power. More recently, NPPs achieved increased output based on improved measurement systems for the core thermal power inputs. Thus, the core thermal power license limits have been based on the actual calculated uncertainty of the measurement systems. These measurement systems have achieved uncertainty down to approximately 0.3%. Most of these types of uprates have been achieved by improving the measurement of feedwater flow and feedwater temperature.

Since using these more accurate measurement systems is the basis for the actual licensed power limits, failures or reduced performance of these systems will result in a power reduction. Use of the DVR technologies have been proposed as another method to achieve reduced uncertainty without reliance on single point failures.

The margin uncertainty recapture process has been initiated at NPP Gundremmingen B and C with approval of TÜV Süd and the Bavarian Environmental Ministry [21]. The Technical Inspection Association of Southwest Germany accepted the software for use at Germany's Neckarwestheim NPP (GKN2).

While most measurement systems used for core thermal power are not necessarily safety related they need to meet 'high commercial standards' similar to those described in [22]. The utility, in order to meet licensing requirements, needs to ensure the measurements remain accurate within the licensed uncertainty specified in the license.

Over the years a significant number of operating events related to potential operation in excess of licensed power have been reported by the utilities. The utilities have been required to analyse these events, report the results, and implement corrective measures; all with oversight by the country's regulatory body. It is important that the instrument systems that comprise the plant calorimetric heat balance have traceability to recognized national laboratories or calibration standards.

Therefore, if any methodology is proposed as a margin uncertainty recapture or alternative calorimetric method, unequivocal proof that the method is valid and reliable is warranted.

4.4. INDEPENDENT MONITORING OF THE REACTOR THERMAL POWER MEASUREMENT DRIFT

Up to the present, many NPPs are suffering from the RTP measurement drift which directly affect the generator power output. Any overestimation of the RPT, such as that resulting from the final feedwater flow measurement error, will lead to reduced generator power output. As a general guide, a 1% overestimation of the RTP will directly result in a reduction of 1% generator power output. On the other hand, any underestimation of the RTP impacts plant safety because the actual power will be higher than indicated with the measurement systems.

Nuclear power industry experiences highlighted the need for methodologies to monitor the RTP measurement drift and error. To address this need, an EPRI project was undertaken (in Phase 1) to identify the practices used by the TPEs when confronted with suspicions of error or drift of the RTP measurement and (in Phase 2) to compare four prevailing methods for monitoring and adjusting the RTP measurement drifts and to propose guidelines on the use of these methods. The results are presented in [23] and [24].

The following four methods in general use for monitoring the RTP measurement drift are presented in these reports, along with a discussion of their advantages and limitations.

- Trend analysis method;
- EDF's ΔP/P method;
- Best estimation method (River Bend calorimetric verification method);
- Data reconciliation method.

4.4.1. Trend analysis method

The trend analysis consists of monitoring key parameters in direct relation to the RTP. This method makes use of the turbine characteristic that turbine stage pressures are linearly proportional to steam flow passing each stage and resultantly feedwater temperature is also proportional to steam flow if the feedwater heater performance is maintained constant.

Because the RTP is supposed to be maintained constant at full load operation, trend analysis of the key parameters is considered a reliable indication of the RTP measurement drift. The diagnosis of this method compares the trended indication of key parameters to a predetermined threshold. For example, if the RTP is overestimated, the actual final feedwater flow is reduced while the indicating final feedwater flow is maintained constant.

In this case the following key parameters trend downward:

- Generator power output corrected for turbine back pressure;
- Steam generator outlet steam flow;
- HP turbine first-stage pressure;
- Feedwater extraction steam pressure;
- Feedwater temperature;
- Reactor cooling water ΔT.

In the case of RTP underestimation, the actual final feedwater flow is increased while the indicating final feedwater flow is maintained constant, and the above key parameters trend upward.

4.4.2. EDF's ΔP/P Method

EDF's ΔP/P method (Figure 53) is based on monitoring a single, relevant parameter (HP turbine first stage pressure) during the operating cycle between refuelling outages. This method also relies on the turbine characteristic that the HP turbine first stage pressure (P1st) is proportional to the HP turbine inlet steam flow (Qt). Because the P_{1st} is not influenced by the venturi fouling, the pressure sensor can detect a very small deviation due to the final feedwater flow drift.

The monitoring method is based on the deviation of the following quantity (pressure P and flow Q) between the beginning of the cycle and any other operating point during full load operation as described in Eq. (43):

$$E = \frac{dP1st}{P1st} - \frac{dQf}{Qf} = \frac{dQt}{Qt} - \frac{dQf}{Qf} \tag{43}$$

Where E is the deviation flow between inlet steam flow and final feedwater flow, Qf is the final feedwater flow (kg/h), Qt is the HP turbine inlet steam flow (kg/h) and $P1st$ is the HP turbine first stage pressure (kPa).

A threshold is defined to take the acceptable RTP measurement drift into account regarding the safety requirement and the method accuracy. This method assumes that the SG blowdown flow (Qb) and the auxiliary steam flow (Qaux) are constant at the same RTP and resultantly the Qt is directly linked to the final feedwater flow (Qf). The responsible engineers need to verify this assumption before quantifying the RTP measurement drift.

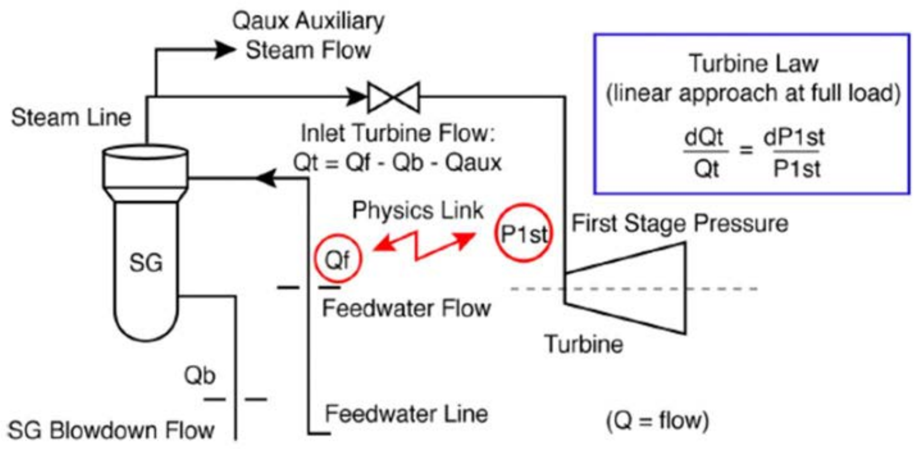


FIG. 53. EDF's ΔP/P method for the RTP measurement drift monitoring [25]

4.4.3. Best estimation method (River bend calorimetric verification method)

The best estimation method (data represented in Figure 54 and another example may be found in [26]), which is commonly referred to as the River Bend Method, makes use of sixteen key measurements as an indicator of the RTP on the turbine cycle. These parameters include HP turbine first stage pressure, main steam flow, feedwater flow (from venturis or orifice), MSR inlet and outlet pressures, feedwater heater extraction steam pressure and final feedwater temperature.

Whatever the primary flow measurement (main steam flow, ultrasonic flow meter (UFM), venturi, etc.) used in the calorimetric reactor power measurement, it is not supposed to be used in the River Bend method. The principle and implementation of this method is described in [24]. The general execution process is summarized below.

(a) Perform the verification test when the plant is perfectly stable at full load. Then calculate the average of the 16 key measurements over 2 hours with an acquisition update every 2 minutes.

(b) Determine the characteristic reference value of the 16 measurements at full load when the unit is running without any suspected RTP measurement drift.

(c) For these measurements, apply a proportional relation to calculate the reactor power during the test by comparing it with the reference power associated with the reference value of the measurement. For temperatures, since they are not 0°C at 0% power, replace the proportional relationship with a linear relationship calculated using a data historian. Based on the 16 key measurements, calculate the 16 reactor power values.

(d) Use the 16 key measurements' uncertainties to calculate 1/variance (1/uncertainty²) for each measurement. Then divide these terms by the sum of the 1/variance on all measurements. These values are called 'weighing factors' (or confidence factors) since they correspond to the confidence in the measurement according to its uncertainty. The lower the uncertainty, the greater the weighing factor.

(e) Calculate the reactor power value called '(River Bend) best estimate core thermal power' as the sum of the products of the power images of each of the 16 measurements multiplied by the confidence factor associated with the measurement.

(f) Carry out an uncertainty calculation of the best estimate core thermal power by propagation of the uncertainties of the 16 original measurements.

(g) For each of the 16 measurements, define a low and high limit to ensure that the measurement used is correct. If the measurement is not correct, it is rejected in the calculation of the reactor power, and maintenance is scheduled for the instrumentation.

(h) Perform the comparison between reactor power measurement and the (River Bend) best estimate core thermal power periodically (typically, days or weeks) to monitor any drift in reactor power measurement. The monitoring consists of comparing both powers according to their respective uncertainty.

The same precautions on the SG blowdown flow and the auxiliary steam flow are required, like the EDF ΔP/P method, to use this method for monitoring of the RTP measurement drift. This is because this method also assumes the flows are maintained constant at full load operation. In order to have an accurate diagnostic, it is also very important to have a reliable reference set point, which is referred to as baseline data at the beginning of each cycle.

			2012-05-15	2012-10-02								% Change Limit	3.00%
No	Parameter Description	Unit	BASELINE DATA	SAMPLED DATA	%Change	%MWt / %change	Best Estimate CTP, Pi	Uncertainty Ui	$(1/Ui)^2$	Weighting Factor, Wi	Wi * Pi	Wi * Ui	Overpower %
	Calorimetric Core Thermal Power	**MWt**	**1,868.58**	**1,874.33**	**-0.308%**			**1.50**					
1	Reactor Cooling System T_Average	℃	34.616	33.964	1.884%	1.1404	1,828.44	1.00	1.00	8.4%	154.24	0.084	-2.54%
2	Steam Generator Inlet Feedwater Flow	tons/hr	3,732.660	3,737.875	-0.140%	0.9363	1,871.03	2.50	0.16	1.3%	25.25	0.034	-0.27%
3	Steam Generator Outlet Steam Flow	tons/hr	3,609.834	3,549.404	1.674%	0.8480	1,842.06	1.00	1.00	8.4%	155.39	0.084	-1.81%
4	Main Feed Pump Discharge Flow	tons/hr	4,195.033	4,144.180	1.212%	0.9333	1,847.44	1.00	1.00	8.4%	155.84	0.084	-1.52%
5	Booster Pump Discharge Flow	tons/hr	4,086.950	4,037.132	1.219%	0.9802	1,846.26	1.00	1.00	8.4%	155.74	0.084	-1.59%
6	Condensate to Deaerator Flow	tons/hr	2,349.910	2,331.974	0.763%	0.9162	1,855.51	2.00	0.25	2.1%	39.13	0.042	-1.09%
7	Turbine First Stage Pressure (Average)	kg/cm²g	56.682	55.847	1.472%	0.9090	1,843.58	0.50	4.00	33.7%	622.07	0.169	-1.73%
8	HTR#6 Shell Pressure (Average)	kg/cm²g	24.670	24.480	0.771%	0.9837	1,854.41	1.00	1.00	8.4%	156.43	0.084	-1.15%
9	HTR#5 Shell Pressure (Average)	kg/cm²g	12.141	12.043	0.807%	0.9566	1,854.16	1.00	1.00	8.4%	156.41	0.084	-1.16%
10	MSR Outlet Pressure (Average)	kg/cm²g	12.206	12.092	0.933%	0.9444	1,852.12	1.50	0.44	3.7%	69.44	0.056	-1.27%
11	Final Feedwater Temperature (Average)	℃	222.345	221.704	0.288%	4.6012	1,843.79	1.00	1.00	8.4%	155.54	0.084	-1.72%
	Total								**11.85**	**100.0%**	**1,845.49**	**0.290**	

		Calorimetric	Best Estimate	
Core Thermal Power	MWt	1,874.33	1,845.49	
Uncertainty	%	1.500	0.290	
Uncertainty	MWt	28.115	5.360	
% Power (CTP/Rated)	%	99.91	98.37	1.54
Difference (Calorimetric-BECTP)	MWt		28.84	
Rated CTP	MWt	1876.00	1876.00	

FIG. 54. Best estimation method for the RTP measurement drift monitoring

4.4.4. Data reconciliation method

The key objective of the data reconciliation method is to take advantage of information redundancies coming from all process measurements.

When the process is within the nuclear turbine cycle, information redundancies are of two types:

- **Direct measurement redundancies**. These are directly redundant measurements (sensors) that measure the same physical value.
- **Redundancies resulting from physical relationships between measurements**. These are the existing physical relationships between measurements, in particular, heat and mass balance, efficiencies, and Stodola coefficients. These relationships result in generating more information that is strictly necessary to estimate the thermodynamic state of the fluid at each point of the process.

Direct measurement redundancies and redundancies resulting from physical relationship between measurements generate a set of equations that constitutes the process modelling. The system is generally over-determined (i.e. there are more equations than unknowns).

Figure 55 illustrates example equations of redundancy correlation to reconcile the final feedwater flow.

$$m_{FD1} = m_{sgo1} + m_{sgo2} + m_{bdn1} + m_{bdn2}$$

$$m_{FD2} = m_{ffw1} + m_{ffw2}$$

$$m_{FD3} = m_{fwp1} + m_{fwp2}$$

$$m_{FD4} = m_{ext7} + m_{rht2} + m_{ext6} + m_{ext5} + m_{rht1} + m_{msd} + m_{ext4} + m_{cond}$$

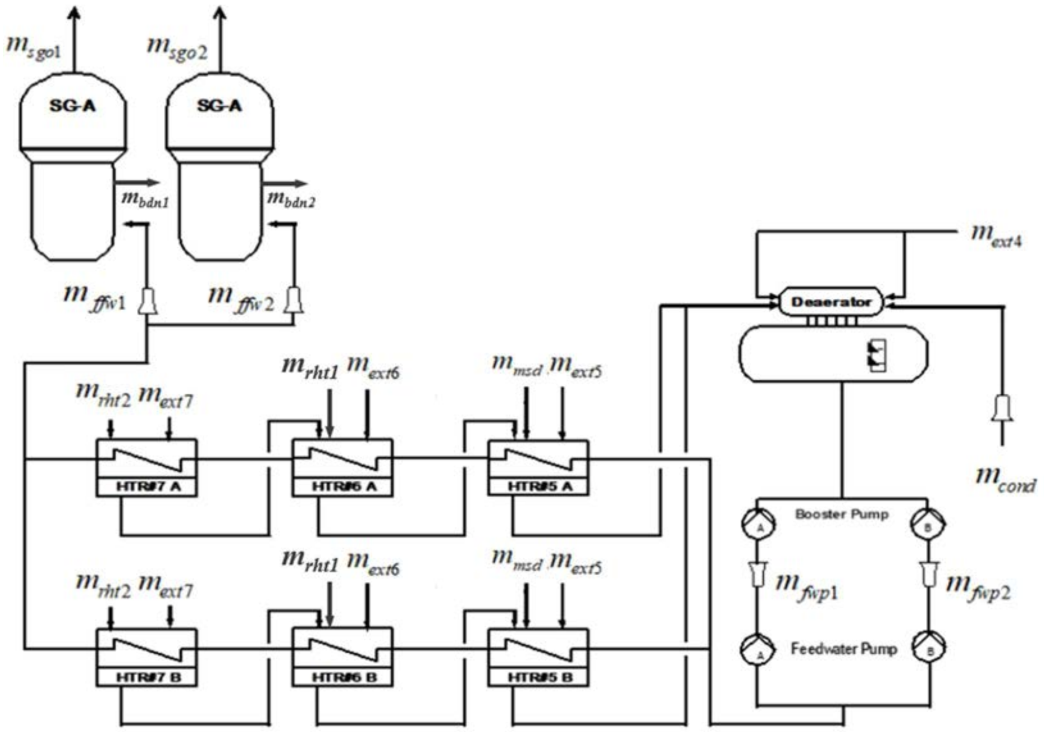


FIG. 55. Data reconciliation for the final feedwater flow

Where m_{FD} is a mass flow rate(kg/h), m_{sgo} is SG outlet mass flow rate (kg/h), m_{bdn} is a mass flow rate (kg/h), m_{ffw} is SG inlet final feedwater mass flow rate (kg/h), m_{fwp} is feedwater pump suction mass flow rate (kg/h), m_{ext} is LP turbine extraction mass flow rate (kg/h), m_{rht} is reheather drain mass flow rate (kg/h), m_{msd} is moisture separator drain mass flow rate(kg/h) and m_{cond} is condensate mass flow rate (kg/h).

The principle of the data reconciliation method is to correct each measurement as little as possible so that all the equations are strictly respected, regarding (if possible) the uncertainties that affect each measurement. With the assumption that the errors affecting measurements can be described by Gaussian Laws (independent or with known correlations) with known standard deviations, the search for the solution with the maximum probability consists of an optimization (minimization) under constraints (resulting from the information redundancies) of Eq. (44):

$$\sum_{i=1}^{n} penalty(i) = \sum_{i=1}^{n} \left(\frac{y_i^* - y_i}{\sigma_i}\right)^2 \tag{44}$$

Where y^*_i is the reconciled value (system unknown), y_i is the measured value and σ_i is the measurement uncertainty. Mathematical basis and example calculation of the data reconciliation method is presented in [18].

According to [24], the data reconciliation method according to [18] is the most useful because it allows a diagnosis at any power level and delivers the quality of all single measurements and of the calculation. There is an initial investment and effort to build and implement the data reconciliation model but after a test phase including improvement of erroneous measurements, the reconciliation application will run with minimal maintenance.

4.4.5. Case study – Reactor thermal power measurement drift caused by venturi fouling

This case study shows an example analysis of the RTP measurement drift in a 680MW PWR unit in Korea. In case of this unit, the HP feedwater heater tube bundles were replaced during the planned outage and after unit restart overestimation of the RPT was occurred with strong suspicion of venture fouling. The best estimation method is used to identify and trend the RTP measurement drift.

4.4.5.1. Phenomenon: Reactor thermal power vs generator power output mismatch

After restart of the unit from the planned outage, the RTP was increased from 99.4% to 100%, but the generator power output was remained constant. The RTP was kept constant after reaching 100%, but from that point the generator power output was continuously reduced. The performance indicators of the turbine cycle components, described in Section 5, and the cycle isolation was monitored and checked during the time period, but there was no indication which explains this power loss phenomenon. See Figure 56.

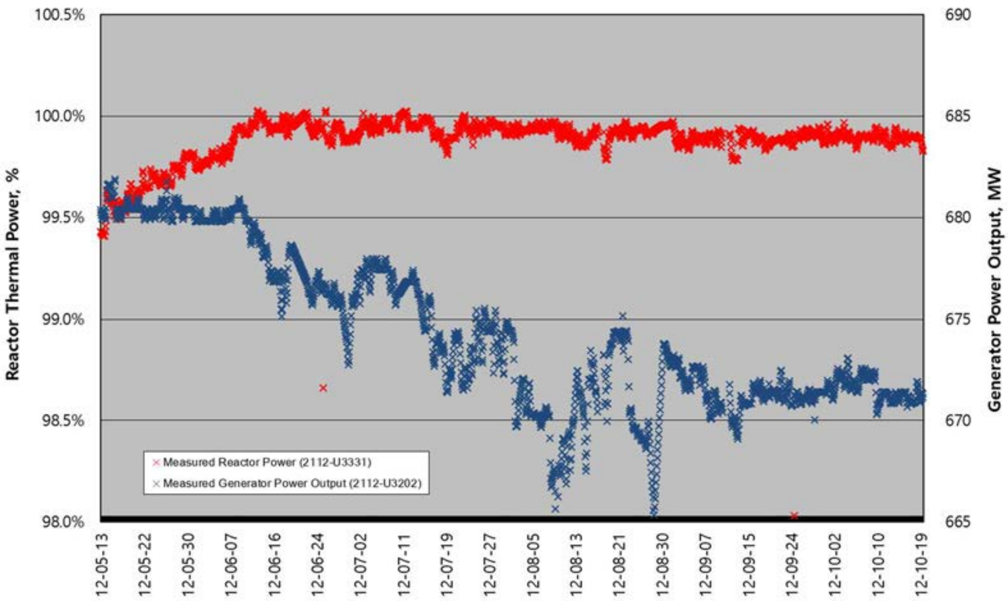


FIG. 56. RTP vs Generator Power Output

4.4.5.2. Slope of key measurement with reference to change in RTP

For the key measurements used to estimate the RTP with the Best estimation method, proportional relations between the key measurements and the RTP were determined (see Figures 57 – 61) making use of the plant instrument readings during unit start on May 13, 2012. These slopes were verified again during the unit shut down and restart respectively on August 17 and August 18, 2012.

Proportional relations were analysed for following key measurements and their slopes are summarized in Table 11.

- Final feedwater flow;
- Main feed pump discharge flow;
- Feedwater booster pump discharge flow;
- HP turbine first stage pressure;
- Feedwater heater 6 shell pressure;
- Feedwater heater 5 shell pressure;
- MSR outlet cycle steam pressure;
- Final feedwater temperature;
- Reactor cooling water system ΔT.

Section 4.4.3 mentions the 16 key parameters of the turbine cycle operation variable, but these parameters are specific for the River Bend NPP and can be changed depending on configuration of the turbine cycle.

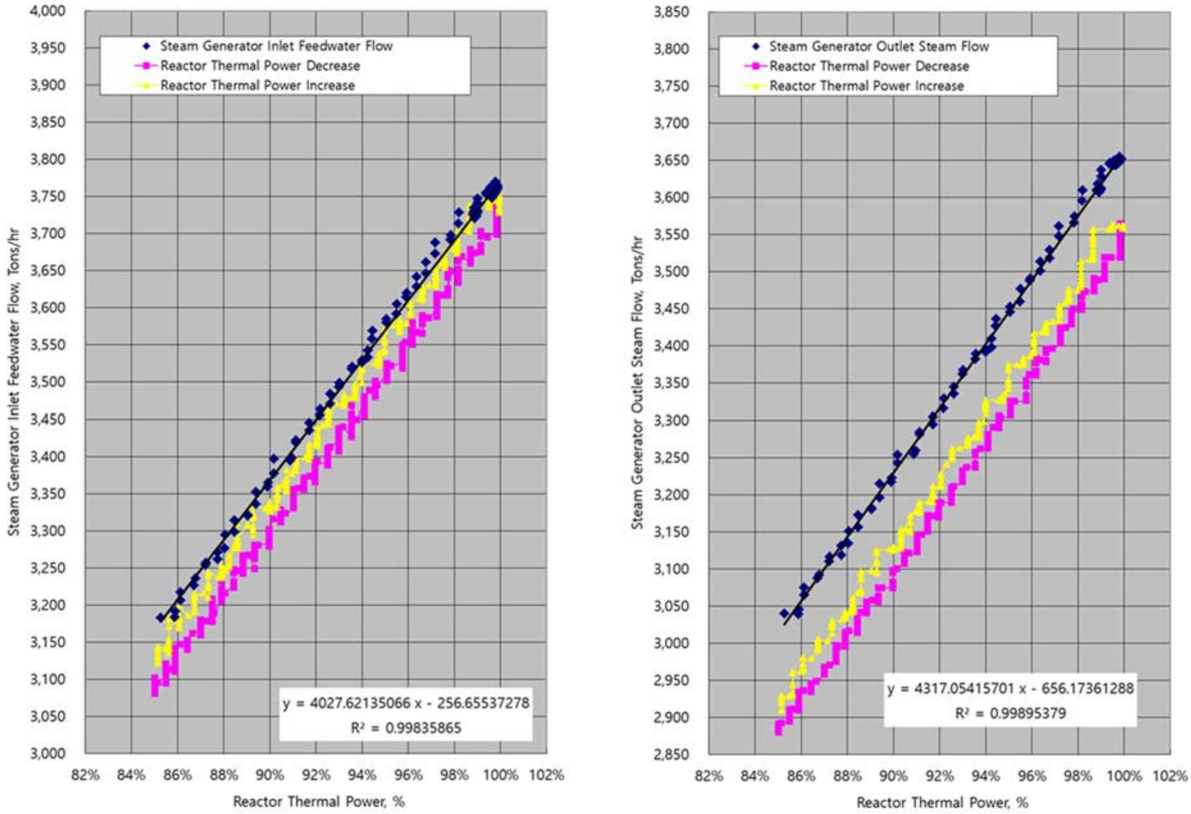


FIG. 57. RTP vs final feedwater flow(left); RTP vs SG outlet steam flow (right)

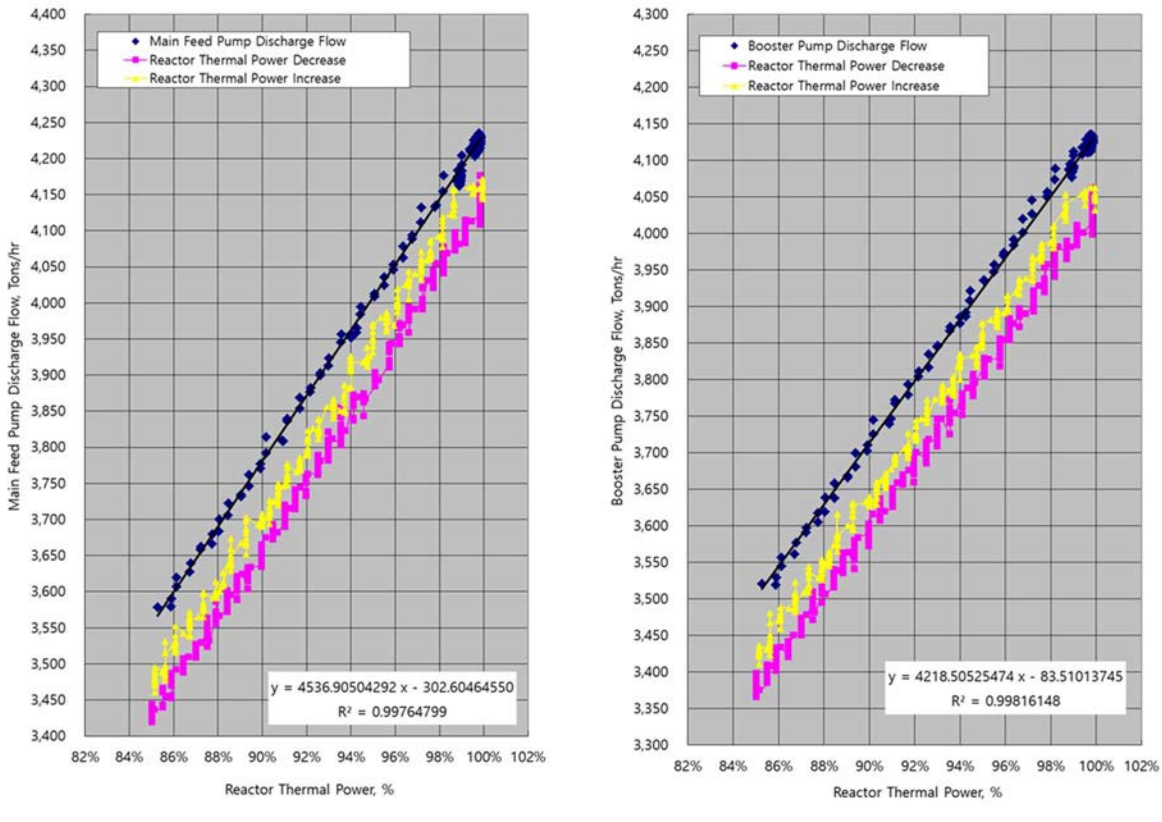


FIG. 58. RTP vs MFP discharge flow (left); RTP vs BP discharge flow (right)

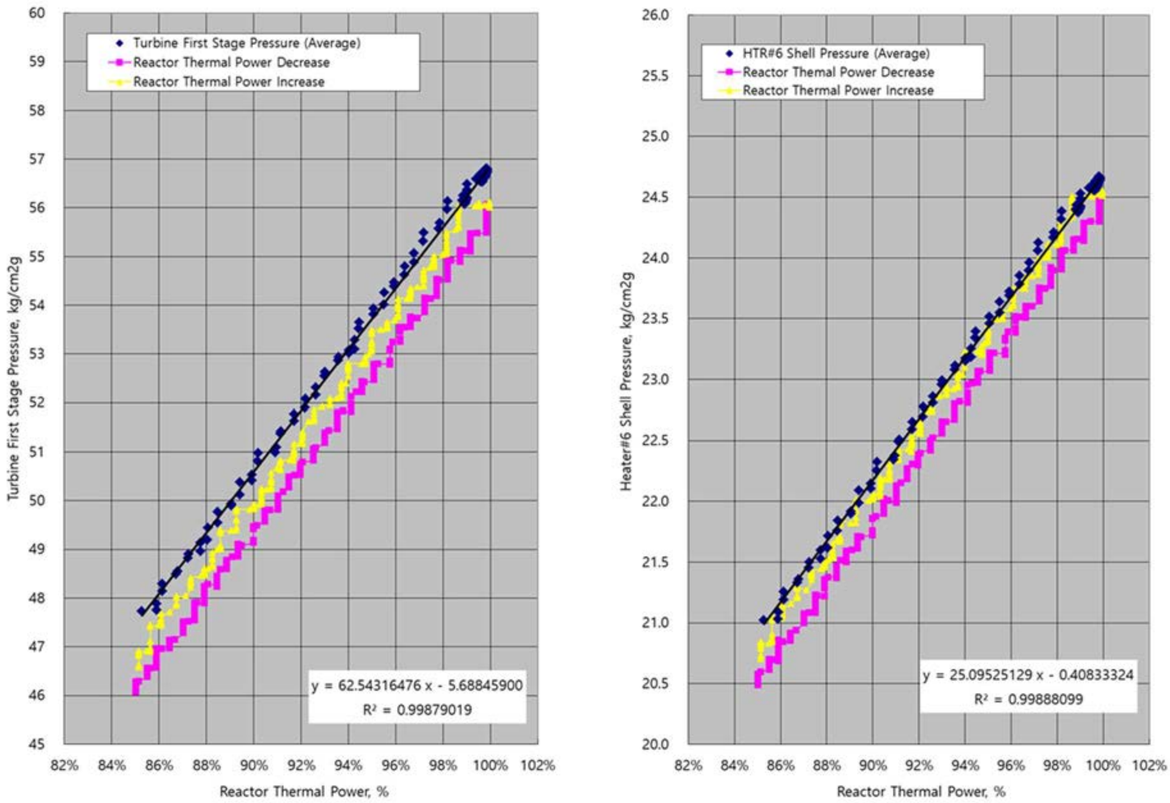


FIG. 59. RTP vs Turbine 1st stage Pressure(left); RTP vs HTR#6 Shell Pressure(right)

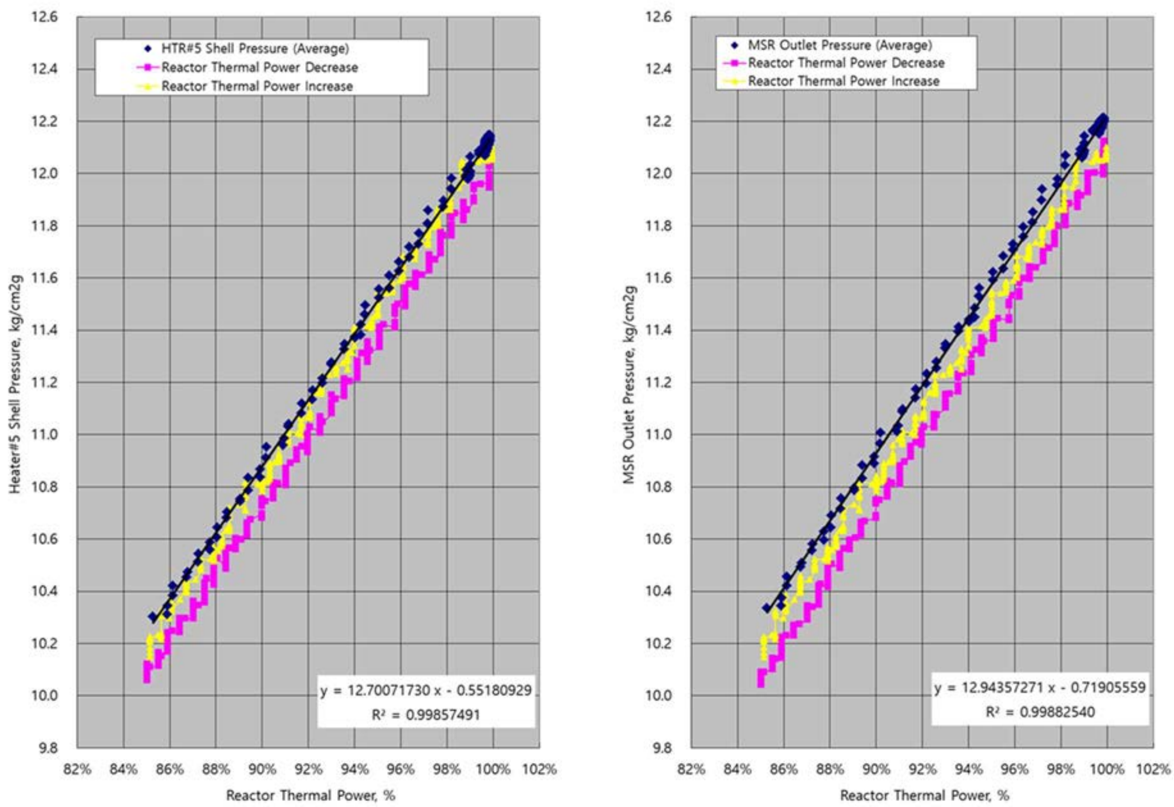


FIG. 60. RTP vs HTR#5 shell pressure(left); RTP vs MSR outlet pressure(right)

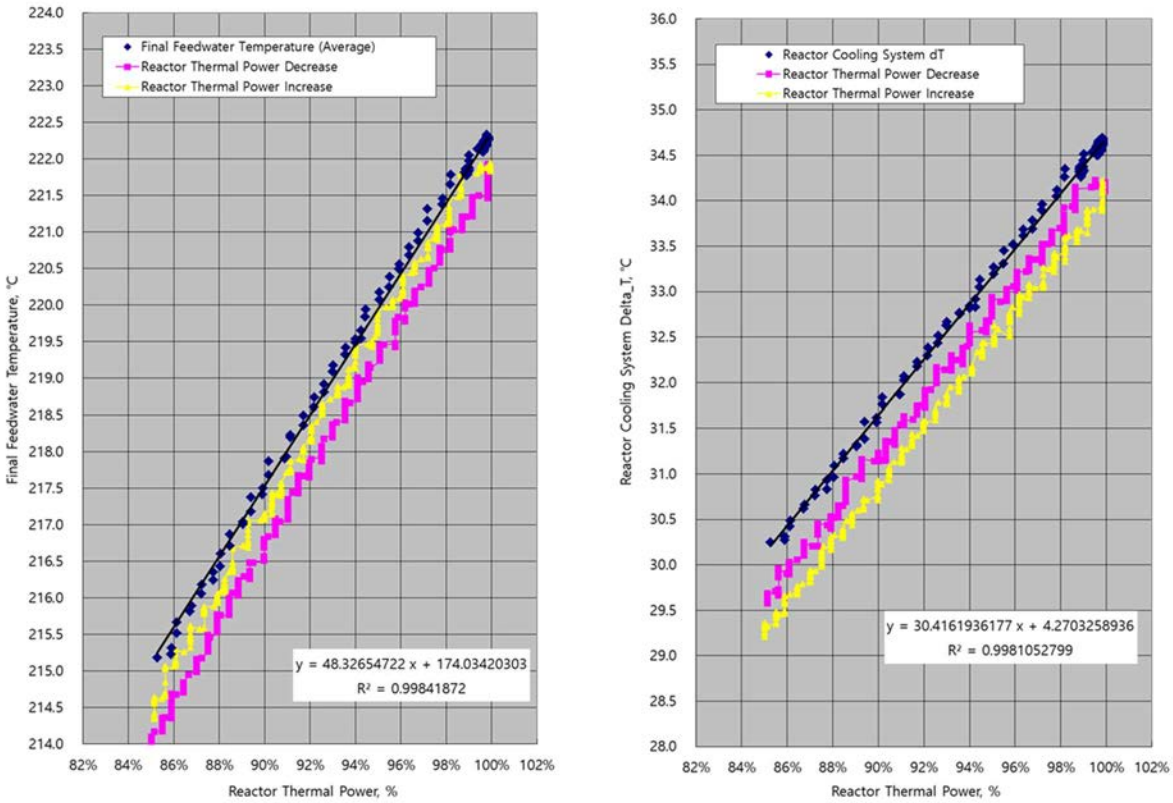


FIG. 61. RTP vs final feedwater temperature (left); RTP vs RCS ΔT (right)

TABLE 11. SUMMARY OF PROPORTIONAL RELATIONSHIP BETWEEN KEY MEASUREMENTS AND RTP

No	Operating parameters	Unit	Reactor thermal power		%Change	%MWth / %Change
			100%	95%		
1	Reactor cooling system delta T	℃	34.687	33.166	4.38%	1.1404
2	Steam generator inlet feedwater flow	t/h	3770.97	3569.58	5.34%	0.9363
3	Steam generator outlet steam flow	t/h	3660.88	3445.03	5.90%	0.8480
4	Main feed pump discharge flow	t/h	4234.30	4007.46	5.36%	0.9333
5	Booster pump discharge flow	t/h	4135.00	3924.07	5.10%	0.9802
6	Turbine first stage pressure (avg.)	kg/cm^2g	56.855	53.728	5.50%	0.9090
7	HTR#6 shell pressure (avg.)	kg/cm^2g	24.687	23.432	5.08%	0.9837
8	HTR#5 shell pressure (avg.)	kg/cm^2g	12.149	11.514	5.23%	0.9566
9	MSR outlet pressure (avg.)	kg/cm^2g	12.225	11.577	5.29%	0.9444
10	Final feedwater temperature (avg.)	℃	222.361	219.944	1.09%	4.6012

4.4.5.3. Calculation of the RTP measurement error and monitoring

Making use of two sets of key measurement on May 15, 2012 (baseline data) and on October 2, 2012 (sampled data) and pre-analysed slopes of each measurement, the estimated RTP (1845.25 MW_{th}) was determined and compared with the RTP (1874.33 MW_{th}) indicated on the nuclear control system. This result can be interpreted that the estimated RTP using the key measurements on October 2, 2012 is lower than the RTP directly measured from the final feedwater flow, and overestimation of the RTP has been occurred by 1.55% as shown in Figure 62.

2012-05-15 2012-10-02 % Change Limit 3.00%

No	Parameter Description	Unit	BASELINE DATA	SAMPLED DATA	%Change	%MWt / %change	Best Estimate CTP, Pi	Uncertainty Ui	$(1/Ui)^2$	Weighting Factor, Wi	Wi * Pi	Wi * Ui	Overpower %
	Calorimetric Core Thermal Power	**MWt**	**1,868.58**	**1,874.33**	**-0.308%**			**1.50**					
1	Reactor Cooling System T_Average	℃	34.616	33.964	1.884%	1.1404	1,828.44	1.00	1.00	8.3%	152.36	0.083	-2.54%
2	Steam Generator Inlet Feedwater Flow	tons/hr	3,732.660	3,737.875	-0.140%	0.9363	1,871.03	50.00	0.00	0.0%	0.06	0.002	-0.27%
3	Steam Generator Outlet Steam Flow	tons/hr	3,609.834	3,549.404	1.674%	0.8480	1,842.06	1.00	1.00	8.3%	153.50	0.083	-1.81%
4	Main Feed Pump Discharge Flow	tons/hr	4,195.033	4,144.180	1.212%	0.9333	1,847.44	1.00	1.00	8.3%	153.95	0.083	-1.52%
5	Booster Pump Discharge Flow	tons/hr	4,086.950	4,037.132	1.219%	0.9802	1,846.26	1.00	1.00	8.3%	153.85	0.083	-1.59%
6	Turbine First Stage Pressure (Average)	kg/cm^2g	56.682	55.847	1.472%	0.9090	1,843.58	0.50	4.00	33.3%	614.51	0.167	-1.73%
7	HTR#6 Shell Pressure (Average)	kg/cm^2g	24.670	24.480	0.771%	0.9837	1,854.41	1.00	1.00	8.3%	154.53	0.083	-1.15%
8	HTR#5 Shell Pressure (Average)	kg/cm^2g	12.141	12.043	0.807%	0.9566	1,854.16	1.00	1.00	8.3%	154.51	0.083	-1.16%
9	MSR Outlet Pressure (Average)	kg/cm^2g	12.206	12.092	0.933%	0.9444	1,852.12	1.00	1.00	8.3%	154.34	0.083	-1.27%
10	Final Feedwater Temperature (Average)	℃	222.345	221.704	0.288%	4.6012	1,843.79	1.00	1.00	8.3%	153.64	0.083	-1.72%
	Total								**12.00**	**100.0%**	**1,845.25**	**0.289**	

		Mesured	Best Estimate
Core Thermal Power	MWt	1,874.33	1,845.25
Uncertainty	%	1.500	0.289
Uncertainty	MWt	28.115	5.327
% Power (CTP/Rated)	%	99.91	98.36
Difference (Calorimetric-BECTP)	MWt		29.08
Rated CTP	MWt	1876.00	1876.00

1.55 ⟶ Overestimation 1.55%

FIG. 62. Example calculation of the RTP measurement error

The best estimation method originally uses the uncertainties of key measurements to determine the weighing factors. These factors correspond to the confidence of each measurement according to its uncertainty. But there are two limitations.

One limitation is that the uncertainties considered here are mostly theoretical values, instead of being issued from a detailed analysis or field test of the measurement chains. In the opinion of experts contributing to this document, obtaining technically traceable uncertainty values for each measurement seems impractical for the plant instruments readings. In many cases, plant instruments are not calibrated at an internationally accredited laboratory to achieve traceability and, even if calibrated, the readings are not typically corrected for data acquisition system errors.

The other limitation is the validity of using the turbine stage pressures downstream of the HP turbine first stage. For example, the HP turbine first stage pressure and the LP turbine inlet pressure have similar levels of measurement uncertainty, but the weighing factor for these two measurements is not supposed to be similar. This is because the HP turbine first stage pressure is directly linked and proportional to the SG outlet steam flow. Unlike the LP turbine, where inlet pressure is additionally affected by performance of the turbine cycle components, such as feedwater heaters or the feedwater pumping system. Accordingly, uncertainty of key measurements at downstream of the HP turbine first stage is supposed to be higher in order to decrease the corresponding weighing factor. This process requires engineering judgement to establish the uncertainty values.

Despite these limitations, the Best Estimation method is a powerful tool to monitor the RTP measurement drift and prepare corrective actions when it occurs. Trend analysis also shows consistency between the estimated RTP and the key measurements as illustrated in Figures 63 – 72. Trending of the RTP measurement drift over time is also illustrated in Figure 73.

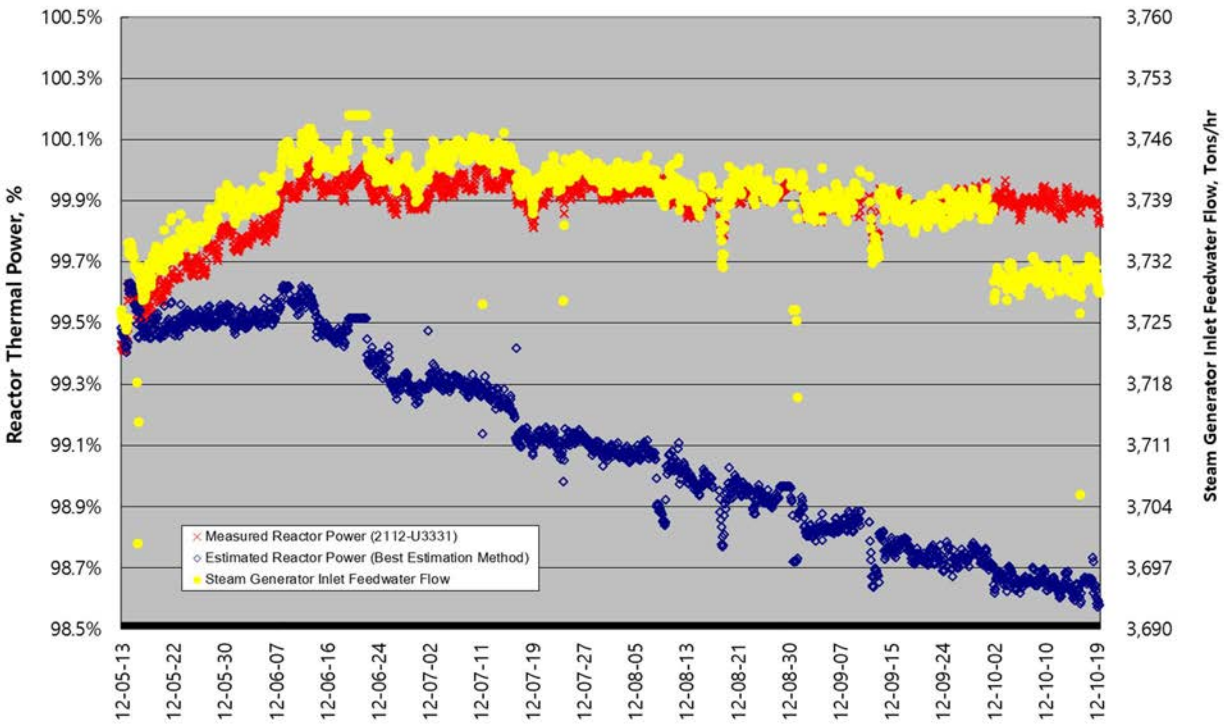


FIG. 63. Trending of RPT vs final feedwater flow

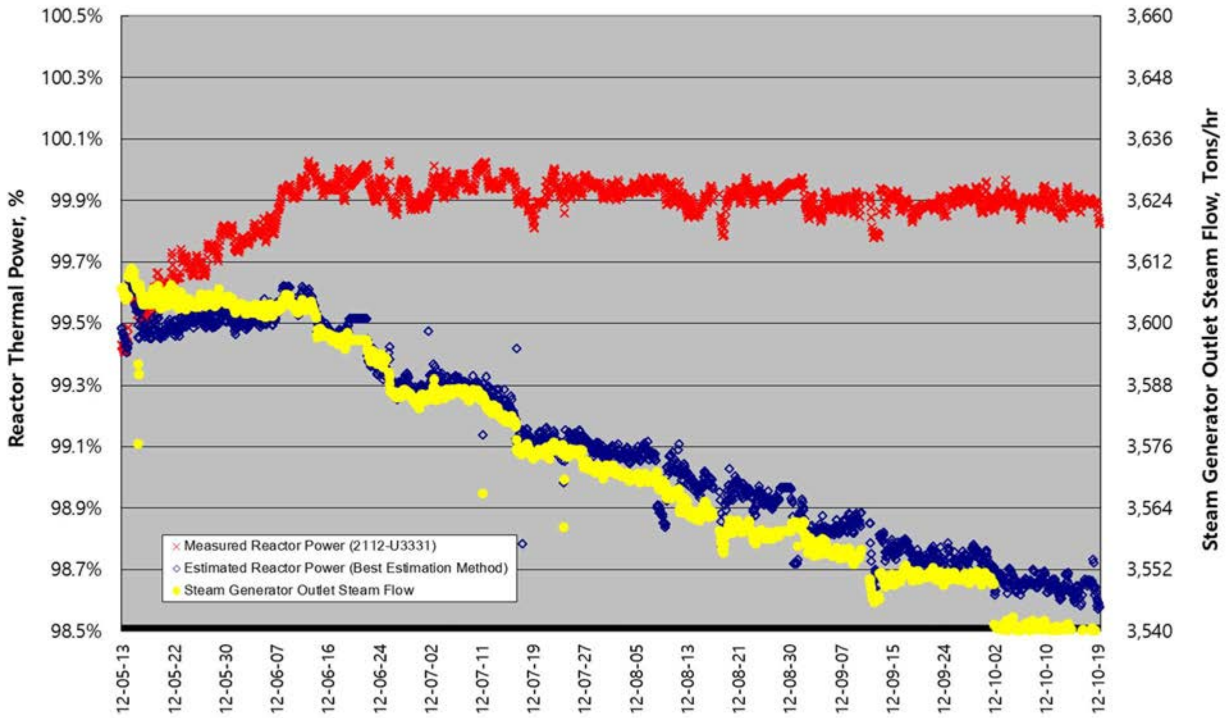


FIG. 64. Trending of RPT vs main steam flow

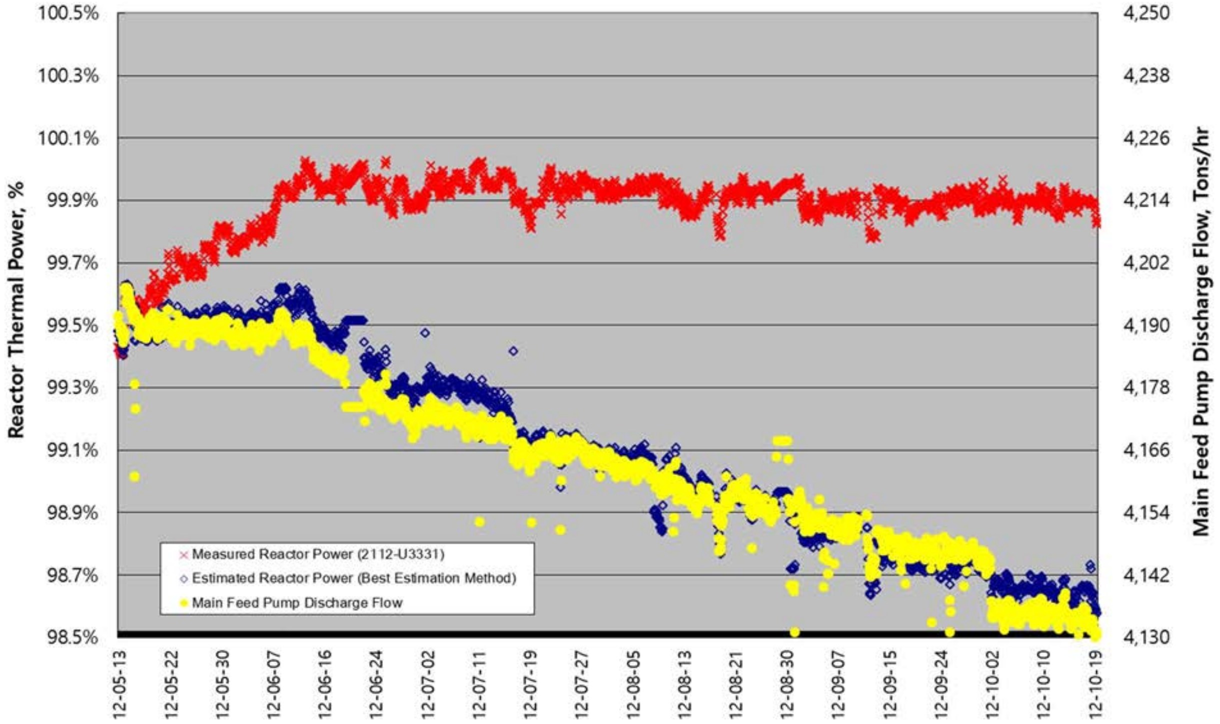


FIG. 65. Trending of RTP vs main feed pump discharge flow

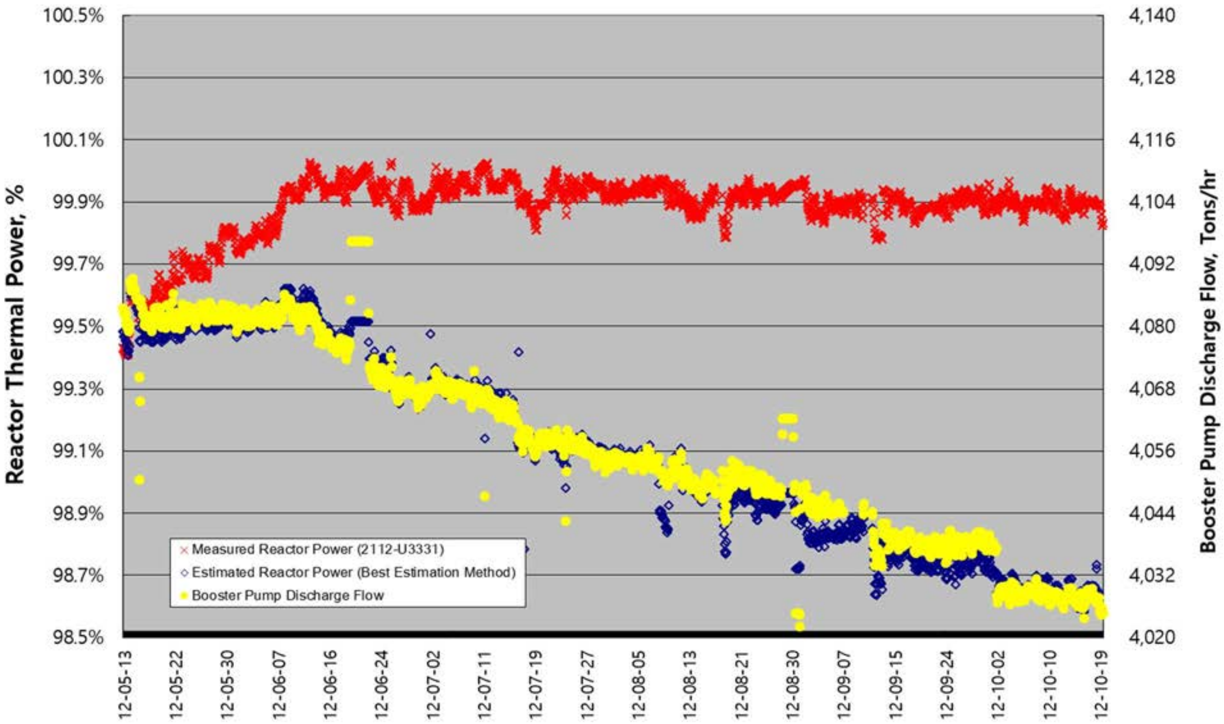


FIG. 66. Trending of booster feed pump discharge flow

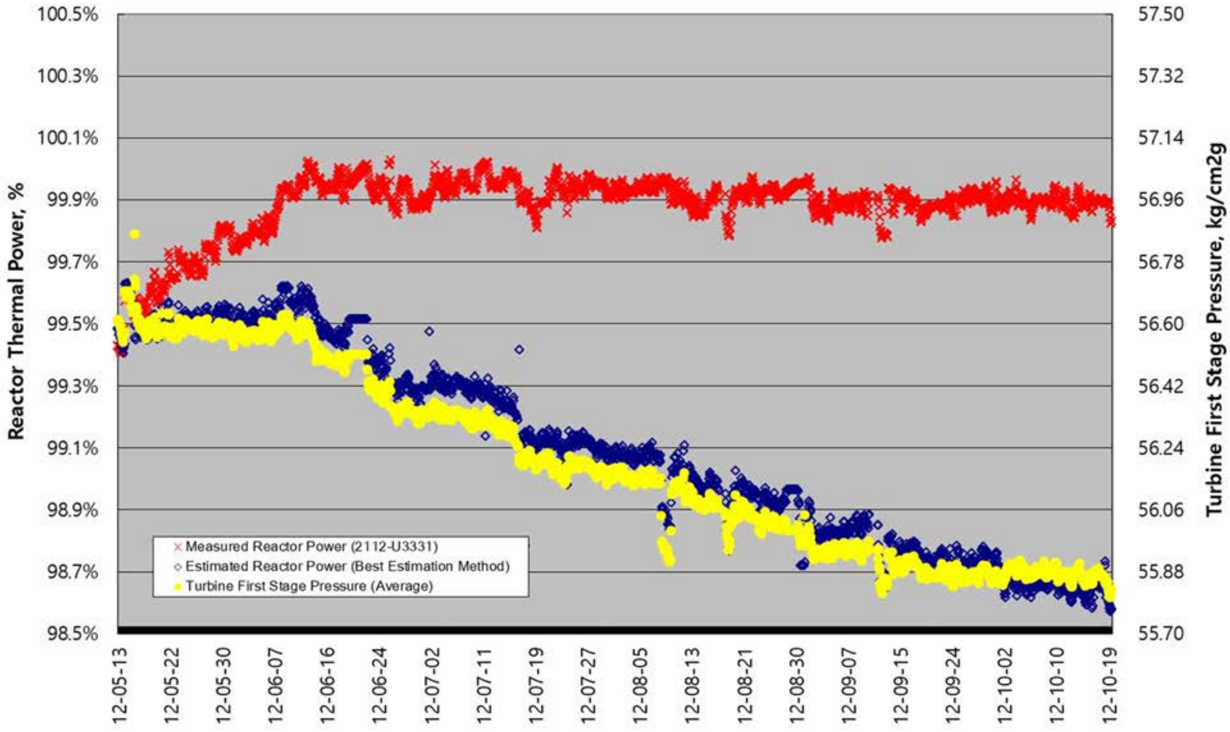


FIG. 67. Trending of RTP vs HP turbine 1st stage pressure

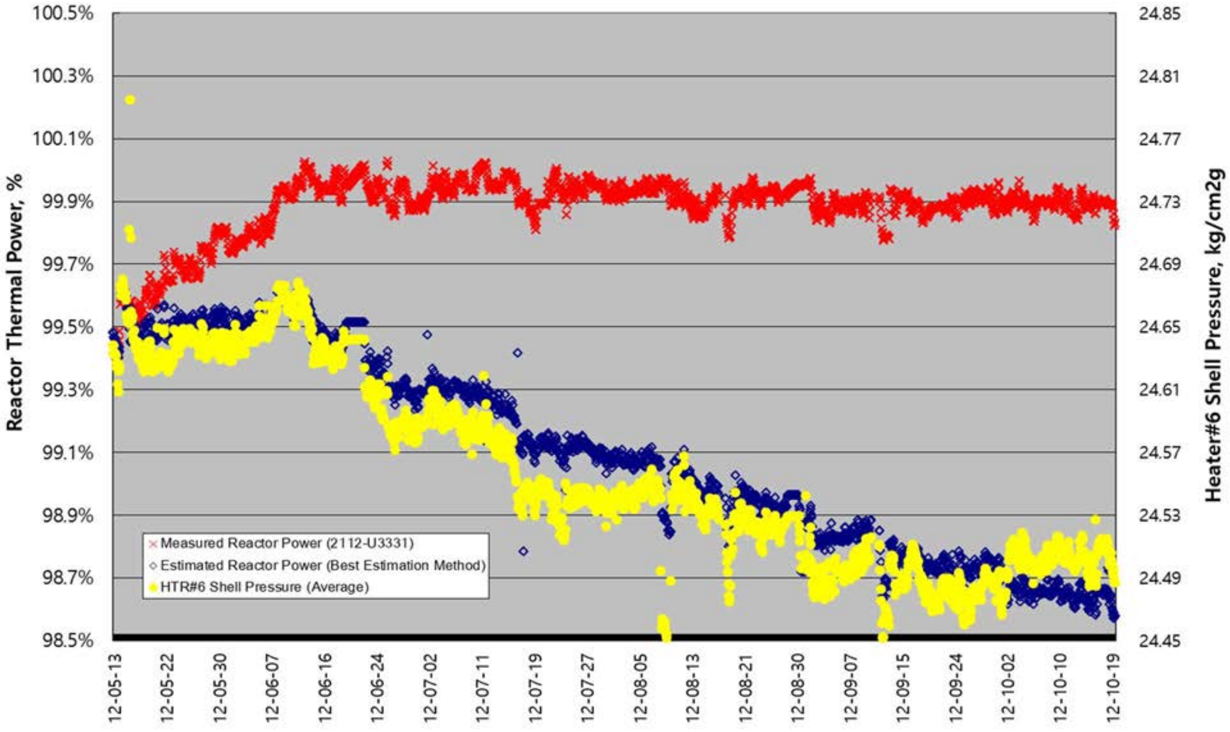


FIG. 68. Trending of RTP vs feedwater heater 6 shell pressure

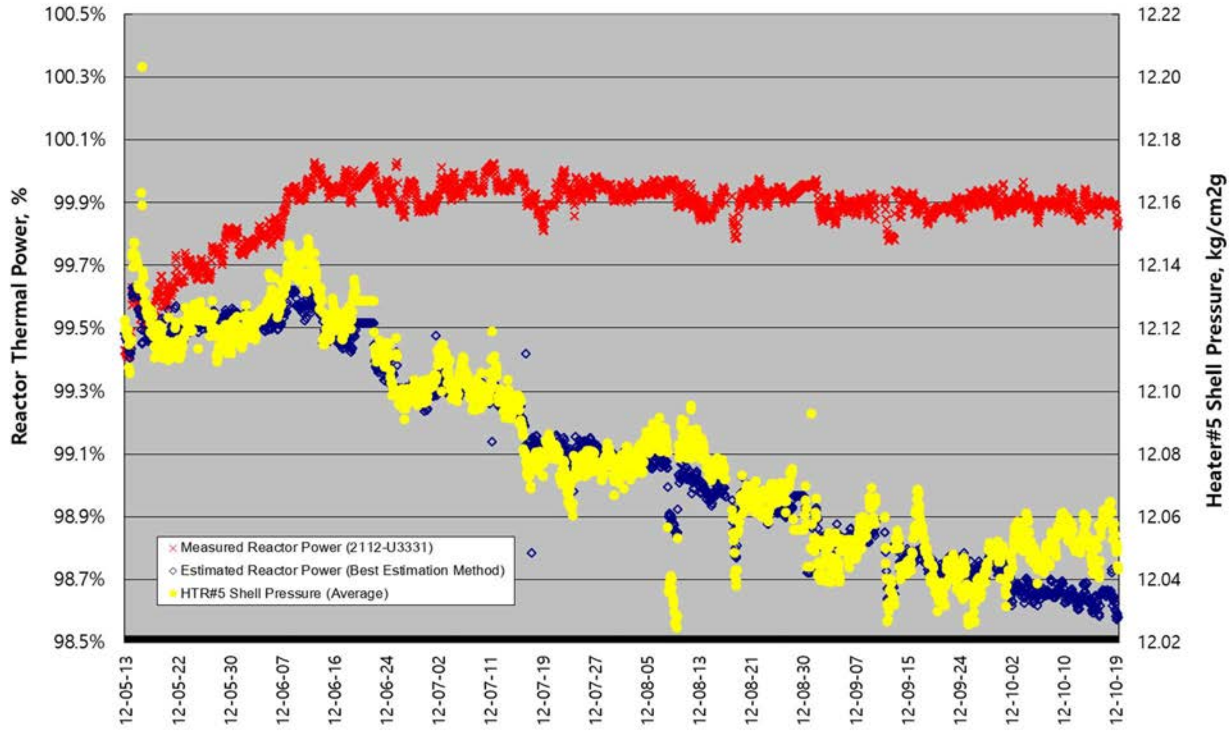


FIG. 69. Trending of RTP vs feedwater heater 5 shell pressure

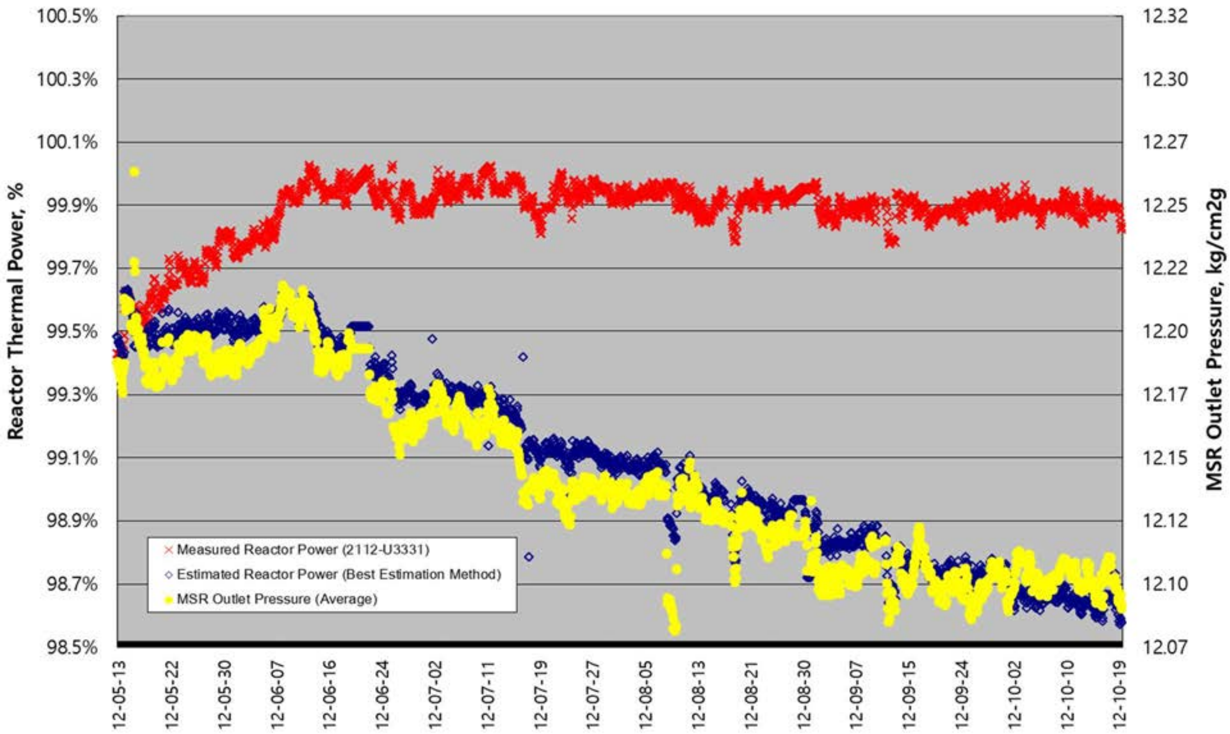


FIG. 70. Trending of RTP vs MSR outlet steam pressure

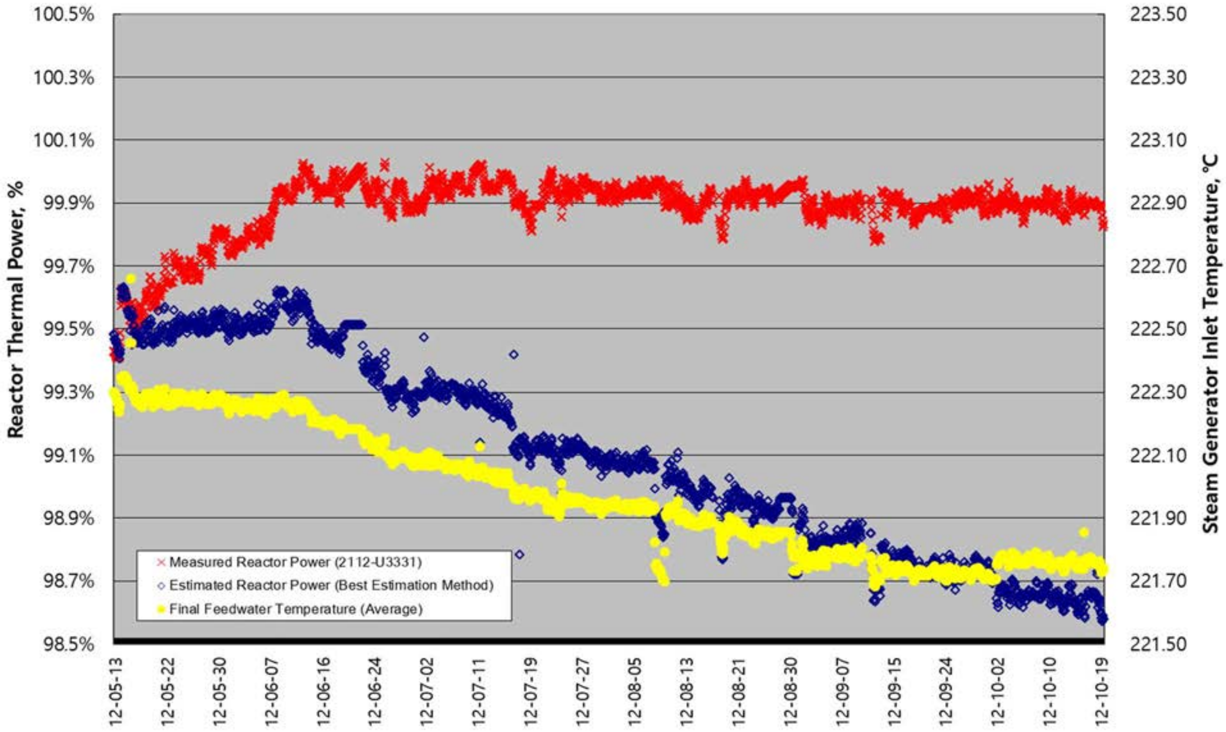


FIG. 71. Trending of RTP vs final feedwater temperature

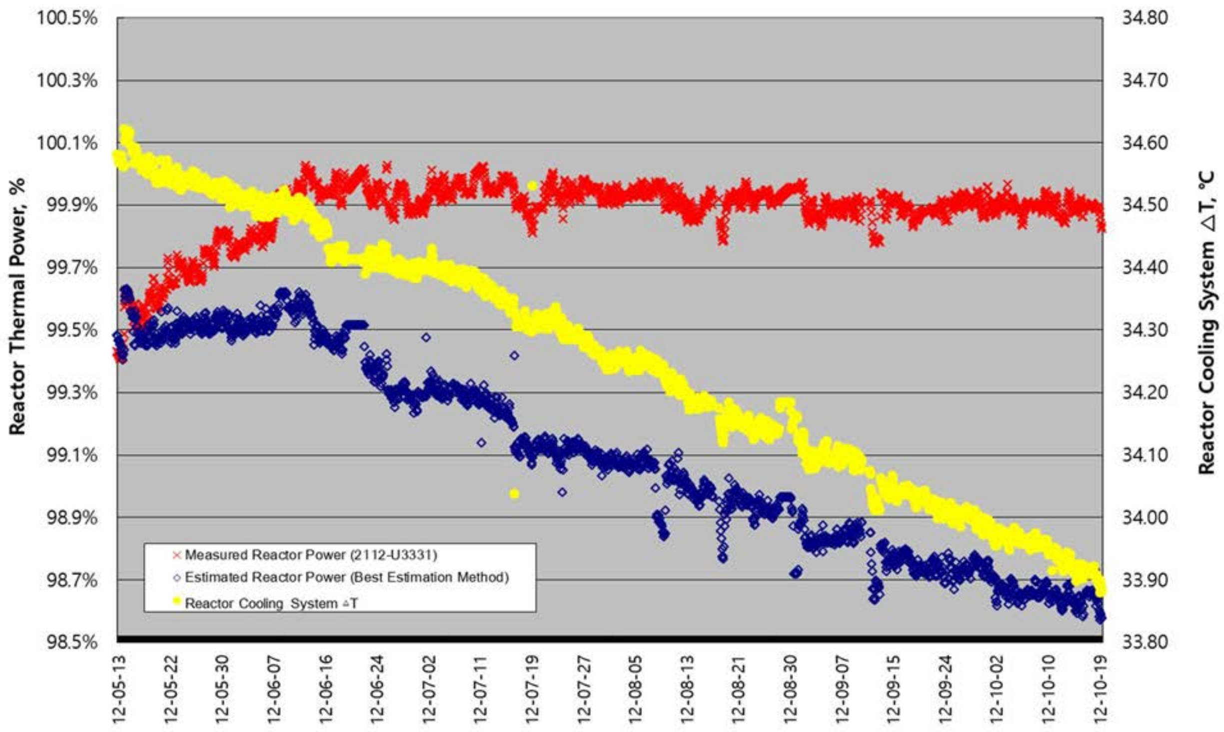


FIG. 72. Trending of RTP vs reactor cooling system ΔT

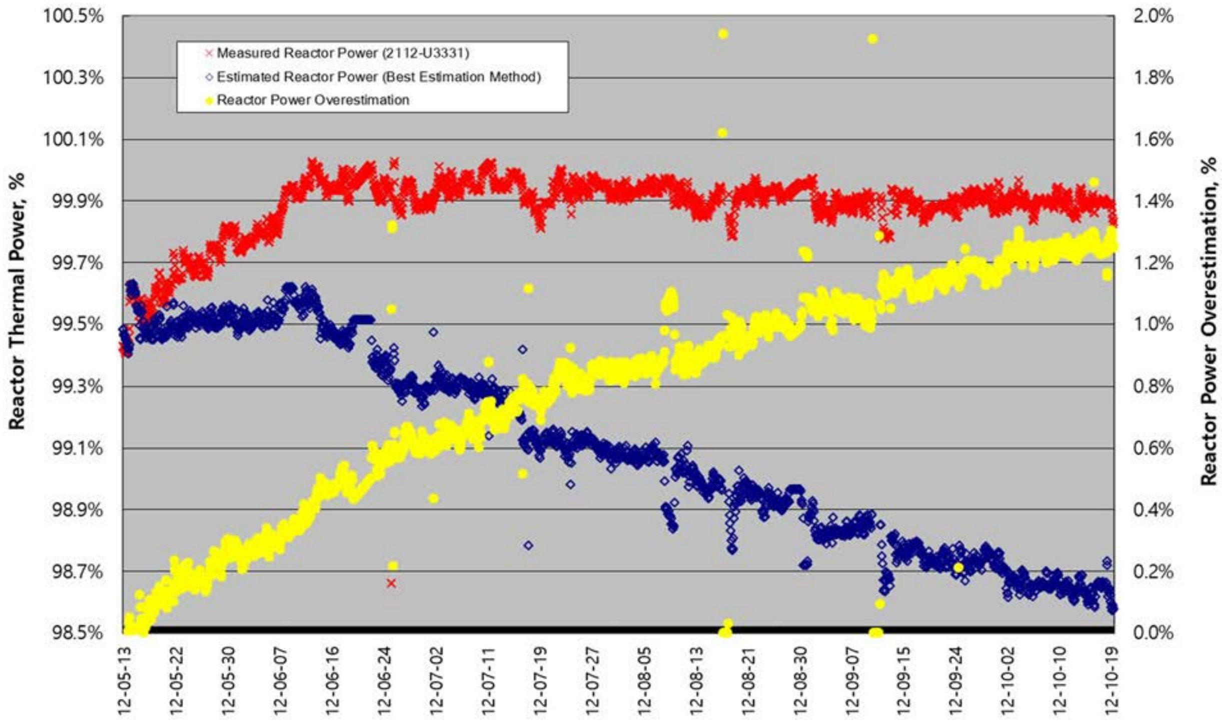


FIG. 73. Trending of the RTP measurement drift

4.5. STEAM GENERATOR THERMAL PERFORMANCE (PWR ONLY)

4.5.1. Overall heat transfer coefficient

The overall heat transfer coefficient can be calculated to monitor SG performance. This is the primary performance monitoring calculation and can be used to assess the condition of the SG tubes.

The thermal resistance of SG tubes is not consistent throughout the tube bundle. Different areas of the tube bundle will be dependent on local coolant conditions and local deposit accumulation. Deposits can affect the heat transfer coefficient, the area and sites for nucleate boiling; all of which will change the overall heat transfer capability. A mean overall thermal resistance can account for local variation and be used to evaluate and trend the performance of SGs. The overall heat transfer coefficient is the inverse of the overall thermal resistance and can be calculated by Eqs (45-47):

$$U = \frac{Q}{A \cdot \Delta T_m} \tag{45}$$

$$Q = m_{fw} \cdot (h_{ms} - h_{fw}) + m_{bd} \cdot (h_{bd} - h_{fw}) \tag{46}$$

$$\Delta T_m = \frac{T_{hot} - T_{cold}}{ln\left[\frac{T_{hot} - T_{sat}}{T_{cold} - T_{sat}}\right]} \tag{47}$$

Where U is the overall heat transfer coefficient, A is the active heat transfer area of the SG (m^2), m_{fw} is the feedwater flow rate into SG (kg/h), h_{ms} is the SG outlet steam enthalpy (kJ/kg), h_{fw} is the feedwater enthalpy (kJ/kg), m_{bd} is the SG blowdown rate at the time data is taken (kg/h), h_{bd} is the blowdown enthalpy (kJ/kg), DT_m is the log mean temperature difference (°C), T_{hot} is the primary coolant hot leg (SG inlet) temperature (°C), T_{cold} is the primary coolant cold leg (SG outlet) temperature (°C) and T_{sat} is the saturation temperature at the SG outlet pressure (°C). After the overall heat transfer coefficient has been calculated, the overall film resistance, R_f, can be calculated from Eq. (48):

$$R_f = \frac{1}{U} - \frac{1}{U_O} \tag{48}$$

where U_o is the design heat transfer coefficient of the SG.

The values of Ps, R_f, and U can be used for trending because they are measures of the SG heat transfer degradation. Another important indicator is the quantity U·A because it includes the impact of plugged SG tubes.

The calculation accuracies are affected by uncertainties in the measurement of moisture in the steam and feedwater flow. An approach for trending SG heat transfer performance has been used that reduces the effect of measurement uncertainties.

4.5.2. Other performance parameters

In addition to the overall heat transfer calculation described above, the SG performance parameters are the following:

— Steam generator outlet pressure;
— Steam generator outlet temperature (once-through SG);
— Steam generator outlet enthalpy;
— Steam generator inlet temperature (final feed temperature);
— Steam flow;
— Hot leg temperature (PWR);
— Cold leg temperature (PWR);
— Blowdown flow (PWR);
— Control rod drive flow (BWR);
— Primary side flow (PWR).

Steam generator outlet pressure is usually measured in the outlet piping, several feet downstream of the outlet nozzle. Usually in a main steam valve room or reactor building where the pressure connections and transmitters are accessible outside of the containment building or reactor shield walls.

Steam flow can be determined directly by measurement of the differential pressure across the SG or reactor vessel outlet nozzle. These nozzles generally have orifices in them which restrict flow in case of a tube leak. Although these nozzles are not calibrated flow meters, the pressure drop across these nozzles can be used to calculate a reliable steam flow for monitoring and trending purposes. Often the steam flow measurements are not density compensated and therefore, especially in the case of a PWR, as pressure changes the flow indication will be affected. In addition to the direct measurement of steam flow, it can be determined from feedwater less any blowdown plus control rod drive flow for a BWR unit. Measurement of liquid flows are generally more accurate than steam flow measurements because they measure a non-compressible flow and do not required correction for expansion. Often, plants will calibrate their steam flow measurements to the feedwater flow measurements.

Feedwater flow is usually measured with high accuracy calibrated flow nozzles or ultrasonic flow meters. Feedwater flow nozzles have a history of fouling due to deposits from the feedwater and therefore have resulted in losses due to the measured flow being higher than the actual flow resulting over conservative core thermal power calculations. The steam flow nozzles are typically unaffected by these deposits. So, a comparison of steam flow indicated by the steam flow nozzles and the steam flow calculated by the feedwater flow nozzles can be used to detect and trend feedwater nozzle fouling. Ultrasonic meters also are not affected by deposits and are commonly used to create correction factors to feedwater nozzle indicated flows or are used for direct feedwater flow measurement.

For a SG or reactor which produces saturated steam, the outlet enthalpy is determined from a moisture carryover test or design data provided by the manufacturer. Typical values of moisture carryover are between 0 ~ 0.25%, resulting in a thermodynamic quality of between 0.9975 and 1.0. The enthalpy leaving the SG or reactor can be calculated using the quality and the measured pressure. The moisture carryover determined from testing illustrates the moisture leaving the SG or reactor separator section before the outlet nozzle. When calculating the enthalpy from steam tables, the internal shell pressure at the separator outlet is required. Many SG or reactor outlet pressures are measured after the outlet nozzles or after several elbows and many feet of pipe. Therefore, a design pressure drop calculation may be necessary to back out the pressure at the separator outlet.

For SGs which produce superheated steam, the enthalpy can be determined directly using the measured pressure and temperature. Blowdown flow needs to be measured downstream from the SG where there is sufficient static head to prevent flashing across the flow meter or after it passes through a blowdown cooler. Mass flow to the Control Rod System for a BWR is usually measured using an orifice meter.

Primary coolant flows are sometimes measured using elbow taps. This method is good for trending but is not as accurate as a calibrated flow meter. When higher accuracy is required, the primary side flow metering method may be calibrated by comparison to secondary side calculations of reactor power, or ultrasonic flow meters may be installed on the primary coolant piping.

T_{hot} and T_{cold} are measured using installed plant instrumentation. The T_{hot} measurements are subject to hot leg streaming which can cause a differential between the actual bulk hot leg temperature and that indicated by the temperature instruments. Feedwater temperature used in SG performance calculations is anticipated to be the most accurate available. These usually are the same ones used for the reactor calorimetric calculations.

4.6. UNUSUAL UNIT OPERATIONS

4.6.1. Reduced reactor thermal power operation (CANDU)

Pressurized Heavy Water Reactors, also known as CANDU power reactors feature at-power fuelling and a relatively large reactor core. These factors result in near continuous burn-up distribution change with potential thermal flux and power oscillations due to xenon transients. Thus, CANDU reactors are designed with several types of spatially distributed devices permitting reactivity control in three dimensions.

At the heart of CANDU safety philosophy, a defence-in-depth approach limits the occurrence and consequences that could lead to potentially unsafe conditions. Three separate reactor systems independently act to prevent overpowers in the reactor fuel. They are the Reactor Regulating System and two fully independent shutdown systems (SDS1, SDS2). The Reactor Regulating System controls both the distribution and overall level of neutron flux and power within the core to stabilize spatial control and bulk reactivity control. Each of SDS1 and SDS2 is driven by several process and neutronic trips. If an overpower arises in the core due to a Reactor Regulating System failure, each shutdown system is designed to independently detect the condition and initiate a shutdown. To assure their independence, the two shutdown systems are diverse in design and principle of operation, spatially separate and functionally equivalent.

Overpowers in the reactor fuel might occur due to a localized peak while the reactor remains at normal power, or due to an uncontrolled power excursion as a result of Loss-of-Reactivity-Control (LORC). By design, potential failures of the regulating programmes result in a fail-safe condition to automatically shut down the reactor. Despite the inherently fail-safe design, CANDU reactors are provided with a diverse variety of appropriate trip parameters which will initiate a reactor trip in case of LORC. This include on high log rate of increase of neutron power (fast LORCs), high pressure in the primary heat transport system (intermediate rate LORC), and the high neutron power or regional overpower trip. The regional overpower trip is the primary trip for slow LORCs, but also acts as a backup trip for the more rapid transients.

Because of the flux detector response characteristics and the lack of transport delays in transferring thermal power to the coolant, the most restrictive LORC is a slow, gradual increase in reactor power. Thus, the 'slow' LORC serves as a design basis of the Regional Overpower Protection (ROP) systems. If an overpower condition in the reactor were identified by either ROP system, trip sequences SDS1 and SDS2 would initiate a reactor shutdown.

A ROP set point from which overpower is defined needs to be annually decreased due to aged deterioration in the reactors. The deterioration is categorized into two phenomena. The first kind of deterioration phenomenon changes flow distribution in a fuel channel as pressure tubes are deflected due to irradiation of neutron or radial/axial extension in the tube. The second kind of deterioration phenomenon reduces flow rate in the primary side heat transfer system and consequently raises temperature of coolant at the reactor inlet. The decrease in flow rate is caused by magnetite removed from feeder tubes by flow accelerated corrosion that is deposited on the surface of SG inlet or the cold leg of the reactors. Both deterioration phenomena result in fuel channel flow rate changes, making it possible to reach the critical channel power, which results from the dry out of the fuel sheath.

Unfortunately, fuel sheath damage is likely from dry out in CANDU reactors as the reactors operate. Thus, the set-point is required to be lower in order to have ROP successfully trip the reactor before the fuel bundles are damaged. As a result, it is unavoidable to reduce the reactor power annually in the CANDU power reactors. Although the ROP trip was initially developed with relatively simple Neutron Overpower Trips on earlier CANDU reactors it has been revised to comprehensive designs on contemporary reactors. The latest designs are praised for optimized techniques applied to the problem of providing protection for a wide variety of possible flux shapes, while minimizing restrictions on reactor operating margins.

4.6.2. Operation at reduced (primary) temperature (PWR)

Operation at reduced temperature (ORT) literally means that the reactor (Rx) is operated while the temperature of primary coolant is decreased in hot legs and cold legs. The ORT is one of the methods which can mitigate SG degradation rate due to primary water stress corrosion cracking (PWSCC). Damages has been reported in Ni-based alloys particularly in Alloy 600MA (Mill Annealed), 600TT (Thermal Treated) and its parent weld metals 82, 132, and 182. Despite a large number of research efforts, the physical and chemical mechanisms of PWSCC are not completely proven yet. However, the contributions to PWSCC occurrence include metallurgical condition, non-uniform cold work, and residual stress in welding [1]. PWSCC is a thermally activated mechanism that can be correlated with an Arrhenius relationship and is quite temperature dependent. Thus, the PWSCC can be postponed by ORT. On the other hand, no crack has been detected in Alloy 690TT and 800 which have been used for manufacturing SG tube bundles from the mid-1980s [2].

The reactor temperature in the ORT needs to be determined regarding the plant economics so that the number of plant modifications and loss of electric power generation can be minimized. In case of Hanbit #3, 4 the reactor temperature of the ORT was obtained as shown in Figure 74. A range of cold leg temperature from 90% to 100% reactor power was changed from 294.4 °C (562 °F) to 288.8 °C (552 °F), which was identical to cold leg temperature limit under 90% reactor power. It was found through sensitivity analysis that this temperature range not only made hot leg temperature employed without modifications in major component equipment but also secure 14.2 °C (6.5 °F) cold leg temperature as an operation margin in the initial ORT condition.

Table 12 shows the designed operation variables of secondary side in SG determined for Hanbit #3, 4 ORT. The hot leg temperature was decreased to 321.6 °C (611 °F) from 327.3 °C (621.2°F) that is initially designed in construction of NPPs.

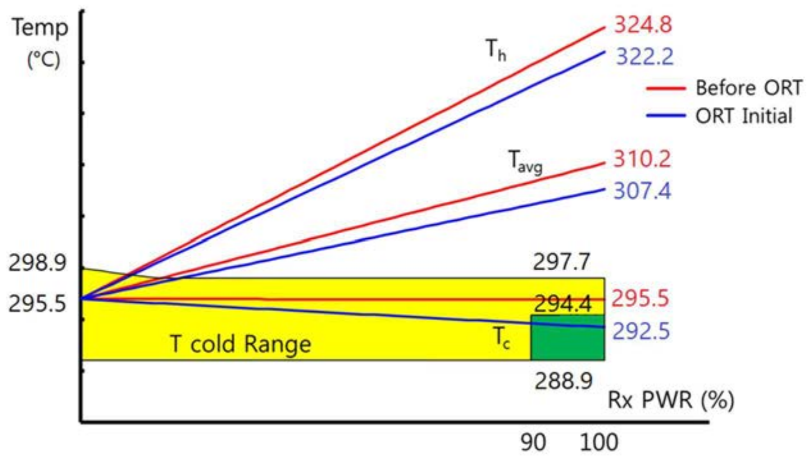


FIG. 74. Reactor temperature map for Hanbit #3, 4 ORT

TABLE 12. MAIN OPERATION VARIABLES IN SECONDARY CYCLE FOR HANBIT #3, 4 ORT

Variables	Design operating condition	ORT operating condition
SG outlet pressure @ 100% reactor (Rx) PWR (MPa / psia)	7.37 / 1070	6.798 / 986
Final feedwater temperature @ 100% reactor (Rx) PWR (°C / °F)	232.2 / 450	232.2 / 450
Main steam mass flowrate @ 100% reactor (Rx) PWR (10^6 kg/h / 10^6 lb/h)	5.76 / 12.72	5.74 / 12.66

TABLE 12. MAIN OPERATION VARIABLES IN SECONDARY CYCLE FOR HANBIT #3, 4 ORT (cont.)

Variables	Design operating condition	ORT operating condition
Feedwater mass flowrate @ 100% Reactor (Rx) PWR, 0.2% B/D (10^6 kg/h / 10^6 lb/h)	5.78 / 12.75	5.75 / 12.69
SG outlet pressure @ 0% reactor (Rx) PWR (MPa / psia)	8.06 / 1170	8.06 / 1170

SG outlet pressure needs to decrease for cold leg temperature reductions at constant reactor power. Due to main steam pressure drop, the volume flowrate dramatically increases in accordance with small changes in enthalpy of saturated steam under the condition that the final feedwater enthalpy does not change. On the other hand, the mass flowrate of the main steam can be considered constant.

It needs to be also noted that the reactor temperature for the ORT needs to be determined considering the current actual volume flow margin of HP turbine accompanied by investigation on the probability of capacity acceptance in BOP facilities. The economic feasibility evaluation for the ORT ought to consider the SG life cycle, power loss of the turbine cycle, and possibility to accept flow capacity in the HP turbine. Firstly, the SG life cycle is estimated in accordance with operation and maintenance strategies, where the ORT can constitute an integrated countermeasure for an extended SG life cycle with other technologies to suppress cracking. Other suppressing techniques include chemical cleaning, improved water chemistry, enhancing SG management programme, and SG tube plugging. Secondly, the loss of turbine cycle power is calculated for economic feasibility evaluation in accordance with the change of pressure and the steam quality at the SG outlet, which are obtained through thermal-hydraulic analysis of the SG. Thirdly, the economic feasibility is reviewed by investigating the range of component equipment required to be modified (or retrofitted) in order to recover the loss of turbine cycle output. It is advised that the analysis includes the change in volume flowrate in front of the turbine inlet control valves due to the ORT and flow margin with respect to valves wide open condition in turbine design.

After the previously mentioned evaluation of economic feasibility the decision of reactor temperature range for the ORT starts with a prediction of a trend of damage in the SG tube bundle. This is considering the selected strategy for SG life cycle extension, in-service inspection in SG tubes, and experiences from NPPs of a similar type. Then an amount of and cost for the required maintenance in SGs can be predicted in accordance with the maintenance strategy for the damaged SG tubes. Finally, plant maintenance authorities are advised to choose the optimized solution from three alternatives by comparing the expenses for them; loss of power generation, BOP modifications due to the ORT, and SG replacement. The expenditure for SG replacement is estimated based on the tube plugging rate predicted from the amount of required maintenance in the SG tubes.

The evaluation of thermal performance in the turbine cycle is conducted by a heat balance for as-found operating conditions and affects the decision of the reactor temperature range for the ORT. If generator output has to be maintained after the ORT, plant modifications have to be considered. The essential field tests are moisture carryover and performance tests in the turbine cycle before and following the ORT. The thermal heat balance of the turbine cycle in an as-found condition helps to estimate turbine power in normal operating and design valve wide open (VWO) conditions. Unless the ORT condition exceeds the VWO condition, the rated turbine power could be maintained and modifications become unnecessary for BOP facilities. However, it needs to be pointed out that the ORT conditions are not supposed to be compared with design VWO conditions but with actual VWO condition resulted from the manufacture of HP turbine. Meanwhile the turbine cycle power will decrease if the ORT condition exceeds the VWO volume flow condition. If it is planned to modify or retrofit HP turbine so that the power loss of turbine cycle is recovered, it results in a new VWO condition. And if the new VWO condition goes further the initial design VWO condition structural integrity needs to be evaluated for a turbine-generator system for the increased flowrate.

The next factor to be considered is the time period for the ORT if the HP is scheduled to be replaced or modified. The longer life cycle of the NPP than the ORT schedule possibly recommends a SG replacement to increase SG outlet pressure. In conclusion, when the reactor temperature range for the ORT is determined, precise and integrated engineering judgment and economic feasibility need to be obtained by investigating the turbine-generator system as well as the nuclear steam supply systems and BOP systems.

4.7. SUMMARY

The importance of improving the reliability of the RTP measurement cannot be emphasized enough for both plant safety and performance optimization. However, many units suffer from the RTP measurement drift which sometimes causes significant losses in the electrical power output. The plant may need to invest several millions of USD to improve the electrical power output (of thermal efficiency) by 1% through repair or replacement of major components. Still is not unusual to lose 1% of the electrical power output due to overestimation of the RTP.

This section introduces four independent methodologies presented in EPRI reports and used for monitoring and adjusting the RTP measurement drift. A case study that shows an example analysis of the RTP measurement drift in a 680 MW rating PWR unit in Korea is introduced in Section 4.4.5. In case of this unit, the HP feedwater heater tube bundles were replaced during a planned outage. After the unit restart, a RTP overestimation occurred with strong suspicion of venturi fouling. The Best Estimation method was used to identify and trend the RTP measurement drift.

5. KEY COMPONENTS PERFORMANCE – TURBINE CYCLE

Fossil steam turbines are dominantly operated at the superheated steam region, except for the latter two or three stages of the LP turbine. Most of the nuclear steam turbines are operated in the wet steam region, except for the first or up to the second stages of the LP turbine. This operating characteristic makes it very difficult to evaluate the test cycle heat balance for the nuclear turbine cycle. This is because the enthalpy of the cycle steam and the extraction steam cannot be determined without measuring the moisture content. As an alternative, engineering assumptions for the HP turbine exhaust enthalpy and the moisture removal effectiveness of the LP turbine wet stages can be used for performance monitoring purposes.

Additionally, while the fossil plants routinely cycle load to accommodate changing power demand, most NPPs operate at constant RTP, supplying base load power to the grid. Consequently, although evaluation of the turbine cycle heat balance can be challenging, performance monitoring of the nuclear turbine cycles and their key components is relatively easy compared to turbine cycle performance monitoring at fossil plants. This is because, at least theoretically, performance parameters of turbine cycle components remain constant while the plants operate at 100% RTP.

Section 5 introduces definitions and governing equations of performance indicators for the overall turbine cycle and their key components. The performance indicators are supposed to be continuously monitored during normal operation of the plant.

5.1. ENERGY EFFICIENCY

The energy efficiency of an NPP is an essential indicator for the assessment of the efficient conversion of nuclear fuel energy into electricity.

5.1.1. Thermal efficiency

"The definition of thermal efficiency considers only the actual cycle process used in the power plant. The efficiency is then the ratio of the useful mechanical output to the heat flow transferred to the cycle process media." [27]

"In this context, the useful mechanical output is the mechanical output from the turbine. Strictly speaking when the feed pump is driven by a turbine which is operated with extraction steam from the main turbine. As the condensate pump also contributes to raising the pressure, it is considered to be part of the feed pump (from a thermodynamic point of view). Its mechanical output has therefore to be subtracted from the mechanical output of the turbine. The mechanical output of the turbine, in a thermodynamic sense, is in this case the output resulting from the steam mass flow and the enthalpy difference." [27]

"If the feed pump is driven by an electric motor, then the useful mechanical output is equal to the difference between the mechanical output of the turbine minus the drive outputs of the feed pump plus the condensate pump." [27]

The heat flow transferred to the process is the heat flow transferred to the water/steam cycle.

5.1.2. Electrical efficiency, system boundaries for efficiency definition

For the definition of electrical efficiency, it is important that the system boundaries are defined carefully. A simplified scheme is shown in Figure 75 below.

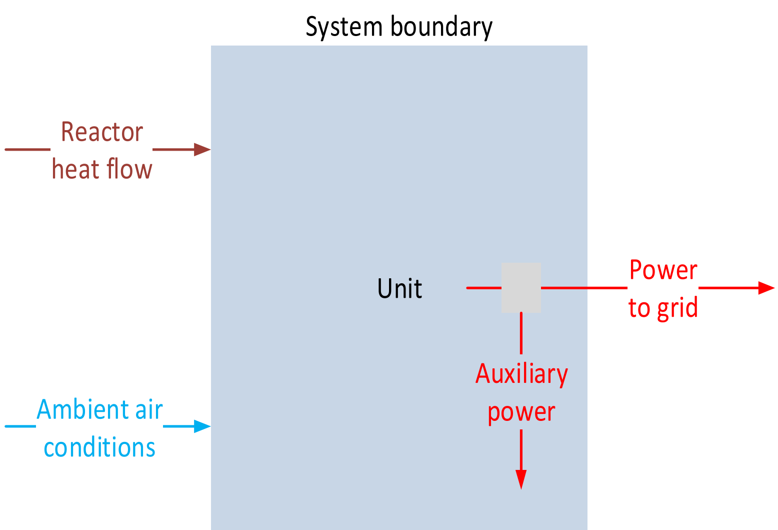


FIG. 75. System boundaries for efficiency definitions

All components within the system boundaries can impact the auxiliary power and therefore impact the net values.

“The electrical efficiency is defined as the ratio of delivered or generated effective electrical output of a power plant to the supplied heat (fuel) input. In the power plant, the output is delivered in the form of electricity (electricity generation only).” [27]

5.2. TURBINE CYCLE CARNOTIZATION

The objective of turbine cycle carnotization is to apply thermodynamic principles to improve the Rankine cycle. This is done by using concepts of reheating and regeneration via additional components such as feedwater heaters or reheater. These means develop a cycle closer to the Carnot cycle i.e. to introduce carnotization.

The common base is:

- Increasing the mean temperature of heat addition in cycle.
- Degreasing the mean temperature of heat rejection in cycle.

Useful tools for quantitative understanding of the Rankine cycle process (and sub-processes) are T-s, h-s, and other diagrams.

5.2.1. Carnot cycle, Carnot principle, Carnot efficiency

The Carnot cycle is a theoretical thermodynamic cycle which sets the upper limit on the efficiency of any thermodynamic engine (as shown in Figure 76). It can be used as a theoretical comparison model, and as a good tool for learning how different operating parameters influence the performance of thermodynamic cycles.

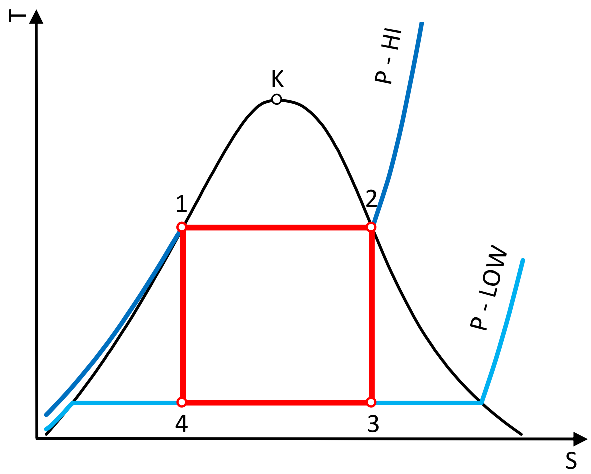


FIG. 76. Carnot cycle

The Carnot cycle consist of:

- Isentropic compression (Pump) 4 – 1;
- Constant temperature heat addition (Boiler, SG) 1 – 2;
- Isentropic expansion (Turbine) 2 - 3;
- Constant temperature heat rejection (Condenser) 3 – 4.

The Carnot principle states that the ratio of the work and heat received by an engine which operates according to a Carnot cycle depends only on the temperature of hot and cold reservoirs. The Carnot thermal efficiency (ideal efficiency) is a measure of the quality of the conversion of heat into work between two temperature levels.

The Carnot thermal efficiency can be expressed as in Eq. (49):

$$\eta_{Carnot\ thermal} = \frac{W}{Q_H} = 1 - \frac{T_C}{T_H} \tag{49}$$

Where W is the work done by the engine, Q_H is the heat put into engine, T_C is the absolute temperature of the cold reservoir and T_H is the absolute temperature of the hot reservoir.

The definition above does not cover a pump work. The required pump work can be expressed by back work ratio as in Eq. (50):

$$bwr = \frac{W_{Pump}}{W_{Turbine}} \tag{50}$$

Where bwr is a back work ratio, W_{Pump} is the pump work input (– work) and $W_{Turbine}$ is the work produced by turbine (+ work).

5.2.2. Ideal Rankine cycle, real Rankine cycle

The problem is that the Carnot cycle is not practical for steam power cycles. The Rankine cycle is a practical steam power cycle that is most similar to a Carnot cycle.

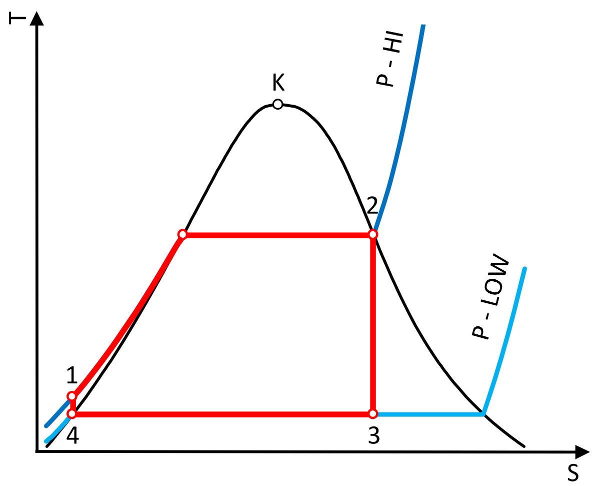


FIG. 77. Ideal Rankine cycle

In the real world there are deviations between a real cycle and an ideal cycle. The phenomena which transform the ideal Rankine cycle (as shown in Figure 77) to the real Rankine cycle are:

- Heat losses;
- Fluid friction;
- Mechanical losses;
- Condenser subcooling.

These phenomena make the cycle irreversible and lead to the lost work in the real Rankine cycle as shown in Figure 78.

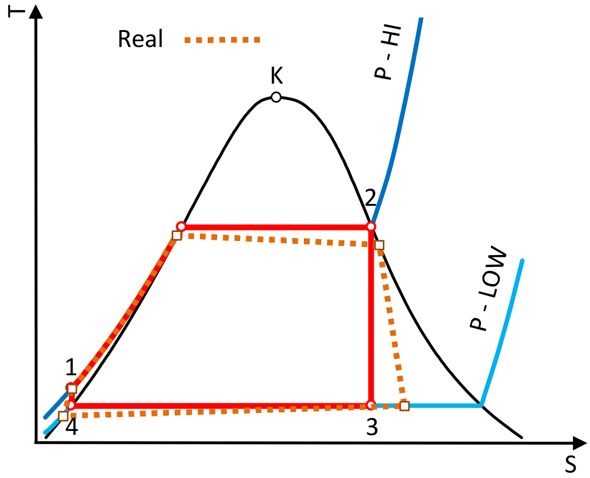


FIG. 78. Real Rankine cycle

5.2.3. Rankine cycle carnotization

Thermal efficiency can be improved by cycle modification which manipulates temperatures and/or pressures in various components of a cycle.

For example:

(a) Steam

"The most important factors in increasing efficiency are the highest possible temperature and pressure of the working medium. In NPPs, the partially expended steam is reheated by one or more reheating stages at the MSR." [27]

(b) Vacuum in the condenser

"After leaving the low-pressure section of the steam turbine, the steam is condensed in condensers and the heat released into the cooling water. In order to ensure the maximum pressure-drop over the steam turbines, it is desirable to reduce the vacuum to a minimum. In general, the vacuum is dictated by the temperature of the cooling water, which is lower with once-through cooling systems than with a cooling tower. The best electrical efficiency is possible by seawater or freshwater cooling and a condenser pressure of approximately 3.0 kPa. Air cooling usually results in significantly lower efficiency." [27]

(c) Condensate and feedwater preheating

"The condensate coming out of the condenser and the feedwater are heated by steam to just under the saturation temperature of the extracted steam. The thermal energy from the condensing process thus feeds back into the system, reducing the amount of heat otherwise released from the condenser, therefore improving the efficiency." [27]

(d) Influence of climate conditions on efficiency (cooling tower)

The climate expressed in terms of wet- and dry-bulb temperatures is an extremely important site-specific condition. A certain operational flexibility of the cooling system can then be very important.

"The cooling medium temperature depends on the dry- and wet-bulb temperatures. A wet-bulb temperature is always lower than a dry-bulb temperature. The wet-bulb temperature depends on the measured temperature of the atmosphere, the humidity, and the air pressure. For latent (evaporative) heat transfer, wet-bulb temperature is the relevant temperature. It is theoretically the lowest temperature to which water can be cooled by evaporation. For sensible heat transfer, dry-bulb (dry air) temperature is relevant, where air is the coolant." [27]

The basic idea behind all these modifications to increase the cycle efficiency is the same. Increasing **the mean temperature** at which heat is transferred to the working fluid in SG or decrease the mean temperature at which heat is rejected from the working fluid in the condenser.

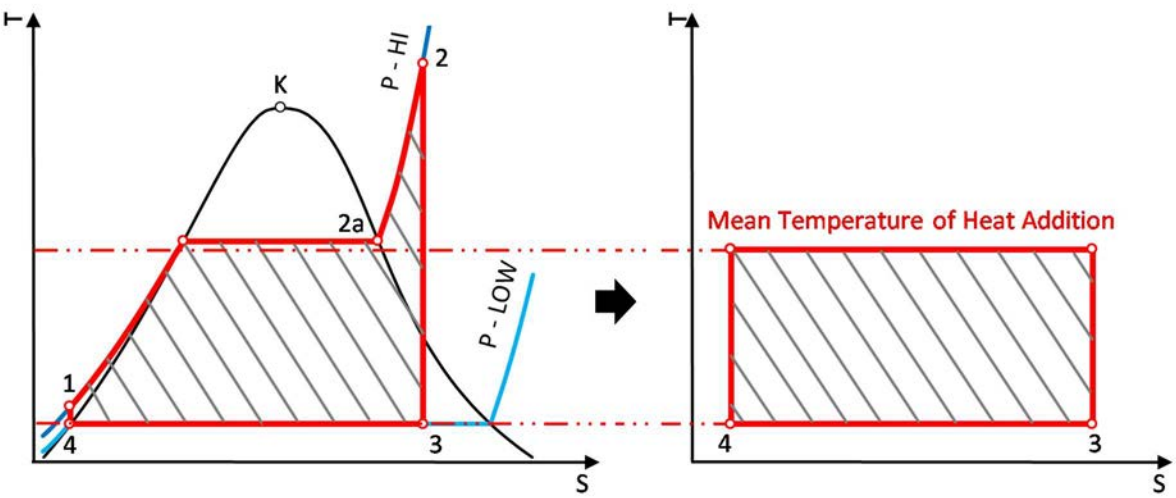


FIG. 79. Rankine cycle carnotization

Modification can be referred to:
- Simple Rankine cycle;
- Improved Rankine cycle (**carnotization**).

Modification related to **Simple Rankine cycle** are:
- Increase maximum temperature (superheat) – higher mean temperature of heat addition.
- Increase SG pressure – higher mean temperature of heat addition.
- Decrease condenser temperature (and pressure) – less heat rejected.

The Rankine cycle carnotization (shown in Figure 79) is a modification which moves the Rankine cycle closer to the Carnot cycle and is based on addition of extra equipment to the cycle. The general principle of carnotization is based on the separation of the ideal Rankine cycle on the three partial cycles as shown in Figure 80. Each partial cycle has different mean temperature of the added heat.

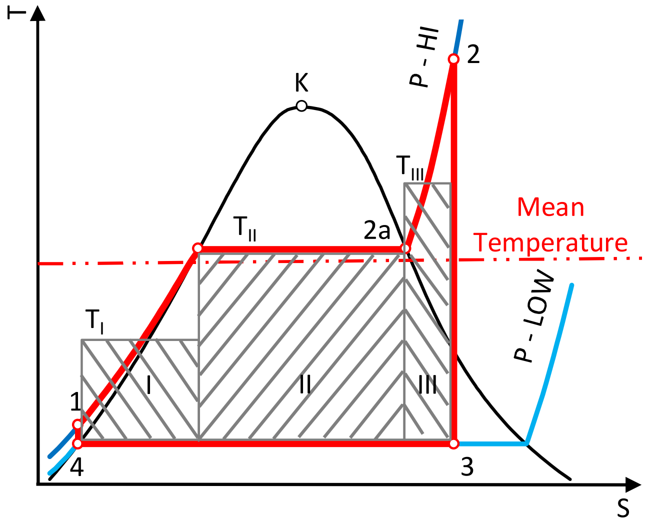


FIG. 80. Partial cycles of Rankine cycle

The main goal of carnotization is to increase the mean temperature at which heat is transferred to the working fluid in one or more of the partial cycles.

Modifications related to the **improved Rankin cycle** (carnotization) are:

(a) Reheating (shown in Figure 81)

(b) Regenerative feedwater heating (shown in Figure 82)

— A closed feedwater heater (a non-contact heater).
— An open feedwater heater (a direct contact heater).

Modifications mentioned above are limited by its advantages and disadvantages. It depends on a type and size of equipment and connection in thermal cycle.

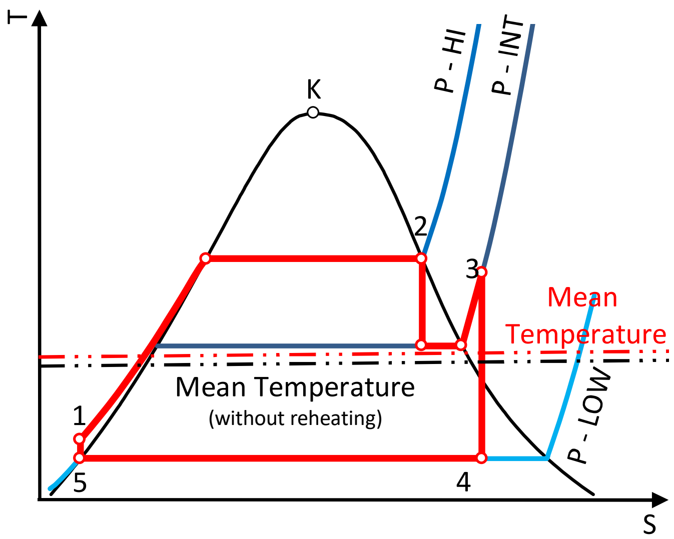


FIG. 81. Ideal reheat Rankine cycle

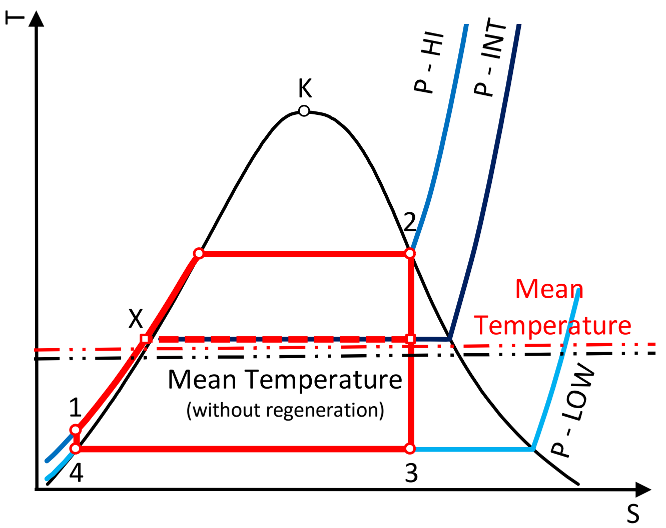


FIG. 82. Ideal regenerative Rankine cycle with closed feedwater heater

5.2.4. Realistic applicability of cycle carnotization

In general, the Carnot cycle, ideal Rankine cycle and real Rankine cycle (Figure 83) are concepts that serve for better understanding of what parameters are significant in cycle behaviour and performance.

Generally, factors affecting the **level of cycle carnotization** (optimization) are:

- Cycle and equipment design;
- Control strategy;
- Operational practices;
- Ambient conditions at site.

The phenomena affecting the level of cycle carnotization in plants under operation are:

(a) Quality of control strategy, control algorithm, and set points;

(b) Component health:

- Leakage losses and leakage in the system (cycle isolation);
- Pressure losses;
- Equipment fouling etc.

These phenomena cause process or component anomalies and recommended to be under continuous supervision and control. The advanced concept is based on comparison of actual state and expected state using physical or data based mathematical models.

The turbine cold end optimization has a specific position inside the control strategies. It maximizes the electricity delivered to the net by control of cooling water flow under given ambient conditions, see also Section 6.1.7.

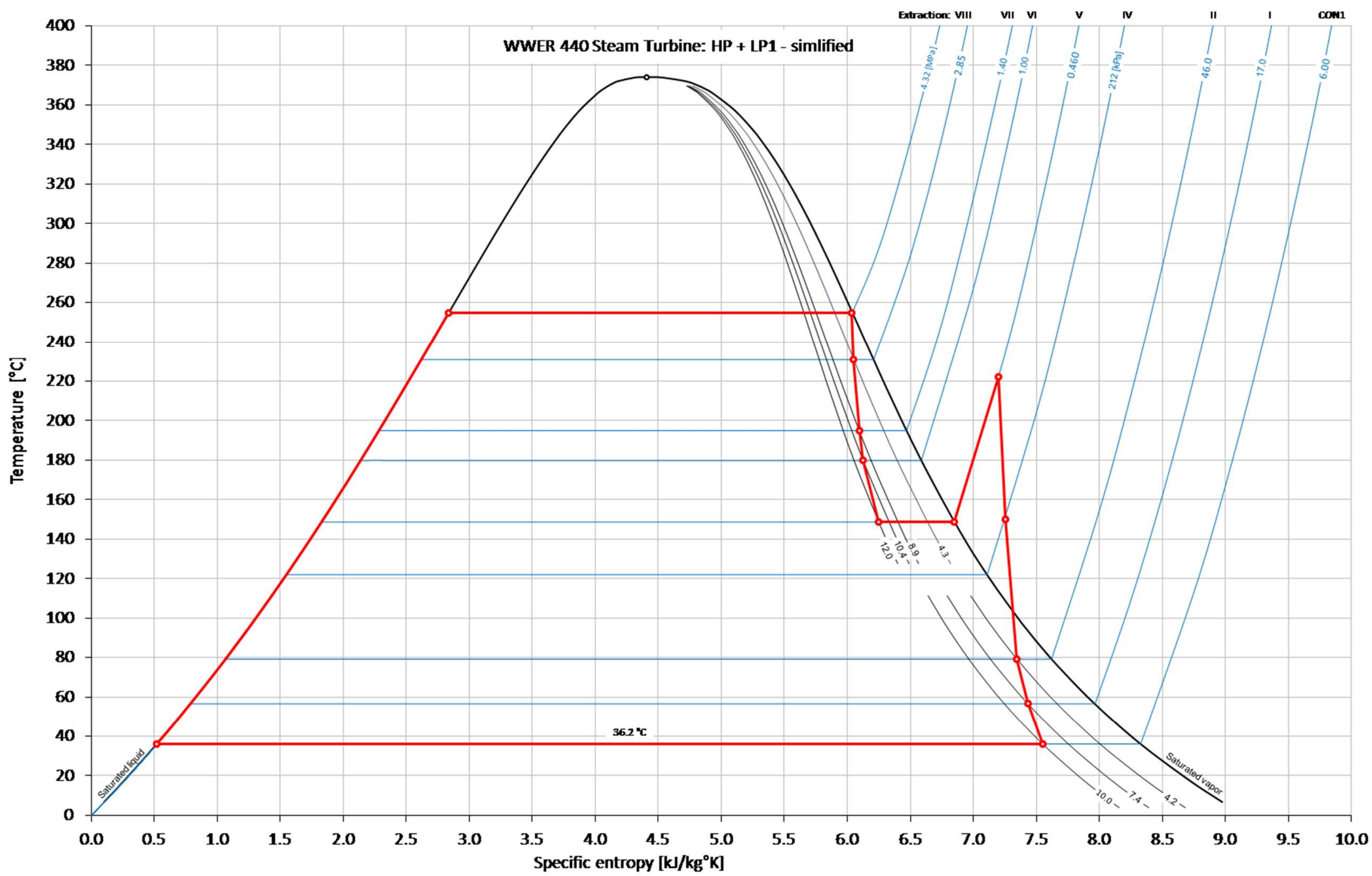


FIG. 83. Real reheat-regenerative Rankine cycle with closed feedwater heater

5.3. OVERALL TURBINE CYCLE PERFORMANCE

The nuclear turbine cycle consists of components, such as the reactor core, SG, steam turbine-generator, MSR, condenser, cooling tower, feedwater heaters, and condensate and feedwater pumps including their driving system. All these components affect the generator power output. The corrected turbine cycle power output needs to be monitored. It is an overall nuclear turbine cycle performance indicator reflecting the combined performance level of these components.

In the power industry, the heat rate vs. electrical power output locus curve has been widely used as plant performance indicator. Unlike in the NPP, especially for those operated at the constant RTP, the measured power output is corrected to the specified reactor (or SG) thermal power. Through this process the corrected power output becomes a measurement of efficiency level rather than of the unit maximum capacity.

According to the definition of the heat rate expressed below in Eq. (51), if the corrected power output is increased by 1%, then the cycle heat rate or efficiency will be improved exactly by 1%. This is because the SG thermal power is a fixed constant.

$$Corrected\ Heat\ Rate = \frac{Specified\ Constant\,SG\ Thermal\ Power}{Corrected\ Power\ Output} \tag{51}$$

It would be more convenient for NPPs to use the corrected power output, instead of heat rate, as a KPI. Even the units for load cycling operation, using the reactor (or SG) thermal power vs. corrected power output curve will be more intuitive rather than the electrical output vs. the heat rate locus curve. This methodology is common in the nuclear industry.

5.3.1. Corrected turbine cycle power output

The corrected turbine cycle power output is defined as the expected generator power output when the cycle is operated at the specified reactor or SG thermal power with any other relevant variables that cross the turbine cycle test boundary at their base reference conditions.

Typical affecting variables external to the turbine cycle test boundary changing the generator power output are;

- Throttle steam pressure;
- Throttle steam moisture content;
- LP turbine exhaust pressure, CW inlet temperature or ambient temperature;
- Generator power factor;
- Steam generator thermal power (MWth).

In an ideal condition in which these affecting variables duplicate the base reference conditions, the measured generator power output may be directly compared to the benchmark power output. However, it is unlikely happening in actual unit operation and therefore it is necessary to correct the measured generator power output for an effect from such deviations.

Mostly the turbine manufacturer provides a set of technical information, such as power output correction curves or tables.

This information considers off-design operating conditions which always occur and may mask the true turbine cycle performance. The corrected turbine cycle power output can be expressed with the following governing Eq. (52);

$$P_{gross,corr} = \left(P_{gross,meas} + \Delta_{pf}\right) \cdot \frac{1}{f_1} \cdot \frac{1}{f_2} \cdot \frac{1}{f_3} \cdot \frac{1}{f_{MWt}} \tag{52}$$

Where $P_{gross,corr}$ is the corrected unit gross power output at generator terminal (kW), $P_{gross,meas}$ is the measured unit gross power output at generator terminal (kW), Δ_{pf} is the additive correction for the generator power factor (kW), f_1 is the multiplicative correction factor for throttle steam pressure, f_2 is the multiplicative correction factor for throttle steam moisture content, f_3 is the multiplicative correction factor for LP turbine exhaust pressure or condenser cooling water inlet temperature and f_{MWth} is the multiplicative correction factor for the SG thermal power.

5.3.2. Diagnostic approaches

Decrease of the corrected turbine cycle power output from the reference or benchmark power output can be caused by the following four general causes.

5.3.2.1. Measurement errors on affecting variable

Errors relating to the measurement of certain affecting variables or variables directly used to calculate the SG thermal power results in incorrect correction of the measured generator power output. Thus, any suspected measurement needs to be double checked using measurement redundancy or physical relationships such as mass and energy balances or the thermo-dynamic property of the wet steam. If test provisions are available, it is also effective to install temporary precision instruments and compare their readings with the permanent sensors.

5.3.2.2. Steam generator thermal power measurement drift

Over-estimation of the SG thermal power caused by fouling of the primary flow meter is one of the biggest contributors to turbine cycle power losses.

This drift can be monitored and quantified using methodologies introduced in Section 4. However, the best approach to recover the power losses from the over-estimation of the SG thermal power is to install additional watchdog meters to adjust the measured primary flow. Another way is to periodically calibrate these meters if they can be dismantled.

5.3.2.3. Valve leakage or auxiliary steam supply

Cycle steam bypassing the turbine through leaking valves or malfunctioning steam traps reduces the generator power output. So, the cycle isolation needs to be periodically checked through bare pipe temperature measurement with an IR temperature detector. Another way is a frequency measurement on the valve body with a noise detector.

5.3.2.4. Severe performance degradation or damage of key components.

Performance indicators of the key turbine cycle components are advised to be continuously checked and trended to rule out the possibility of severe degradation or damage.

5.4. TURBINE

In the turbine cycle, the throttle steam generated from the SG is admitted through control valves to the steam turbine. Thermal energy is converted to kinetic energy and then to mechanical energy by expansion through the turbine stages.

The nuclear steam turbines typically consist of the high pressure (HP) turbine and the low pressure (LP) turbine sections. Most of stages except for the first or up to the second stages of the LP turbine are operated at wet steam region. The HP turbine exhaust steam is going through the MSR where the moisture in the cycle steam is separated and reheated to the superheated steam before entering the LP turbine. That is to improve the cycle efficiency and to protect the LP turbine stages from moisture erosion.

The steam turbine is the primary component that has the greatest impact on the turbine cycle performance. For example, a 1% decrease in HP turbine efficiency produces approximately 0.26% ~ 0.41% decreases in the generator power output. This effect increases significantly to approximately 0.59% ~ 0.74% for the LP turbine depending on the shaft power distribution between the two turbine sections. However, it is very difficult to determine and monitor the efficiency of each turbine section for nuclear applications because both HP and LP turbine exhaust steams are in wet condition. So, it is practical to use the corrected turbine-generator power output according to the ASME PTC 6 [3] as an overall performance indicator of the steam turbine.

5.4.1. Corrected turbine-generator power output

The basic concept of the corrected turbine-generator power output corresponds to that of the corrected turbine cycle power output. However, in order to determine the genuine steam turbine generator performance, additional corrections are required to consider the change of the turbine extraction steam flows. These corrections are aimed at adjustment of the measured generator power output for the feedwater heating system of which performance levels are different from the base reference conditions.

According to the ASME PTC 6 [3], correction of the measured generator power output for affecting variables that cross the turbine-generator test boundary can be categorized as group 1 and group 2 correction as below:

Group 1 Corrections: These corrections include variables primarily consisting of the feedwater heating system. Generator power factor is included in this group of corrections for convenience.

If correction curves (or tables) are used to perform this correction, correction factors for each affecting variable are multiplied to calculate a combined group 1 correction factor. The measured generator power output is (additively) adjusted for the nominal power factor before multiplying the combined correction factor. This correction can be also made by a heat balance calculation using the measured steam turbine performance parameters and the design performance parameters of feedwater heating systems and MSR. The correction curves method is more common in the nuclear turbine cycle.

- Feedwater heater terminal temperature;
- Differences feedwater heater drain-cooler approach differences;
- Extraction line pressure drops and heat losses;
- System water storage changes;
- Feedwater enthalpy rises through condensate and feedwater pumps;
- Condenser-condensate temperature depression;
- Make-up feedwater flow;
- Generator power factor.

The eventual purpose of the group 1 corrections is to correct the measured generator power output for the change of the turbine extraction steam flows as explained in the beginning of this section.

Group 2 Corrections: These corrections include variables relating to the cycle steam conditions entering and leaving the steam turbine. Multiplicative correction is also used to determine the group 2 combined correction factor.

- — HP turbine initial steam pressure;
- — HP turbine initial steam moisture;
- — LP turbine exhaust pressure;
- — Moisture separator effectiveness;
- — Reheater TTD;
- — Cycle steam pressure drops through the MSR.

The eventual purpose of the group 2 correction is to correct the measured generator output for the change in available energy of the HP and LP turbine sections caused by the cycle steam conditions which are different from the base reference conditions. Figure 84 below defines boundaries between turbine-generator test boundary and nuclear turbine cycle test boundaries.

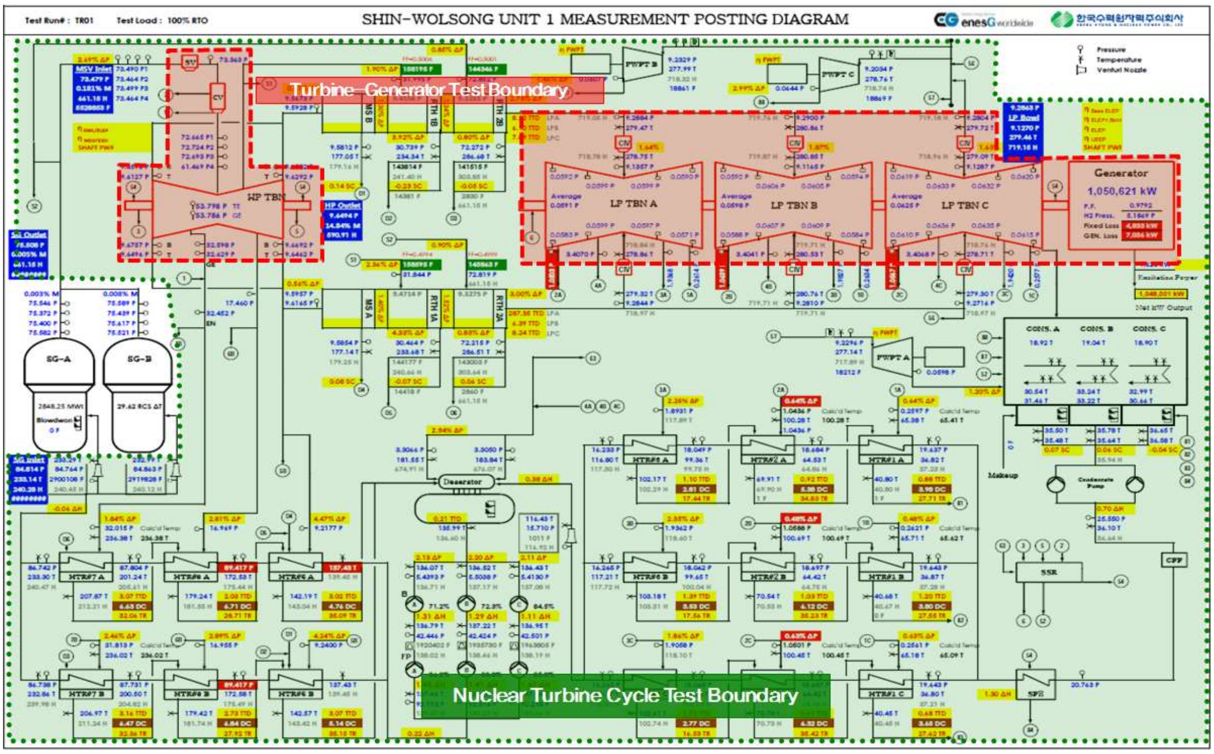


FIG. 84. Turbine cycle and turbine-generator control (or test) boundary

In addition to the Group 1 and Group 2 corrections as described above, The ASME PTC 6 [3] requires correction of the measured generator power output for deviation of the SG thermal power from its reference (typically corresponding to licensed RTP). Through this process the corrected turbine-generator power output becomes a measure of efficiency level, like heat rate, rather than the turbine maximum capacity. In the nuclear turbine cycle, the corrected turbine heat rate is a duplicated performance indicator because of the same reason explained in the introduction of Section 5.3. Another correction mentioned in ASME PTC 6 [3] is throttle losses at the HP turbine control valves. These losses have a direct impact on the turbine-generator power output depending on the control valve positions, the SG thermal power and the throttle steam pressure. In most of the turbines designed with the partial arc HP turbine first stage, these losses are not reflected on the correction curves for the throttle (HP turbine inlet) steam pressure and moisture.

The turbine manufacturer's control valve design data stem lift vs. flow rate for each valve can be used to determine the throttle losses at a certain control valve opening position. In case the nuclear turbine cycle is operated at a constant RTP, the change in kW output losses caused by throttle losses is almost negligible for performance monitoring purpose.

Then, the corrected turbine-generator power output, according to the ASME PTC 6 [3], can be expressed with the following governing Eq. (53);

$$P_{st,corr} = \left(P_{st,meas} + \Delta_{pf}\right) \cdot \frac{1}{\prod_{i=1}^{n} f_{1i}} \cdot \frac{1}{\prod_{i=1}^{n} f_{2i}} \cdot \frac{1}{f_{MWt}} \cdot \frac{1}{f_{throttle}} \tag{53}$$

Where $P_{st,corr}$ is the corrected turbine-generator power output (kW), $P_{st,meas}$ is the measured turbine-generator power output (kW), Δ_{pf} is the additive correction for the generator power factor (kW), f_{1i} is the multiplicative (combined) correction factor for the group 1 corrections, f_{2i} is the multiplicative (combined) correction factor for the group 2 corrections, f_{MWth} is the multiplicative correction factor for the reactor thermal power and $f_{throttle}$ is the multiplicative correction factor for the turbine control valve throttling losses.

5.4.2. Turbine expansion line efficiency

The turbine expansion line efficiency (= turbine isentropic efficiency) of the turbine section is a very important performance parameter in the turbine cycle. Despite that it would not be practical to monitor this performance indicator in NPP. That is because both the HP and LP turbine sections are predominantly operated at the wet region. Instead, the corrected turbine-generator power output is commonly used as a representative performance indicator of the turbine-generator.

However, the HP turbine section efficiency is still necessary performance input data for the test cycle performance analysis and the thermal performance modelling. The ASME PTC 6 [3] suggests a tracer technique or feedwater heater drain flow measurement to determine the wet steam enthalpy. This allows the calculation of the turbine expansion line efficiency. However, both methods are not practical at all for the performance monitoring purpose due to their high measurement uncertainties.

A more realistic approach would be to use a design efficiency level of the HP turbine, such as Δh/ Δ slope of the expansion line, to assume the HP turbine exhaust enthalpy.

Once this property is known the LP turbine exhaust enthalpy can be determined from a heat balance calculation of the overall turbine cycle including the turbine shaft power. This is possible when the turbine extraction steam enthalpies are taken from the HP and LP turbine expansion lines. The turbine shaft power can be calculated from summation of the measured generator output, generator loss at the measured power factor, and mechanical (fixed) loss.

The turbine expansion line efficiency is defined as a ratio of used energy and available energy of each turbine section in the Mollier chart. The HP turbine expansion line efficiency is illustrated in Figure 85 and can be expressed with the following governing Eq. (54):

$$\eta_{HP} = \frac{Used\ Energy}{Available\ Energy} = \frac{h_{MS} - h_{HPEXH}}{h_{MS} - h_{HPEXH\_S}} = \frac{① - ②}{① - ③} \tag{54}$$

Where η_{HP} is the HP turbine expansion line efficiency (%), h_{MS} is the HP turbine inlet main steam enthalpy (kJ/kg), h_{HPEXH} is the HP turbine exhaust steam enthalpy (kJ/kg) and h_{HPEXH_S} is the enthalpy with isentropic expansion from HP turbine inlet to exhaust (kJ/kg).

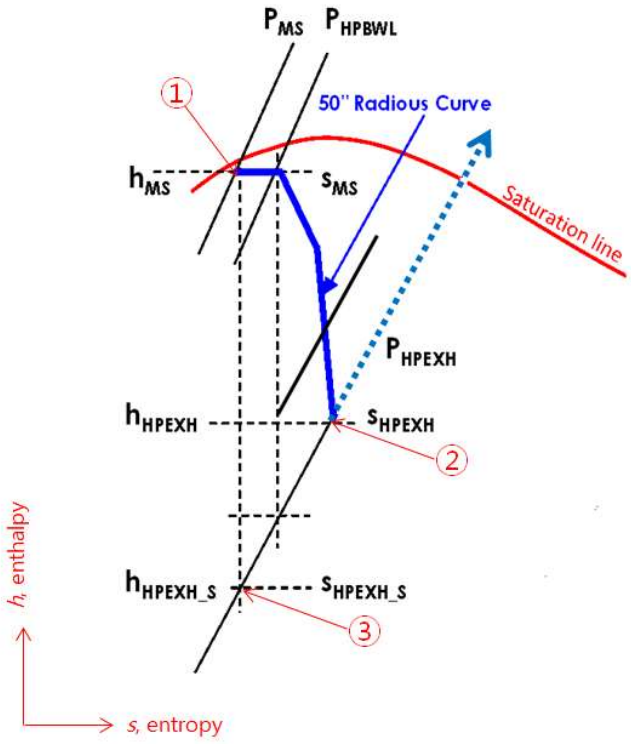


FIG. 85. HP turbine expansion line

The LP turbine expansion line efficiency is illustrated in Figure 86. For the nuclear LP turbines, following three different kinds of the turbine section efficiency can be determined as described in Eqs (55 – 57);

$$\eta_{UEEP} = \frac{Used\ Energy}{Available\ Energy} = \frac{h_{LPBWL} - h_{UEEP}}{h_{LPBWL} - h_{ELEP\_S}} = \frac{① - ②}{① - ⑤} \tag{55}$$

$$\eta_{ELEP} = \frac{Used\ Energy}{Available\ Energy} = \frac{h_{LPBWL} - h_{ELEP}}{h_{LPBWL} - h_{ELEP\_S}} = \frac{① - ③}{① - ⑤} \tag{56}$$

$$\eta_{BASE\ ELEP} = \frac{Used\ Energy}{Available\ Energy} = \frac{h_{LPBWL} - h_{BASE\ ELEP}}{h_{LPBWL} - h_{ELEP\_S}} = \frac{① - ④}{① - ⑤} \quad (57)$$

Where η_{UEEP} is the LP turbine used energy end point (UEEP) efficiency (%), η_{ELEP} is the LP turbine expansion line end point (ELEP) efficiency (%), $\eta_{BASE\ ELEP}$ is the LP turbine BASE ELEP efficiency (%), h_{LPBWL} is the LP turbine bowl steam enthalpy (kJ/kg), h_{LPELEP} is the LP turbine ELEP (kJ/kg), $h_{BASE\ ELEP}$ is the LP turbine base ELEP (kJ/kg), h_{LPUEEP} is the LP turbine UEEP (kJ/kg), h_{ELEP_S} is the enthalpy with isentropic for expansion from LP turbine bowl to exhaust (kJ/kg).

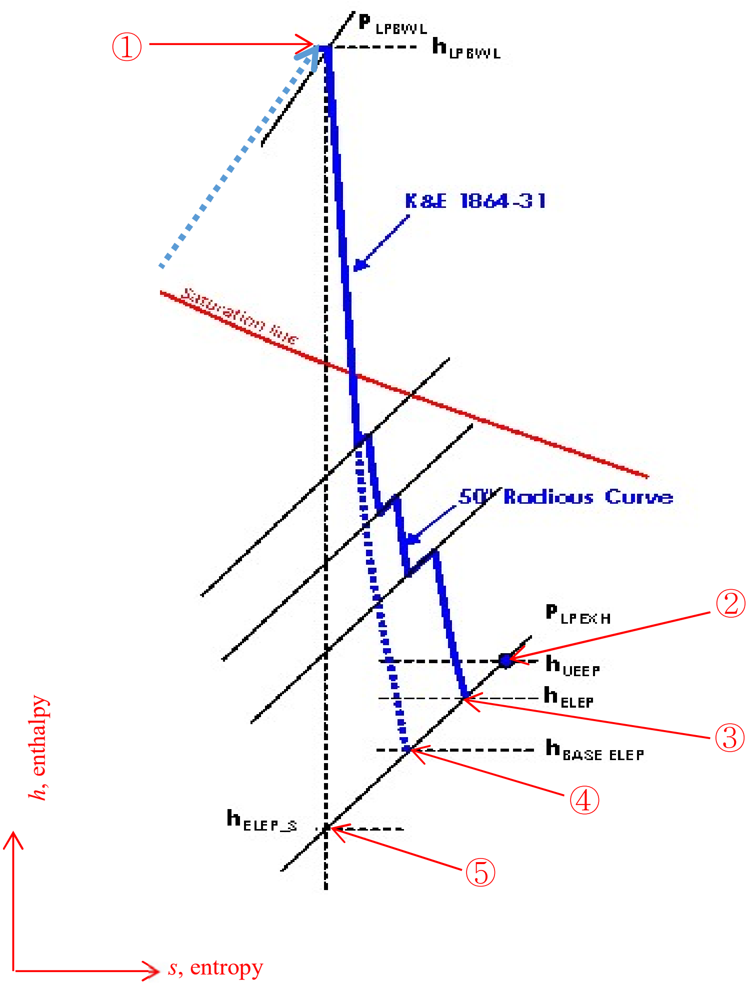


FIG. 86. LP turbine expansion line

The LP turbine UEEP efficiency (η_{UEEP}) is based on the enthalpy of LP turbine exhaust steam actually leaving the last stage blade (h_{UEEP}).

This enthalpy is called UEEP and is calculated from mass and energy balance calculation of the overall turbine cycle including the turbine shaft power as in Eqs (58 – 60) below:

$$m_{LP\ EXH} = m_{MS} - \sum_{j=1}^{n} m_{leakoff}j - \sum_{j=1}^{n} m_{extraction}j - m_{MS\ DRN} \tag{58}$$

$$h_{\ UEEP} = \frac{(m_{MS} \cdot h_{MS} - \sum_{j=1}^{n} m_{leakoff}j \cdot h_{leakoff}j - \sum_{j=1}^{n} m_{extraction}j \cdot h_{extraction}j - m_{MS\ DRN} \cdot h_{MS\ DRN} - KW_{shaft})}{m_{LP\ EXH}} \tag{59}$$

$$KW_{Shaft} = kW_{measured} + Mechanical\ Loss + Generator\ Loss \tag{60}$$

Where $m_{LP\ EXH}$ is the LP turbine exhaust steam flow to condenser (kg/h), h_{UEEP} is the LP turbine exhaust enthalpy (UEEP) (kJ/kg), m_{MS} is the throttle steam *flow* (kg/h), h_{MS} is the throttle steam enthalpy (kJ/kg), $m_{leakoff}$ is the control valves and turbine shaft leak-off steam flow (kg/h), $h_{leakoff}$ is the control valves and turbine shaft leak-off steam enthalpy (kJ/kg), $m_{extraction}$ is the feedwater heater, reheater and FWPT extraction steam flow (kg/h), $h_{extraction}$ is the feedwater heater, reheater and FWPT extraction steam enthalpy (kJ/kg), $m_{MS\ DRN}$ is the moisture separator drain flow (kg/h), $h_{MS\ DRN}$ is the moisture separator drain enthalpy (kJ/kg) *and* KW_{Shaft} is the turbine shaft power.

The LP turbine ELEP efficiency (η_{ELEP}) is based on the enthalpy of LP turbine exhaust steam after expansion to a fictitious stage which is capable of utilizing kinetic energy of the steam leaving the last stage blade (h_{ELEP}). This enthalpy assumes no thermodynamic losses at the LP turbine exhaust and is called ELEP. The LP turbine BASE ELEP efficiency ($\eta_{BASE\ ELEP}$) is based on the expected ELEP when the LP turbine inlet steam is expanded without moisture removal at the LP turbine steam path ($h_{BASE\ ELEP}$).

The expansion line end point (h_{ELEP}) can be calculated from the predetermined used energy end point (h_{UEEP}) using the last stage blade total exhaust loss curve provided by the turbine manufacture with following Eq. (61):

$$h_{ELEP} = h_{UEEP} - TEL \times (1 - 0.01Y) \cdot 0.87 \cdot (1 - 0.0065Y) \tag{61}$$

Where *TEL* is the total exhaust loss (from turbine manufacturer's exhaust loss curve), *Y* is the weighted average moisture at the ELEP (%), *0.87* is the LP exhaust fictitious stage dry efficiency and *1-0.0065Y* is the moisture loss correction.

The LP turbine ELEP efficiency (η_{ELEP}) is more useful performance parameter of the LP turbine. Since the LP turbine UEEP efficiency always changes depending on the exhaust steam pressure and flow, the LP turbine ELEP efficiency maintains at constant level regardless of operating conditions.

5.4.3. Steam expansion ratio (pressure ratio)

There is a useful operation characteristic of steam turbines to monitor from a performance diagnostic perspective. The steam expansion ratio (pressure ratio) across the HP turbine first stage and the LP turbine last stage varies with the throttle steam flow rate and the LP turbine exhaust pressure. The pressure ratio across the remaining stages is always constant if there is no severe change in operating conditions of extraction steam consumers, such as feedwater heaters, FWPT or auxiliary steam headers.

The steam expansion ratio (pressure ratio) is typically expressed with ‘downstream stage pressure divided by upstream stage pressure’. A change of steam expansion ratio through turbine stages without an expected change in extraction steam is indicative of some physical change or damage on those stages.

5.4.4. Steam flow passing capacity (flow factor)

The steam flow passing capacity (flow factor) described in Eq. (62) at the inlet of turbine section and following the extraction stage is a hardware factor of the machine. It needs to be maintained constant regardless of operating conditions.

$$K_{stg} = m_{stg} \cdot \sqrt{\frac{p_{stg}}{v_{stg}}} \tag{62}$$

Where K_{stg} is the steam flow passing capacity (flow factor) of the turbine stage, m_{stg} is the mass flow rate of steam entering the turbine stage (kg/h), P_{stg} is the pressure of steam entering the turbine stage (bar, a) and v_{stg} is the specific volume of steam entering the turbine stage (m^3/kg).

This factor can be used to indicate a change of turbine steam path geometry or overestimation of the RTP. For example, the calculated flow factors of turbine stages will be increased in case of overestimation of the final feedwater flow and vice versa.

5.4.5. Diagnostic approaches

Monitoring of the two turbine key pressures, HP turbine bowl pressure (or first stage) and LP turbine bowl pressure, and throttle flow (or final feedwater flow minus SG blowdown flow) provides a starting point to discuss and derive possible root causes of turbine problems. This may involve a decrease of the corrected turbine-generator power output and corrective actions. Table 13 shows typical feature of change in these parameters.

TABLE 13. DIAGNOSTIC APPROACHES FOR STEAM TURBINE

Possible cause of turbine problems	**Throttle flow**	**P_{HPBOWL}, P_{HPSTG1}**	**P_{LPBOWL}**	**HP turbine efficiency 1)**	**LP turbine efficiency 1)**
Decreased throttle steam pressure	$\boxed{\searrow}$	$\boxed{\searrow}$	$\boxed{\searrow}$	→or $\boxed{\searrow}$ 2)	$\boxed{\nearrow}$
Decreased hot reheat steam temperature (MSR Outlet)	→	→	$\boxed{\searrow}$	→	$\boxed{\searrow}$
Increased HP turbine 1st stage area	→	→	→	$\boxed{\searrow}$	→
Decreased HP turbine 1st stage area	→ or $\boxed{\searrow}$ 2)	→ or $\boxed{\searrow}$ 2)	→ or $\boxed{\searrow}$ 2)	$\boxed{\searrow}$	$\boxed{\nearrow}$
Increased HP turbine 2nd stage area	→	$\boxed{\searrow}$	→	$\boxed{\searrow}$	→
Decreased HP turbine 2nd stage area	→ or $\boxed{\searrow}$ 2)	$\boxed{\nearrow}$	$\boxed{\searrow}$	$\boxed{\searrow}$	$\boxed{\nearrow}$
Increased LP turbine 1st stage area	→	→	$\boxed{\searrow}$	→	$\boxed{\searrow}$

TABLE 13. DIAGNOSTIC APPROACHES FOR STEAM TURBINE (cont.)

Possible cause of turbine problems	Throttle flow	P_{HPBOWL}, P_{HPSTG1}	P_{LPBOWL}	HP turbine efficiency 1)	LP turbine efficiency 1)
Decreased LP turbine 1st stage area	→	→	⧅	→	⧅

1) It is not practical to monitor the HP turbine and LP turbine efficiencies in the nuclear turbine cycle. This table just shows mechanism of turbine performance deterioration.
2) Only if the HP control valves cannot be opened enough to compensate for the flow area change

5.5. MOISTURE SEPARATOR REHEATER

The MSR is used in the nuclear turbine cycle to remove moisture from the HP turbine exhaust steam and reheat this cycle steam into the superheated region before it flows into the LP turbine. Moisture removal reduces the potential of moisture erosion or corrosion on the LP turbine steam path and increases turbine mechanical efficiency. Reheating the cycle steam increases available energy of the LP turbine.

The MSR is mostly configured with moisture separator plus single-stage reheater which uses the main steam to reheat the cycle steam or double-stage reheater which uses the high-pressure turbine extraction steam at the first stage and the main steam at the second-stage. However, some nuclear turbine cycles have only moisture separators without a reheat function.

5.5.1. Moisture separator effectiveness

Moisture separator effectiveness in Eq. (63) is defined as the total moisture flow rate of the cycle steam entering the MSR (or moisture separator) over the mass flow rate of moisture removed from the moisture separators, i.e., the moisture separator drain flow.

$$\eta_{MS} = \frac{m_{msdrn}}{m_{\mathrm{csin,steam}} \cdot \%M_{\mathrm{csin}}} = \frac{m_{msdrn}}{m_{csin,moisture}} \tag{63}$$

Where η_{MS} is the moisture separator effectiveness (%), m_{msdrn} is the moisture separator drain flow rate (kg/h), $m_{csin,steam}$ is the cycle steam flow rate entering moisture separator (kg/h), $\%M_{csin}$ is the moisture content of cycle steam entering moisture separator (%), $m_{csin,moisture}$ is the moisture flow rate entering moisture separator (kg/h).

The final feedwater flow method in ASME PTC 12.4 [17] can be used to determine the cycle steam flow entering or leaving the MSR. This method, which is more preferred, is based on accurate measurement of the SG inlet feedwater flow. The HP turbine extraction flow rates need to be calculated by performing a heat balance calculation around each heater.

It is a practical way to estimate enthalpy of the turbine extraction and exhaust steam from the turbine modelling. If that is not available, then from slope of the design expansion line (Δh/Δs). The turbine control valves and shaft packing leak-off flows need to be directly measured or taken from a best available source, such as the turbine manufacturer's thermal kit.

As an alternative, the flow factor method can be used to determine the cycle steam flow leaving the MSR. This method is based on the total LP turbine inlet flow calculated from the measured LP bowl pressure and temperature and the fixed LP turbine flow factor. The flow factor (K_{lpbowl}) is supposed to be calculated using previous test data. If these are not available, design values may be used as in Eq. (64):

$$m_{lpbowl} = K_{lpbowl,design} \cdot \sqrt{\frac{p_{lpbowl,test}}{v_{lpbowl,test}}} \tag{64}$$

Where m_{lpbowl} is the LP turbine inlet cycle steam flow rate (kg/h), $K_{lpbowl,design}$ is the design LP turbine bow flow factor, $p_{lpbowl,test}$ is the measured pressure of cycle steam entering LP turbine bowl (bar, a), $v_{lpbowl,test}$ is the measured specific volume of cycle steam entering LP turbine bowl (kg/h).

The cycle steam flow leaving the MSR can then be determined by adding measured flow to the feed pump turbine or other components if any.

Moisture separator drain flow can be directly measured or preferably determined from the reheater energy balance calculation shown in Appendix A of ASME PTC 12.4 [17]. This approach depends on the availability of plant flow meters and the measurement uncertainty of this flow.

5.5.2. Reheater terminal temperature difference

The reheater TTD in Eq. (65) is defined as difference between the saturation temperature of the heating steam entering the reheater and the cycle steam temperature leaving the reheater.

$$TTD_{rht} = T_{heating} - T_{csout} \tag{65}$$

Where TTD_{rht} is the reheater TTD (°C), $T_{heating}$ is the saturation temperature of the heating steam entering the reheater (°C) and T_{csout} is the cycle steam temperature leaving the reheater (°C).

The single-stage reheater TTD or the second reheater TTD of a two-stage reheater is determined from directly measured heating steam temperature and cycle steam temperature. However, it is impractical to determine the first reheater TTD of the two-stage reheater because the MSR is normally not instrumented to measure the cycle steam temperature at the first reheater outlet. Even if the temperature is measured, it cannot be a representative first reheater outlet temperature since temperature distribution of the cycle steam is not uniform across the reheater outlet cross section. As such, an energy balance around the second reheater needs to be used to determine the cycle steam temperature at the first reheater outlet. Calculation of the first reheater outlet temperature is shown in Appendix A of the ASME PTC 12.4 [17].

5.5.3. Cycle steam pressure drop

The cycle steam pressure drop in Eq. (66) is defined as the percent pressure drop of the cycle steam entering and leaving the MSR.

$$\%\Delta P_{cs} = \frac{P_{\mathrm{csin}} - P_{csout}}{P_{\mathrm{csin}}} \tag{66}$$

Where $\%\Delta p_{cs}$ is the percent pressure drop of the cycle steam (%), P_{csin} is the cycle steam pressure entering the MSR (bar, a) *and* P_{csout} is the cycle steam pressure leaving the MSR (bar, a).

5.5.4. Diagnostic approaches

Monitoring of the key performance parameters and operating parameters, such as reheater TTD, moisture separator level, heating steam flow (or reheater drain flow) and reheater drain temperature, provide a starting point to discuss and derive possible root causes of MSR problems and corrective actions. Table 14 shows typical changes in these parameters.

TABLE 14. DIAGNOSTIC APPROACHES FOR MSR

Possible cause of MSR problems	**Moisture separator effectiveness**	**Reheater outlet temp.**	**Cycle steam press. drop**	**Moisture separator level**	**Heating steam flow**	**Reheater drain temp**
MS chevron failure	↘	↘	↗	→		
Excess moisture in the incoming steam MS drain system restrictions	↘	↘	↗	↗		
Reheater tube bundle uplift Reheater bypass due to Shroud buckling		↘	↘			
Reheater tube leaks		↘	→		↗	↘
Reheater drain system leaks Emergency drain valve open or leaking		↘	→		↗	↗
Excessive leak of heating steam drain valve Heating steam flow restrictions on reheater heating steam supply line (supply valves not fully open, dropped valve seat, foreign object damage lodged in valve)		↘	→		↘	↘
Reheater tube fouling		↘	→		↘	→

5.6. CONDENSER

Nuclear power plants are so-called condensing plants using ambient cooling sources to condense the steam at the lowest available temperature and under vacuum conditions.

The efficiency of a plant depends, to a great extent, on the integrity and cleanness of the condenser and the cooling system.

One of the primary operating parameters that affects efficiency of the turbine cycle is the condenser pressure. Any performance deficiencies in the condenser have a significant impact on the overall turbine cycle performance, that is, both electrical power output and heat rate. An increased condenser pressure will result in a decrease in the electrical power output and increase in the turbine cycle heat rate due to decreased available energy of the LP turbine.

An increase in condenser pressure is mostly due to one or a combination of internal and external factors such as;

- Internal - degradation of heat transfer surface due to tube plugging or fouling.

- Internal - non-condensable gases in the condenser due to excessive air leakage, degraded performance of steam jet air ejectors or vacuum pumps.
- External - high cooling water inlet temperature or cooling water flow restriction.

The two most commonly used measures of condenser performance are the cleanliness factor (CLF) and the condenser pressure corrected to the reference (design) condenser heat duty, cooling water temperature and flow. Once the benchmark condenser CLF is achieved through a condenser performance testing, an expected condenser pressure at current operating condition can be determined and compared with a measured value. That provides practical information about impact of the condenser performance to the electrical power output and the heat rate at current operating condition.

5.6.1. Cleanliness factor

Note: The calculations described below are based on Heat Exchanger Institute (HEI) Standards [28] method and require knowledge of the material of the condenser as well as the cold inlet flow rate for the HEI curves providing the factor. That information might not be available. In that case it is better to use the log mean temperature difference (LMTD) method or empirical curves.

To determine the condenser CLF, the following condenser performance factors need to be calculated from the current operating parameters starting with the condenser heat load in Eq. (67):

$$Q_{meas} = m_{CW} \cdot C_P \cdot (t_2 - t_1) = \sum_{i=1}^{n} Q_{in,i} - \sum_{j=1}^{n} Q_{out,j} \tag{67}$$

Where Q_{meas} is the condenser heat load at test condition (kJ/h), m_{CW} is the cooling water flow to condenser at test condition (kg/h), C_p is the specific heat at constant pressure (kJ/kgK), t_1 is the measured cooling water inlet temperature (°C), t_2 is the measured cooling water outlet temperature (°C), Q_{in} is the heat entering the condenser (kJ/h) and Q_{out} is the heat leaving the condenser (kJ/h).

Next, the logarithmic mean temperature difference is determined as described in Eq. (68):

$$LMTD = \frac{t_2 - t_1}{In\left(\frac{t_S - t_1}{t_S - t_2}\right)} \tag{68}$$

Where $LMTD$ is the log mean temperature difference (°C), t_1 is the measured cooling water inlet temperature (°C), t_2 is the measured cooling water outlet temperature (°C) and t_S is the saturation temperature at the condenser pressure (°C).

Followed by the overall heat transfer coefficient as shown in Eq. (69):

$$U_{meas} = \frac{Q_{meas}}{As \times LMTD} \tag{69}$$

Where U_{meas} is the measured overall heat transfer coefficient (W/(m^{2}K)), Q_{meas} is the condenser heat load at test condition (kJ/h), As is the tuber surface area (m^2) and $LMTD$ is the log mean temperature difference (°C).

Logarithmic mean temperature difference and the overall heat transfer coefficient (U_{meas}) are very basic factors that indicate the heat exchange level of the condenser. These factors are continuously changing depending on the cooling water inlet temperature and flow and as such do not provide an intuitive judgment about the condenser performance.

In this point of view, it is more practical and confirmative to use the CLF, which is corrected for different cooling water inlet temperature and flow (tube velocity). This remains constant over the entire load range, as the condenser performance parameter. The CLF is defined as the ratio of overall heat transfer coefficient measured at current operating conditions and overall heat transfer expected from the Heat Exchange Institute (HEI) Standards for Steam Surface Condensers [28] as shown in Eq. (70):

$$CLF_{meas} = \frac{U_{meas}}{U_{uc,test} \cdot F_{w,test} \cdot F_M} \tag{70}$$

Where CLF_{meas} is the measured condenser cleanliness factor (%), U_{meas} is the measured overall heat transfer coefficient (W/(m^{2}K)), $U_{uc,\mathrm{test}}$ is the uncorrected overall heat transfer coefficient at test tube velocity (W/(m^{2}K))(from HEI Standards [28] Table 1), $F_{w,test}$ is the correction factor for circulating water temperature at test cooling water temperature (from HEI Standards [28] Table 2) and F_M is the correction factor for material and gauge (from HEI Standards [28] Table 3).

Typically, in NPPs, if the condenser CLF decreases by 3%, the condenser pressure increases by 1 mmHg, and vice versa.

5.6.2. Corrected condenser pressure

From the predetermined condenser CLF, the overall heat transfer coefficient (U_{corr}) corrected for the reference (design) cooling water inlet temperature and flow can be calculated using the HEI method as below. The condenser pressure can be corrected to these external variables using the effectiveness-NTU method.

From the determined CLF_{meas}, corrected heat transfer coefficient can be calculated using Eq. (71):

$$U_{corr} = U_{uc,design} \cdot F_{w,design} \cdot F_M \cdot CLF_{meas} \tag{71}$$

Where U_{corr} is the corrected overall heat transfer coefficient (W/(m^{2}K)), $U_{uc,design}$ is the uncorrected overall heat transfer coefficient at design tube velocity (W/(m^{2}K))(from HEI Standards [28] Table 1), $F_{w,design}$ is the correction factor for circulating water temperature at design cooling water temperature (from HEI Standards [28] Table 2), F_M is the correction factor for material and gauge (from HEI standards [28] Table 3) and CLF_{meas} is the measured condenser cleanliness factor (%).
Using U_{corr}, the number of heat transfer unit can be determined as shown in Eq. (72):

$$NTU = \frac{U_{corr} \cdot A_S}{C_P \cdot m_{cw}} \tag{72}$$

Where NTU is the number of heat transfer unit, U_{corr} is the corrected overall heat transfer coefficient (W/(m^{2}K)), A_s is the tuber surface area (m^2), m_{cw} is the cooling water flow to condenser at test condition (kg/h) and C_P is the specific heat at constant pressure (kJ/kgK).

Corrected condenser pressure in Eq. (73) can be determined for the saturation temperature described in Eq. (74):

$$t_S = \frac{t_2 - t_1 \cdot e^{-NTU}}{1 - e^{-NTU}} \tag{73}$$

$$P_{corr} = s\text{aturation pressure at } t_S \tag{74}$$

Where t_s is the saturation temperature at the condenser pressure (°C), NTU is the number of heat transfer unit (-), t_1 is the measured cooling water inlet temperature (°C), t_2 is the measured cooling water outlet temperature (°C).

The corrected overall heat transfer coefficient can be calculated using the ASME PTC 12.2 [16] method with the following Eq. (75):

$$U_{corr} = \frac{1}{R_m^* + R_t^* \cdot \left(\frac{d_o}{d_i}\right) + R_f^* + R_s^o} \tag{75}$$

Where U_{corr} is the corrected overall heat transfer coefficient (W/(m^{2}K)), R_m* is the tubewall resistance at design conditions (m^{2}K/W) (refer to ASME PTC 12.2 [16] Section 5, 5.2.1), R_t * is the tubeside thermal resistance at design conditions (m^{2}K/W) (refer to ASME PTC 12.2 [16] Section 5, 5.2.2), R_f * is the fouling resistance at design conditions (m^{2}K/W) (refer to ASME PTC 12.2 [16] Section 5, 5.2.3), R_so is the shell side resistance adjusted to design conditions (m^{2}K/W) (refer to ASME PTC 12.2 [16] Section 5, 5.2), d_O is the tube outside diameter (m) and d_i is the tube inside diameter (m).

The application of this method is less practical since it is more complicated and the code itself limits the operating conditions such as heat load (±5% from design), cooling water inlet temperature (±5.6℃ from design) and cooling water flow (±5% from design).

5.6.3. Expected condenser pressure

The condenser pressure at current operating conditions can be determined using the HEI method described in Section 5.6.2. This method can be applied using the benchmark condenser CLF from the previous testing (instead of CLF_{meas}) and the current cooling water inlet temperature and flow to determine U_{uc} and F_w at current condition. If there is no benchmark condenser CLF available, the design CLF can be tuned so as the expected condenser pressure to reach the measured value at the time with no tube fouling and/or excessive air leakage are suspected.

5.6.4. Diagnostic approaches

Monitoring of the key performance parameters and operating parameters (e.g. condenser CLF, LMTD, cooling water temperature rise and tube pressure drop) provide a starting point to discuss and derive possible root causes of condenser problems. That is high condenser pressure and low CLF. Table 15 shows typical feature of change in these parameters.

TABLE 15. DIAGNOSTIC APPROACHES FOR CONDENSER

Possible cause of condenser problems	Non-condensable gas(O_2)	LMTD (or TTD)	Cooling water temp. rise*	Cooling water pressure drop
Microfouling	→	$\boxed{\nearrow}$	→	→
High hotwell level Air binding (air removal problem or air leakage)	$\boxed{\nearrow}$	$\boxed{\nearrow}$	→	→
High heat rejection to condenser	→	$\boxed{\nearrow}$	$\boxed{\nearrow}$	→
Increased cooling water system resistance (cooling water pump head increase)	→	→	$\boxed{\nearrow}$	$\boxed{\searrow}$
Cooling water pump performance deterioration. Motor current: — increase: damaged casing or impeller — fluctuate: pump cavitation — decrease: casing or impeller wear/corrosion	→	→	$\boxed{\nearrow}$	$\boxed{\searrow}$

* Cooling water temperature rise at the same cooling water pumps operation profile.

5.7. COOLING TOWER

"Circulation water cooling using cooling towers is generally the least effective technique due to the higher temperatures of the cooling medium. A temperature gradient between the steam and the cooling medium and the environment is always needed." [27]

Cooling towers are used to remove heat from circulating water. They provide resulting water temperature low enough to allow condenser pressure to be maintained at the constant level and to remove the rejected heat in the condenser.

Cooling towers used in the nuclear industry are typically natural draft or mechanical draft, depending on climatic conditions and economics. Cooling towers are designed for the expected meteorological conditions at the NPP location and may operate in a cross flow or counter flow configuration with respect to the air and water in the cooling region.

Using cooling towers for heat removal ultimately results in lower plant efficiency than most 'once through' plant designs which rely on a local water source. Even if water is available many plants have installed cooling towers to minimize environmental impacts.

Cooling towers are also installed in 'once through' systems and are designed to cool water exiting a condenser before it is returned to the water source. This is considered as an open cooling system. A closed cooling system only takes needed make-up flow from the water source. The effectiveness of the closed system is limited by the wet-bulb temperature of the air and is be susceptible to changing weather conditions. An open system is limited by the temperature of the water source that supplies the circulating water.

Forced draft cooling towers that use fans to move the air are used typically in dry climates because fan power costs are low and the smaller towers have a lower capital cost. Natural draft towers are typically used only at larger facilities and in areas where the relative humidity is high when compared to dryer regions. However, the installation costs and footprint of a natural draft tower often results in the use of mechanical draft tower at locations more suitable to a natural draft tower.

A cooling tower cools water by sensible heat exchange (temperature) and latent heat exchange (evaporation) caused by the air moving through the tower and exchanging energy with the water as it passes through the tower as shown in Figure 87. All the energy lost by the water is added to the air as well as some of its mass. The coolest theoretical temperature that can be attained is the adiabatic saturation temperature. This property of general gas mixtures can be measured for air-water vapor mixtures by a wet-bulb thermometer. This is because the wet-bulb temperature is within one degree of the adiabatic saturation temperature for air-water vapor mixtures. It may also be calculated from the dew point and dry-bulb temperatures.

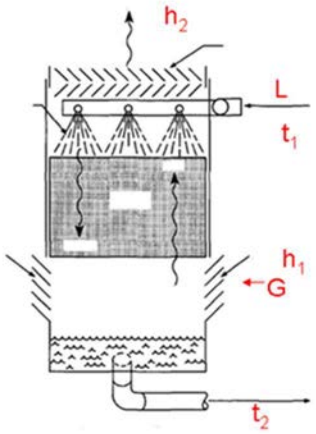


FIG. 87. Cooling tower energy exchange

The process of water cooling is described in Eqs (76,77).

- A cooling tower is a heat exchanger where two fluids (air and water) are brought into direct contact.
- The heat gained by the air equals the heat lost by the water.

$$L \cdot (t_1 - t_2) = G \cdot (h_2 - h_1) \tag{76}$$

$$\frac{L}{G} = \frac{h_2 - h_1}{t_1 - t_2} \tag{77}$$

Where G is the mass flow of dry air (kg/h), h_1 is the enthalpy of entering air (dry air) (kJ/kg), h_2 is the enthalpy of leaving air (dry air) (kJ/kg), L is the mass flow of water (kg/h), t_1 is the hot water temperature entering tower (°C) and t_2 is the cold water temperature leaving tower (°C).

Air flow through a natural draft tower occurs due the buoyancy of the air being warmed and wetted to move through the tower. Mechanical draft tower described in Eq. (78) utilizes fans to move the air through the tower. It essentially operates like a large chimney where the rate of cooling air flow is proportional to the air-vapor density gradient that is established between the ambient environment and the conditions inside of the tower.

$$Draft = \frac{Shell\ weight}{0.0000979\ \frac{bar}{MWC}} \times (\rho_{ambient} - \rho_{exit}) \tag{78}$$

Where *0.0000979* bar/MWC is the conversion from bar to millimetre water column, *shell weight* is the cooling tower shell weight (kg), $\rho_{ambient}$ is the density of the environment (kg/m^3) and ρ_{exit} is the density of the exiting air (kg/m^3).

The cooling region of the tower consists of the water distribution system which is directed over packing (or fill) that increases the water surface area for evaporation and aids in attaining the optimum water to airflow ratio. There are various types of cooling tower fill. Film fill causes a sheeting of the water over the fill material. Film type fill may be sheet or cellular in design. Splash fill causes the atomization of the water into droplets which allow the transfer to take place as the droplets fall through the tower. The splash type fill disperses the water as droplets through horizontal or vertical air flow.

The film fill is typically more efficient than the splash fill. However, the film fill surface area is more apt to become biofouled. Thus, the splash type fill while it is less efficient is less likely to foul which makes it more reliable. Film type fill while it is more efficient is more likely to foul making it less reliable. Generally speaking, there is a trade-off between fill fouling tolerance and efficiency.

5.7.1. Cooling tower performance monitoring

Cooling tower performance can be monitored in various ways, from a simple comparison of cold-water temperature to wet-bulb temperature to performance of a cooling tower capability calculation. Use of data reconciliation can also provide a good monitoring methodology for cooling tower performance. The chosen method depends on the type of engineering tools at the site, available instrumentation and the specific tower design. The important goal is to evaluate how the cold-water temperature compares to the cooling tower design at a given set of conditions.

Unlike other plant equipment the boundary conditions for a cooling tower (wet-bulb, dry-bulb, relative humidity, wind direction and wind speed) can change significantly over a relatively short period of time. Wind speed can have an effect of several degrees on the cold-water temperature. Depending on the tower design and location relative to other towers, recirculation of the tower exiting air back to the inlet air can cause the tower performance to decrease as much as 20%. Meteorological conditions need to always be considered when trying to understand the performance parameters of the cooling tower.

A simple way to trend cooling tower performance is to develop a predicted curve based on empirical data or vendor data to estimate the expected cold-water temperature for a given wet-bulb temperature (see Figure 88).

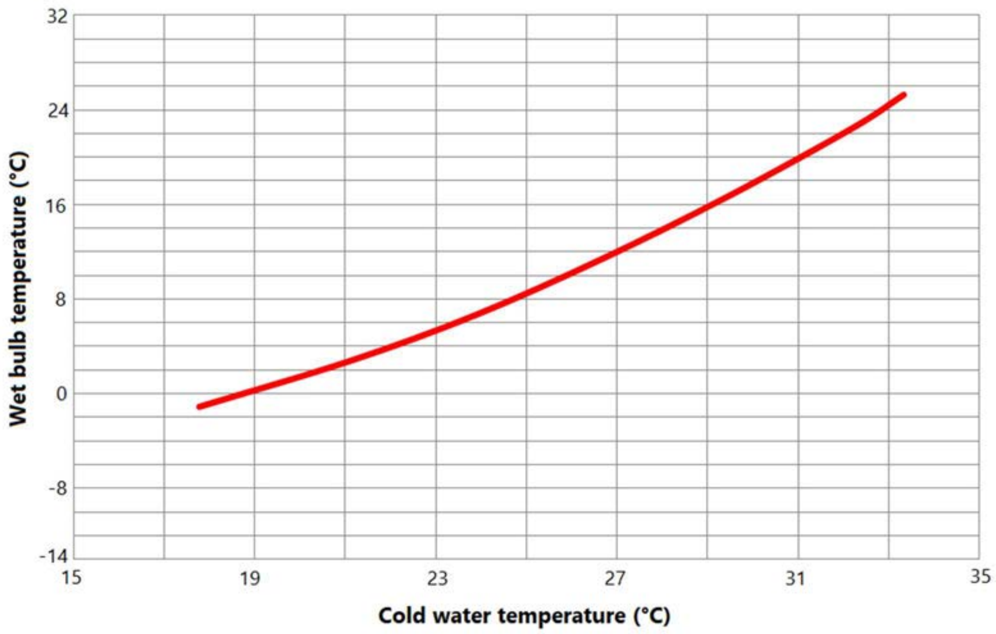


FIG. 88. Wet-bulb temperature vs cold water temperature

The range and the approach temperatures are important for understanding cooling tower performance (see Figure 89). The range is the temperature decrease of the cooling water as it is cooled by the cooling tower and affected by the heat rejected by the condenser and the circulating water flow rate. Range is the difference between hot water temperature and cold water temperature. Approach is the difference between cold water temperature and wet-bulb temperature.

At a constant thermal power and circulating water flow the range is constant over the operating cycle. There are fluctuations in the range value as the cold water temperature changes and affects the amount of heat rejected by the LP turbine to the condenser. The approach temperature is the difference between the wet bulb temperature and the cold water temperature and affected by the tower efficiency and meteorological conditions. As the tower efficiency changes the range will stay the same but the approach will change through the operating cycle.

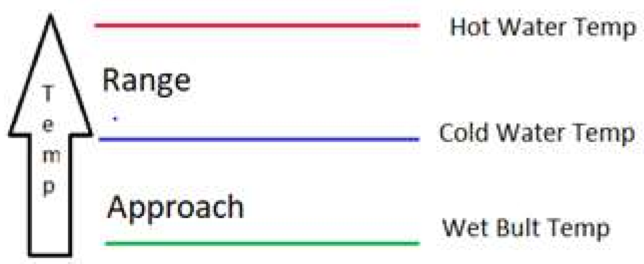


FIG. 89. Approach and range temperatures

5.7.2. Cooling tower capability calculation

The Cooling Technology Institute Standards for cooling tower testing provide instructions for performance of cooling tower capability calculations. Cooling tower capability is basically a comparison of the expected cooling water flow based on the cooling tower performance curves and the actual cooling water flow being cooled by the tower.

The calculations described in the code of the Cooling Technology Institute Standards are used to develop a cooling tower performance calculation for use in tracking cooling tower performance. The calculations can also be used to predict the cold-water temperature based on the tower design and measured parameters.

This predicted cold-water temperature can be compared with the actual cold-water temperature achieved. The cold-water temperature can then be used in conjunction with the condenser performance curves or calculations to determine the condenser pressure and expected megawatts. Using these calculations, the control volume for the turbine cycle can be extended to include the cooling tower allowing the calculation of expected generation to be based on meteorological conditions.

Data required for calculation:

- Cold water temperature (CWT);
- Hot water temperature (HWT);
- Wet-bulb temperature (WBT);
- Circulating water flow rate (Flow);
- Fan horsepower (FHP);
- Make-up temperature;
- Make-up flow rate.

Data required for validation:

- Water analysis (total dissolved solids, organics);
- Wind velocity.

Range is the temperature change in the water affected by the cooling tower, (HWT – CWT). There is a design range and an actual range measured during operation.

Approach is the temperature difference between the wet-bulb temperature and the circulating water temperature, (CWT-WBT).

The capability calculation is based on the measured parameters and the vendor performance curves.

The following example illustrates the process of calculating the capability of a cooling tower using the performance curve method. Performance test measurement locations are shown in Figure 90.

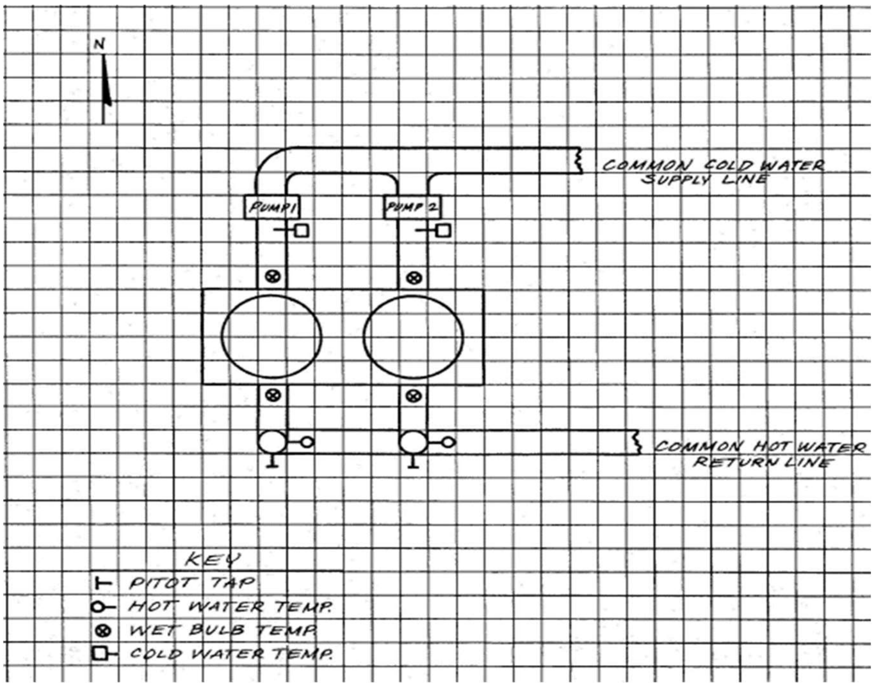


FIG. 90. Performance test measurement locations

TABLE 16. COOLING TOWER PERFORMANCE TEST DATA SHEET

	Units	**Design**	**Test**
Water flow rate	m^3/h	2 271.25	2 124.75
Number of operating cells		2	2
Fan HP per cell	kWe	89.484	80.536
ACFM per cell	m^3/h	1 236 472	N/A
Elevation	m^3/h	sea level	sea level
Barometric pressure	bar,a	1.013	1.012
Hot water temperature	°C	46.11	40.39
Cold water temperature	°C	29.44	26.28
Range (HWT - CWT)	°C	16.67	14.11
Inlet wet-bulb temperature	°C	26.67	22.83
Approach (CWT - WBT)	°C	2.78	3.44
Pumping heat	m	N/A	
Total dissolved solids	ppm	4 000	3 565
Oil content		5	1
L/G		0.86	0.815
KaV/L		2.336	2.335
Characteristic curve slope		-0.06	
Capability			100.6%

(i) Step 1: Perform cooling tower performance test (data shown in Table 16) and record the following data as shown in Table 17.

- Cold water temperature (CWT) (°C);
- Range (R) (°C);
- Wet-bulb temperature (WBT) (°C);
- Circulating water flow rate (Flow) (m^3/h);
- Fan Power FHP (kWe).

TABLE 17. COOLING TOWER PERFORMANCE TEST DATA

Step (1): Test data		
Parameter	**Reading**	**Units**
Flow	2124.75	m^3/h
CWT	26.28	°C
WBT	22.83	°C
Range	14.11	°C
Fan Power	80.54	kWe

(ii) Step 2: Read predicted values from vendor supplied performance curves (as in Figure 91) at measured wet-bulb temperature (22.83°C), graphs are supposed to encompass conditions of values based on the test data (as in Figs 92 – 94), other graphs may be necessary if measured flow is outside the provided graphs.

Range (℃)	Flow (m^3/hr)		
	2,044.12	2,271.25	2,498.37
13.33	25.84	26.33	26.88
16.67	26.23	26.79	27.37
20.00	26.53	27.16	27.75

FIG. 91. Design parameters

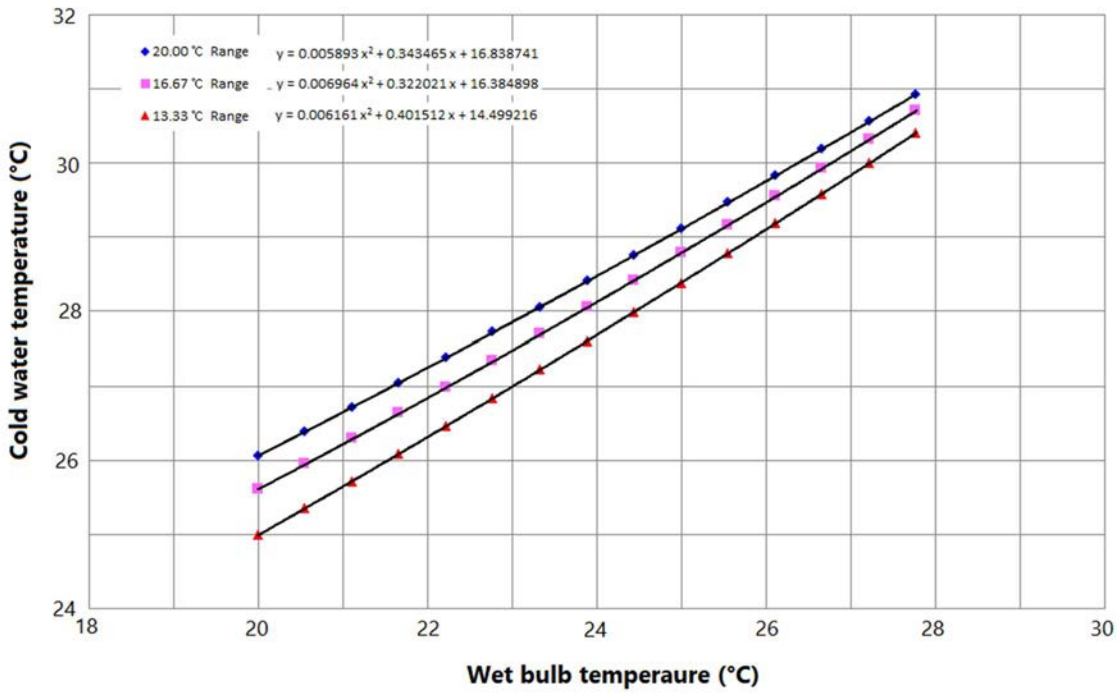


FIG. 92. Vendor performance curve – 2498.37 m3/h (Step 2)

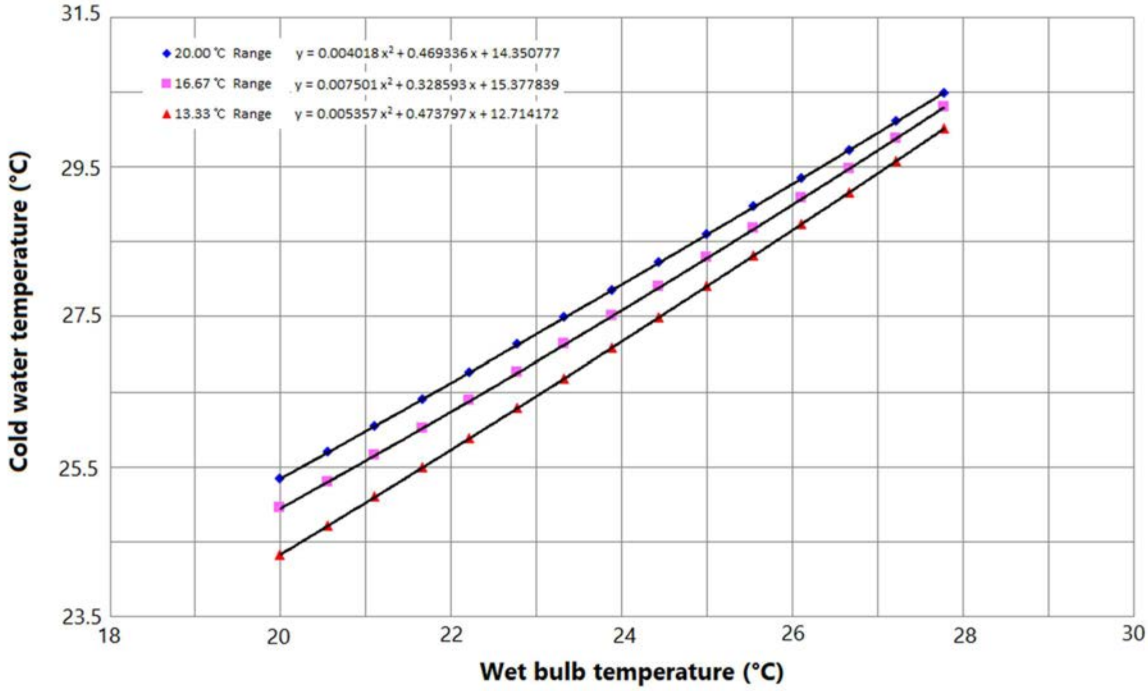


FIG. 93. Vendor performance curve – 2271.25 m3/h (Step 2)

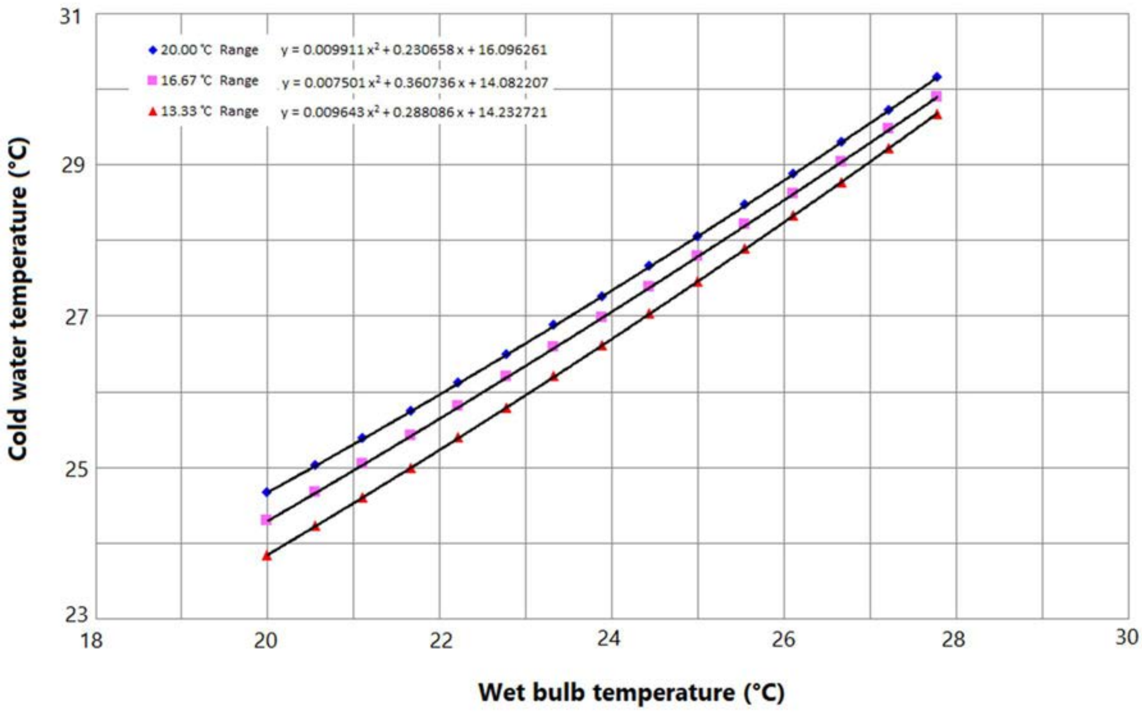


FIG. 94. Vendor performance curve – 2044.12 m3/h (Step 2)

(iii) Step 3: Plot data from Step 2 to produce cold water temperature (results in Figure 95) vs. range curves (see Figure 96). Solve for each flow curve at the measured range (14.11°C).

Flow (m^3/hr)	CWT (℃)
2044.12	25.94
2271.25	26.44
2498.37	27.00

FIG. 95. Cold water temperature vs. flow curve data

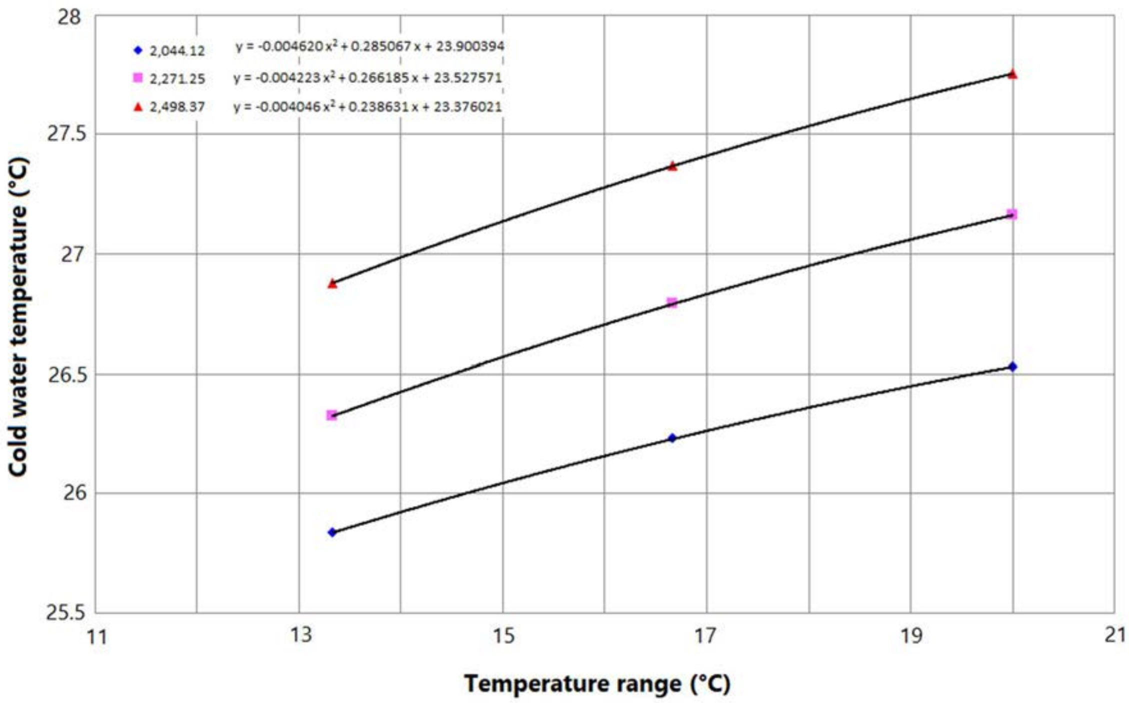


FIG. 96. CWT vs range temperature for each flow (Step 3)

(iv) Step 4: Determine predicted flow

Cross plots Step 3 graph to produce a cold water temperature vs. predicted flow curve (see Figure 98). Read from this curve the predicted flow (results in Figure 97) at the measured cold water temperature (26.28°C).

CWT (℃)	Flow (m^3/hr)
26.28	2199.40

FIG. 97. Cold water temperature vs. predicted flow curve

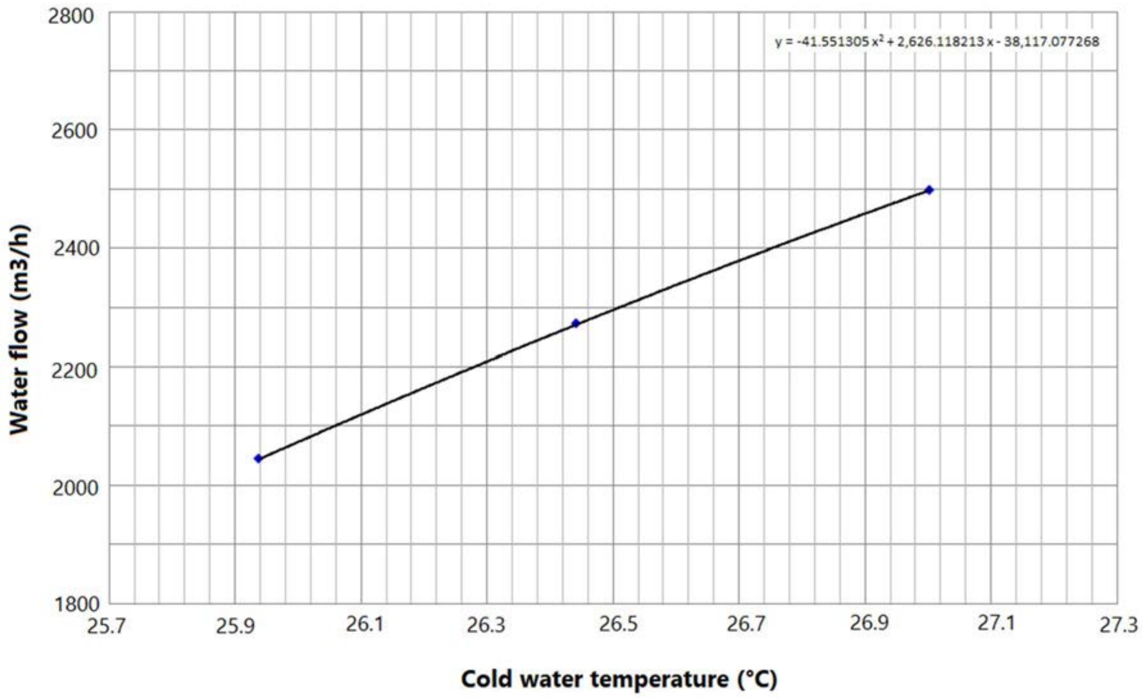


FIG. 98. CWT vs predicted flow and test range and wet bulb temperature (Step 4)

(v) Step 5: Compute adjusted test flow according to Eqs (79, 80):

$$\text{Adjusted test flow} = \text{Test flow} \times \left(\frac{\text{Design fan HP}}{\text{Test fan HP}}\right)^{1/3} \tag{79}$$

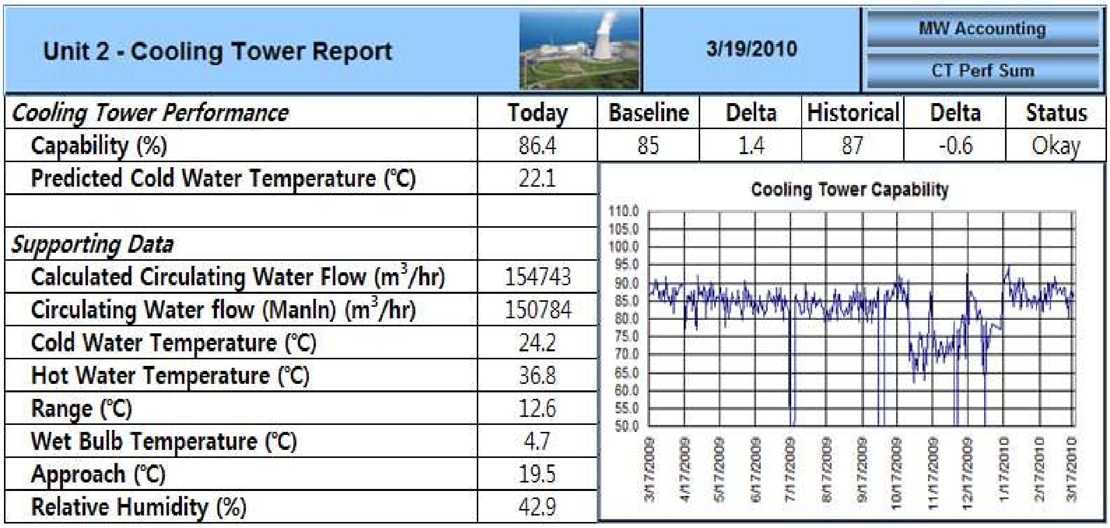
$$\text{Adjusted test flow} \ = \ 2124.75 \cdot \left(\frac{89.484}{80.536}\right)^{1/3} \ = \ 2200.77 \text{ m3/h} \tag{80}$$

(vi) Step 6: Compute performance (see Eq. (81)) from the ratio of adjusted test flow from Step 5 to predicted flow from Step 4.

$$\% \text{ Capability} = \frac{\text{Adjusted test flow}}{\text{Predicted flow}} \times 100 \ \% \tag{81}$$

TABLE 18. COMPUTED RESULTS

Predicted flow	2199.40	Look up from curve (Step 4)
Adjusted test flow	2200.70	Result (of Step 5)
Capability	100.06%	Adjusted/Predicted x 100



Cooling Tower Performance	Today	Baseline	Delta	Historical	Delta	Status
Capability (%)	86.4	85	1.4	87	-0.6	Okay
Predicted Cold Water Temperature (°C)	22.1					
Supporting Data						
Calculated Circulating Water Flow (m^3/hr)	154743					
Circulating Water flow (ManIn) (m^3/hr)	150784					
Cold Water Temperature (°C)	24.2					
Hot Water Temperature (°C)	36.8					
Range (°C)	12.6					
Wet Bulb Temperature (°C)	4.7					
Approach (°C)	19.5					
Relative Humidity (%)	42.9					

FIG. 99. Example cooling tower monitoring

Computed results of the capability of the cooling tower from the performance test are shown in Table 18 above. Example of the cooling tower monitoring report is shown in Figure 99.

5.7.3. Diagnostic approach

Cooling tower failure mechanisms are listed below:

- Fan failures;
  - Vibration.
  - Gearbox problems.
  - Blade alignment.
- Fill clogging;
  - Debris, biofouling.
- Fill failures;
  - Ice damage.
  - Drift eliminators falling into fill.
  - Water logging causing fill separation.
  - Damage during maintenance.
- Drift eliminator clogging;
- Flow balance problems;
  - Design or spray nozzle clogging.
  - Flow distribution deck clogging.
  - Bypass flow either air or water.
- Flow piping failure;
  - Start-up problems.
  - Leaking joints on distribution plenums.

— Cooling tower support failures;
  - Hypochlorite attack on cement.
  - Corrosion of fasteners or support structures.
  - Wind barrier damage resulting in excessive exposure of fill to wind damage.

— Water chemistry not maintained.

Monitoring of the key performance parameters and operating parameters provide a starting point to discuss and derive possible root causes of cooling tower problems and corrective actions. These include range, approach, capacity, wet bulb temperature (WBT), and cold water temperature (CWT). Table 19 shows typical parameter changes based on the possible problem.

TABLE 19. DIAGNOSTIC APPROACHES FOR COOLING TOWER

Possible cause of cooling tower problems	WBT	Range	Approach	CWT	Capacity
• **Fill fouling** Possible follow-up: - Check biocide treatment system - Schedule to clean fill • **Improper air/water distribution or air bypassing sections of tower:** - Fan failure or OOS - Loss of fill - Fill support structure failure - Algae blockage in distribution rings or decks - High winds - Freeze damage Possible follow-up: - Perform air mapping (natural draft tower) - Verify fan operation and pitch - Check for broken water piping - Inspect flow nozzles - Inspect flow distribution deck for blockage - Check for holes in tower allowing air to pass around fill - Verify ambient conditions	→	→	$\boxed{\nearrow}$	$\boxed{\nearrow}$	$\boxed{\searrow}$
• **Reduced CW flow** Possible follow-up: - Verify no bypass flow - Verify flow distribution system is operating correctly especially in cases with multiple towers - Check pump operations	→	$\boxed{\nearrow}$	→	$\boxed{\nearrow}$	$\boxed{\searrow}$ (unless flow is measured then it might be the same or improve slightly)

5.8. FEEDWATER HEATER

In order to improve the cycle efficiency, most NPPs have a string of both HP and LP feedwater heaters that preheat the feedwater entering the PWR SG or BWR reactor vessel. This process is regenerative; that is, by using the heat available in turbine extraction steam that has been partially expended through the turbine.

For nuclear turbine cycle application, two-zone feedwater heaters that condense the extraction steam (a condensing zone) and subcool condensate by means of internal or external drain cooler (a drain cooling zone), which transfer additional heat to the feedwater.

The two most commonly used measures of feedwater heater performance are the TTD and the DCA. The reference values for both TTDs and DCAs are typically obtained from the design heat balance diagrams because they are readily available and easily configured in a thermodynamic turbine cycle modelling.

5.8.1. Terminal temperature difference

Terminal temperature difference of feedwater heater is defined as difference between the saturation temperature of extraction steam entering the feedwater heater and the feedwater temperature at heater outlet as in the Eq. (82):

$$TTD_{htr} = T_{ext,sat} - T_{fwout} \tag{82}$$

Where TTD_{htr} is the feedwater heater TTD (°C), $T_{ext,sat}$ is the saturation temperature of extraction steam entering the feedwater heater (°C) and T_{fwout} is the feedwater heater outlet temperature (°C).

5.8.2. Drain cooler approach

Drain cooler approach of feedwater heater is defined as difference between the feedwater heater drain temperature and the feedwater temperature at heater inlet as in the Eq. (83):

$$DCA_{htr} = T_{drn} - T_{fwin} \tag{83}$$

Where DCA_{htr} is the feedwater heater DCA (°C), T_{drn} is the drain temperature leaving the feedwater heater (°C) and T_{fwin} is the feedwater heater inlet temperature (°C).

For the feedwater heaters in the nuclear turbine cycle operating at a constant RTP, their TTDs and DCAs are also maintained constant and the measured values can be directly used for performance monitoring purpose.

For the lowest feedwater heater of which inlet feedwater temperature varies with corresponding condenser pressure and the feedwater heaters in load cycling unit, more rigorous approach is used. The manufacturer's design data in conjunction with basic heat transfer relationships, such as effectiveness-NTU method, is used to compute expected TTDs and DCAs.

This calculation is exampled in Appendix B of ASME PTC 12.1 [4]. The calculated result is then compared with the measured values to identify any performance deterioration.

The detailed design information required from the manufacturer is as follows:

- Heat transfer surface areas for each heater zone;
- Steam and water-side fouling resistances;
- Steam and water side film resistances;
- Heat transfer rate for each zone;

— Inlet and exit pressures, temperatures, and flows for the extraction steam, feedwater flow, and drain flow.

The measured data required for the calculation is as follows:

— Inlet steam temperature and pressure;
— Feedwater inlet and exit temperatures and pressure;
— Drain flow temperature and flow rate;
— Shell side pressure.

5.8.3. Diagnostic approaches

Monitoring of the key performance parameters and operating parameters, such as TTD, DCA, temperature rise and feedwater heater level, provide a starting point to discuss and derive possible root causes of feedwater heater problems and corrective actions. Table 20 shows typical feature of change in these parameters. Triggering points for changes of DCA to detect the feedwater heater problems need to be higher than that of TTD. That is because DCA is more sensitive to operating conditions such as feedwater heater level. For example, changes of TTD by 1°C may trigger diagnostics of the feedwater heater problems, but in this case DCA is supposed to be at least 2°C.

TABLE 20. DIAGNOSTIC APPROACHES FOR FEEDWATER HEATER

Possible cause of heater problems	TTD	DCA	Feedwater temp. rise	Heater level
Feedwater heater tube leak (rapid level increase) Heater level controller or normal drain valve malfunction (level high)	↗	↘	↘	↗
Drain cooling zone flashing (steam induction through snorkelling) Heater level controller or normal drain valve malfunction (level low)	→ or ↘	↗	→ or ↗	↘
Emergency drain valve open or excessive leaks	→ or ↘	↗	↘	↘
Tube fouling corrosion Pinhole leaks near steam or drain inlet (rapid)	↗	↗	↘	→
Normal vent closed or air leaks into feedwater heater (heaters operating under atmospheric pressure)	↗ Oscillate	↗ or →Oscillate	↘	→
Normal vent closed (heaters operating above atmospheric pressure) Partition plate leaks	↗	→	↘	→

5.9. FEEDWATER PUMPING SYSTEM (TURBINE DRIVEN)

Most of the large-scale nuclear turbine cycles use the steam turbine driven feedwater pumping system to pump feedwater into the SG. The FWPT driving steam is extracted from the MSR outlet. The FWPT driving steam flow rate expressed as a percent of the SG outlet steam flow can be used as performance indicator of the feedwater pumping system.

Feedwater pump turbine and pump efficiencies are theoretical performance indicators of the pumping system, but these indicators are too sensitive to the measured feedwater temperature to use as performance monitoring purpose. For example, measurement error of 0.5°C occurred at feedwater pump suction or discharge temperature result in 3.5% error on the feedwater pump efficiency and 4.3% error on the FWPT efficiency.

5.9.1. The feedwater pump turbine driving steam flow (as a percent of steam generator outlet flow)

The measured FWPT driving steam flow is corrected for LP turbine exhaust pressure which is different from the reference (design) heat balance conditions. If this exhaust pressure is higher than the reference value, the FWPT driving steam flow will be increased due to the reduced available energy of FWPT, and vice versa as shown in Eqs (84, 85):

$$w_{fwpt,corr} = w_{fwpt,meas} \times \frac{AE_T}{AE_D} \tag{84}$$

$$\%w_{fwpt} = \frac{w_{fwpt,corr}}{w_{sgout}} \cdot 100 \tag{85}$$

Where $w_{fwpt,corr}$ is the corrected FWPT driving steam flow (kg/h), $w_{fwpt,meas}$ is the measured FWPT driving steam flow (kg/h), AE_T is the FWPT available energy base on measured turbine exhaust pressure and AE_D is the FWPT available energy base on reference (design) turbine exhaust pressure. And $\%w_{fwpt}$ is the corrected FWPT driving steam flow rate in a percent of SG outlet steam flow (%) and w_{sgout} is the SG outlet steam flow(kg/h).

5.9.2. Feedwater pump discharge pressure vs RPM

Discharge pressure vs. RPM of the feedwater pump and booster pump (if it exists) are also important performance indicators which are used to detect pump damage or system restriction.

5.9.3. Diagnostic approaches

Monitoring of the key performance parameters and operating parameters, such as FWPT driving steam flow rate, feedwater and booster pump discharge pressures, provides a starting point to discuss and derive possible root causes of feedwater pumping system problems and corrective actions. Table 21 shows typical feature of change in these parameters.

TABLE 21. DIAGNOSTIC APPROACHES FOR FEEDWATER PUMPING SYSTEM

Possible cause of feedwater pumping system problems	Driving steam flow	Feedwater pump discharge press.***	Booster pump discharge press.
Booster pump impellor worn or damaged Booster pump wear rings worn	→ or ↗	→	↘
Feedwater pump impellor worn or damaged Feedwater pump wear rings worn	↗	↘	→
Flow restrictions on feedwater line Malfunction of FWPT admission valve	↗	→	→
Excessive FWPT blades worn or damage Excessive leaks in driving steam supply line and/or FWPT admission valve seat drain	↗	→	→
Excessive leakage on the feedwater pump minimum flow line	↗	↘	↘

*** Variation of the feedwater pump relative to its RPM

5.10. SUMMARY

This section introduced KPIs of NPP turbine cycle components. Thermal performance of the NPP is established through a combination of these indicators. Accordingly, any abnormal change of these indicators is advised to be monitored and the root cause identified and corrected.

Monitoring of KPIs at constant RTP is relatively easy because most of indicators are maintained constant from a benchmark. In case of a load cycling unit, these indicators may need to be compared with the expected values from the plant thermodynamic modelling or the empirical values at the same SG thermal power.

6. RECOVERY OR IMPROVEMENT OF ELECTRICAL POWER OUTPUT

6.1. REPAIR, RECOVERY AND OPTIMIZATION

Thermal performance optimization of the NPP eventually means maintaining the performance at a level consistent to when the unit is in a new and clean condition. However, performance deterioration of the turbine cycle components and resultant energy losses during lifetime of the unit are inevitable. Some NPPs in Europe have a history of load cycling operation. However, most of NPPs are operated at base load with constant RTP. At these NPPs, performance deterioration or energy losses directly result with reduction of the allowable electrical power output.

This section introduces typical sources of performance losses in the nuclear turbine cycle and corrective measures to recover the electrical power output. Operation changes to optimize the turbine cycle performance are also discussed in this section.

6.1.1. Overestimation of the reactor thermal power

As discussed in Section 4 in this publication, many NPPs are suffering from the RTP measurement drift which directly affect the generator power output. Many field performance diagnostic experiences and the case study show that overestimation of the RTP is the biggest contributor to the electrical power losses in NPPs.

In PWR and CANDU units, the RTP is calculated using secondary side SG steam properties instead of directly observing the nuclear instruments. This is due to their relatively high measurement uncertainty. The SG thermal power, which is called the secondary thermal power as expressed in Figure 100, is the primary input to calculate the RTP. Overestimation of the RTP mostly occurs due to measurement error on the SG thermal power.

$$SGMW_{thermal} = \frac{m_{SGOUT} \times (h_{SG\,OUT} - h_{FFW}) + m_{bf} \times (h_{bf} - h_{FFW})}{3600 \times 1000}$$

$$m_{SGOUT} = m_{FFW} - m_{bf}$$

Where,

$SGMW_{thermal}$ = Steam Generator Thermal Output, MW
m_{FFW} = Steam Generator Inlet Final Feedwater Flow, kg/h
m_{SGOUT} = Steam Generator Outlet Steam Feedwater Flow, kg/h
m_{bf} = Steam Generator Blowdown Flow, kg/h
h_{FFW} = Steam Generator Inlet Final Feedwater Enthalpy, kJ/kg
h_{SGOUT} = Steam Generator Outlet Steam Feedwater Enthalpy, kJ/kg
h_{bf} = Steam Generator Blowdown Enthalpy, kJ/kg
3600×1000 = Unit Conversion Factor, kJ/h to MW

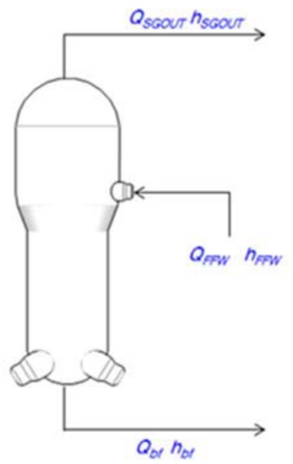


FIG. 100. Calculation of SG thermal power (PWR)

From the formula in Figure 100, the most sensitive and at the same time the most common source of measurement error is the SG inlet flow (final feedwater flow). The next is probably the final feedwater enthalpy which is predominantly decided from the measured feedwater temperature.

The importance of the final feedwater flow and temperature is exactly the same in BWR plants because these parameters are also direct input to calculate the RTP as shown in Section 4.1 of this publication.

6.1.1.1. Final feedwater flow measurement

Many NPPs use the ASME PTC 6 [3] throat tap flow nozzle in order to measure the SG (PWR and CANDU) or reactor (BWR) inlet final feedwater flow. This flow meter provides the best accuracy to measure the primary flow in the power industry and by this reason is widely used in the thermal performance acceptance test. Once the calibrated result meets the acceptance criteria specified in the code, the mass flow rate can be measured within ±0.25% measurement uncertainty.

The biggest advantage of this meter is that even if the calibration of the meter is performed at lower Reynolds number (typically less than 10 million), extrapolation of the flow coefficient up to the operating Reynolds number (typically 20 million ~ 45 million) is possible.

However, a critical disadvantage of this flow meter is that even a small change in physical geometry inside the meter significantly affects the accuracy of the flow measurements. Erosion, corrosion, corrosion product build-up and even a wrong cleaning method can alter the internal condition of the flow meter. Deposition of corrosion products in front of the throat, in the throat section, in the recovery cone of the throat tap flow meter and especially in the low pressure tap hole can increase the pressure drop across the meter.

This may eventually cause an erroneously high flow indication, which would be the root cause of overestimation of the RTP. In Figures 101 – 103 cases of deposit and erosion on the low pressure tap hole are shown.

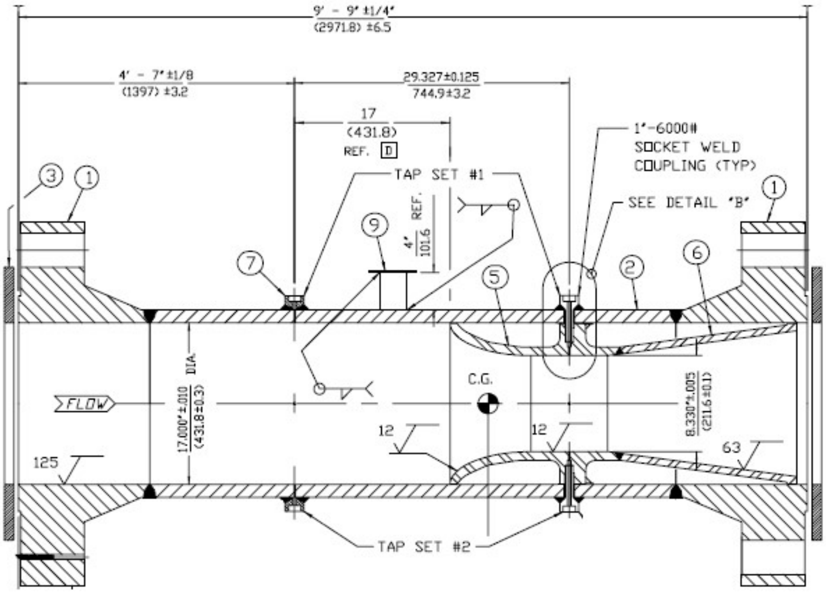


FIG. 101. Cross-section of typical ASME PTC 6 [3] throat tap nozzle for nuclear application

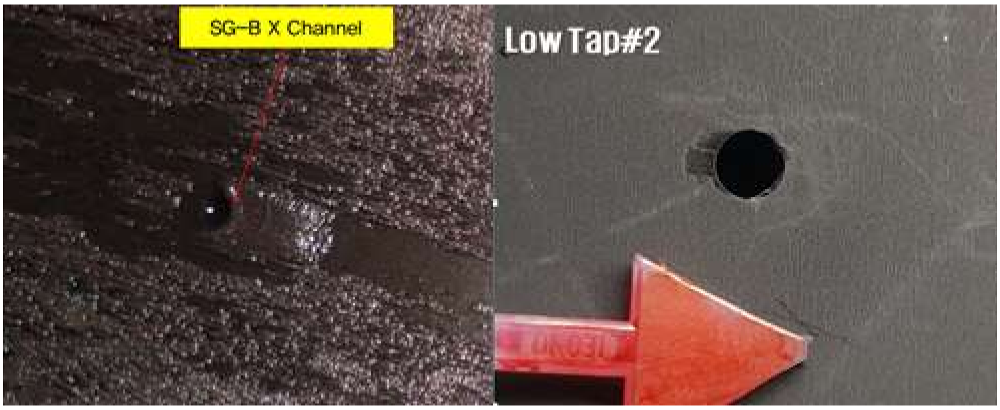


FIG. 102. Case of deposit (left) and erosion (right) on low pressure tap

(a) Corrective measures – Mechanical cleaning, chemical rinsing of the ASME throat tap nozzle, recalibration or replacement shown in Figure 103.

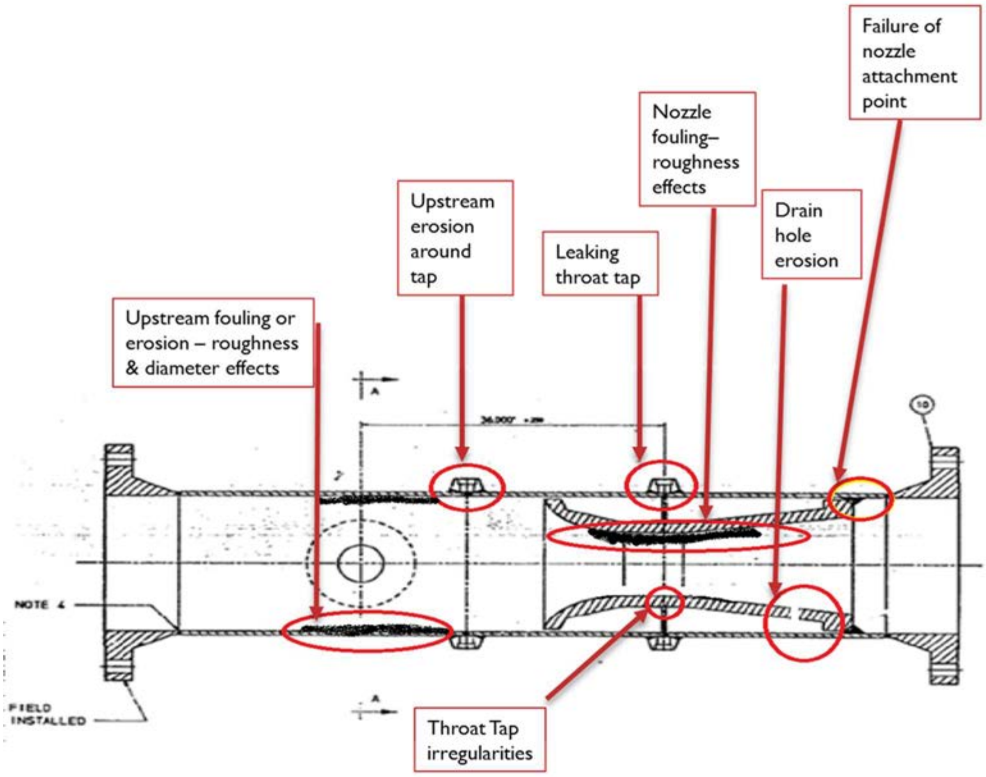


FIG. 103. Nozzle error modes [1]

Several preventive actions can be performed to avoid or at least to minimize fouling of this flow meter and the resulting the flow measurement errors.

Mechanical cleaning of the flow element during a planned outage by means of hydro lazing or chemical rinses can be performed to minimize the effects of fouling. Water treatment programmes can be also designed to reduce fouling. These programmes include the addition of morpholine in PWRs and hydrogen and zinc in BWRs.

Online flushing through the pressure sensing line is also effective to remove deposits near the low pressure tap hole in the early stage of deposit built up. This is done by using high pressure and temperature feedwater during operation.

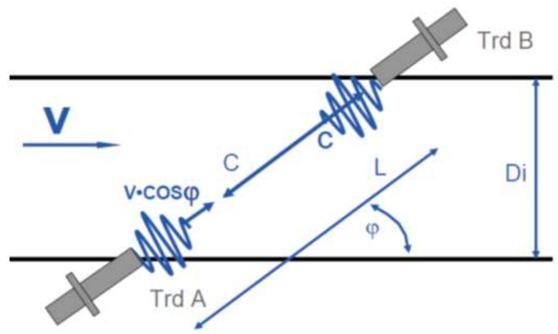
Despite every effort to prevent and correct fouling of the flow meter, it would not be possible to recover the flow coefficient characteristic against the Reynolds number to the new and clean condition. So, if the meter is accessible like in PWRs and CANDUs, the best way to maintain accuracy of the final feedwater flow measurement is to periodically calibrate the meter, use the new sets of flow coefficient to calculate the feedwater flow, or replace with a spare meter in case the calibration data shows the flow coefficient behaviour which does not satisfy the AMSE PTC 6 acceptance criteria. In many cases replacement would be beneficial if the economic losses caused by overestimation of the RTP are significant.

(b) Corrective measures – External transit-time UFM (temporarily installed)

As secondary flow meter, the external mount transit-time UFM, can be installed to detect fouling of the ASME throat tap flow nozzle and adjust RTP to correct the measurement drift.

The external transit-time UFMs determine the fluid velocity by measuring the transit times of pulses traveling with and against the fluid flow. The transducers are mounted on the outside surface of the pipe, and the acoustic paths are along the pipe diameter. External meters are non-invasive and do not require a custom spool piece. Installation of the meter can be performed in a week by trained technicians while the plant is in operation. The meter provides a continuous, fast response of the flow measurement. Figure 104 shows flow measuring principle of this meter.

The benefit of this meter is that even though the systematic uncertainty (bias error) of flow measurement is higher than the calibrated ASME PTC 6 [3] throat tap nozzle, the magnitude of bias error is maintained almost constant without causing fouling problems.



$$t_{ab} = \frac{L}{c + v \cdot \cos\varphi} \qquad t_{ba} = \frac{L}{c - v \cdot \cos\varphi}$$

$$v = \frac{L}{2 \cdot \cos\varphi}\left(\frac{1}{t_{ab}} - \frac{1}{t_{ba}}\right)$$

FIG. 104. Flow measuring principal of the transit-time UFM

In Figure 104, L is a distance between transducer (mm) installed under angle φ (°), *Di* is a inner tube diameter (mm), V is a flow velocity (m/s) and C is a speed of sound in the fluid at rest (m/s).

- Measurement of differences in transit times (A to B and B to A) can be used to calculate fluid velocity, which is the first step to determining volumetric flow.
- Solving the two simultaneous equations for ‘V’ gives the average fluid velocity encountered between transducers.
- Solving for ‘C’ gives the speed of sound in the fluid at rest.

(c) Corrective measures – Multipath transit-time UFM (permanently installed)

With development of the linear elastic fracture mechanics technology, a multipath transit-time UFM has been used for the final feedwater flow in the NPPs.

The multipath transit-time UFM measures the average fluid velocity along multiple chordal paths and combines the results to calculate the volumetric flow rate. This meter also makes a temperature measurement that is used with an input pressure to determine density and to convert the volumetric flow rate into a mass flow measurement.

This flow meter typically has much lower uncertainties than external transit time meters. State of the art technology reduced the flow measurement uncertainty up to ± 0.3%, but the meter requires installation of a costly, custom piping section.

The multipath transit-time UFM (as shown in Figure 105) can be used as the primary flow meter, taking over the role of ASME throat tap flow nozzle and directly calculating the reactor or SG thermal power. However, even in this case, a secondary flow meter is typically still mandatory as a back-up to the primary meter.

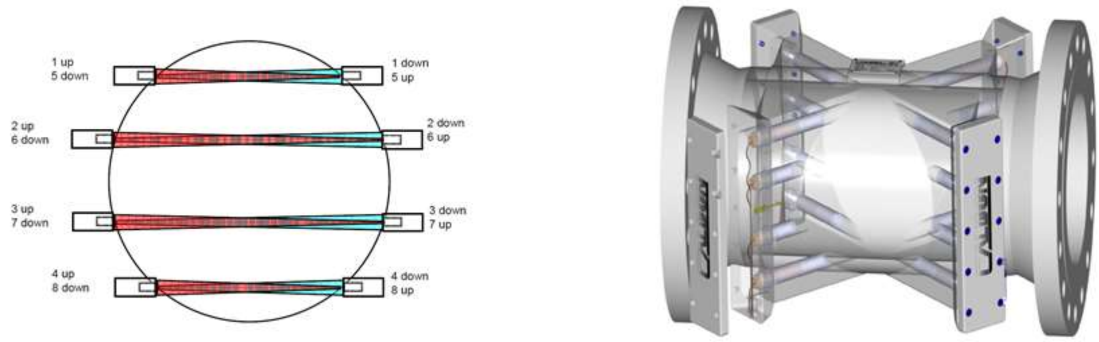


FIG. 105. Multipath transit-time UFM pipe section (courtesy of Caldon. Inc)

(d) Corrective measures – SG MW_{th} calculation with main steam flow

Measurement uncertainty of steam flow is relatively greater than that of the feedwater flow. The general flow equation for differential pressure flow meters, induced from the Bernoulli Principle and Continuity Equation, assumes incompressible fluid like feedwater. However, steam is a compressible fluid expanding while passing though the flow element. A thermal expansion factor for steam needs to be considered to calculate the flow rate. It causes additional uncertainty in flow measurement. This is the reason why the feedwater flow is used to calculate the reactor or SG thermal power. However, like the external mount transit-time UFM, plant flow meters installed to measure main steam flow can be used as the secondary flow meter, for example to detect fouling of the final feedwater flow meter and correct RTP measurement drift RTP.

The ASME wall tap nozzle is typically used to measure the main steam flow. Even though the flow measurement uncertainty is relatively higher with this meter its systematic uncertainty (bias error) is improved once the main steam flow is adjusted to a reference flow (feedwater meter). So, the adjusted main steam flow can be directly used to calculate the SG thermal power. The main advantages of this method are that fouling problems can be avoided with the steam flow measurement and no additional installation of hardware is required.

A correction factor to adjust the feedwater flow to the main steam flow is determined when the RTP has been reached at 100%. It also needs to be stabilized after a unit restart from the planned outage when the feedwater flow measurement is most accurate. During this time, the SG blowdown line needs to be isolated.

As the steam is a compressible fluid, the correction factor will be effective only at the same SG outlet steam pressure at which the steam flow meter is adjusted. Accordingly, this method can be applied only when the RTP is above 95%. In this method, reliability of the feedwater measurement is still very important and preventive and corrective measures for the feedwater flow meter are recommended to be performed in the same way.

Figures 106 – 107 show a case of operation change from the feedwater based (FW BSCAL) to steam flow based (MS BSCAL) SG thermal power calculation after detecting fouling of the feedwater flow meter, and resultant recovery of the electrical power output.

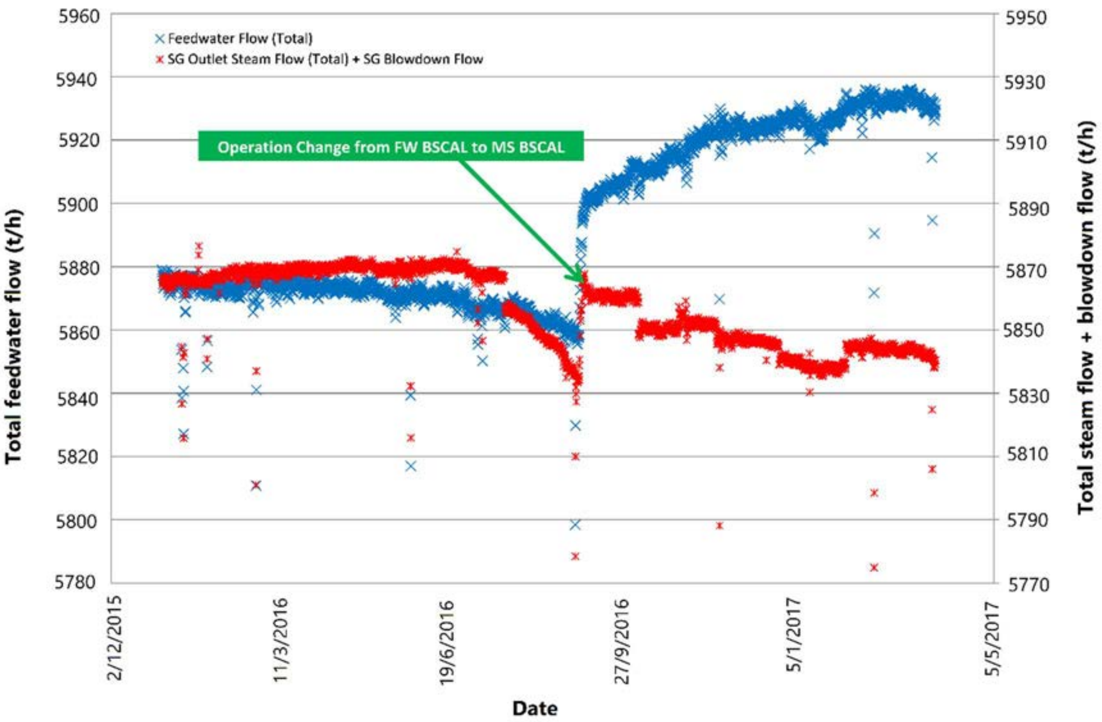


FIG. 106. Trend of FW and MS flows before and after operation change

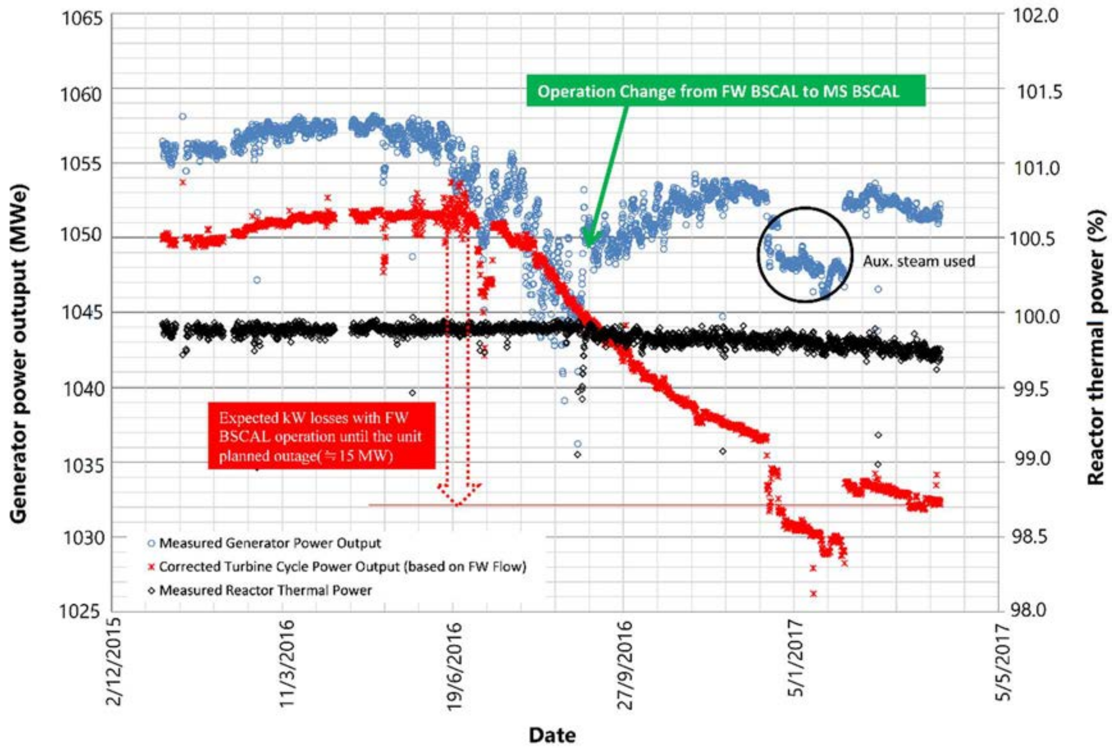


FIG. 107. Trend of electrical power output before and after operation change

6.1.1.2. *Final feedwater temperature measurement*

Even though its sensitivity to the reactor or SG thermal power is less than that of the final feedwater flow measurement, measurement error on the final feedwater temperature is also common contributor to the RTP measurement drift (see Figure 108).

Measurement error of 1 ℃ for the final feedwater flow will change the calculated reactor or SG thermal power by 0.3~0.4%. that is, if the final feedwater temperature is measured by 1 ℃ lower than the true value, the RTP will be overestimated by 0.3%~0.4%. This will reduce the electrical power output by 0.3~0.4%.

Accordingly, the final feedwater temperature measured at each inlet of the SG (or reactor in BWR) is always recommended to be monitored and compared with the top heater outlet temperature to confirm its reliability.

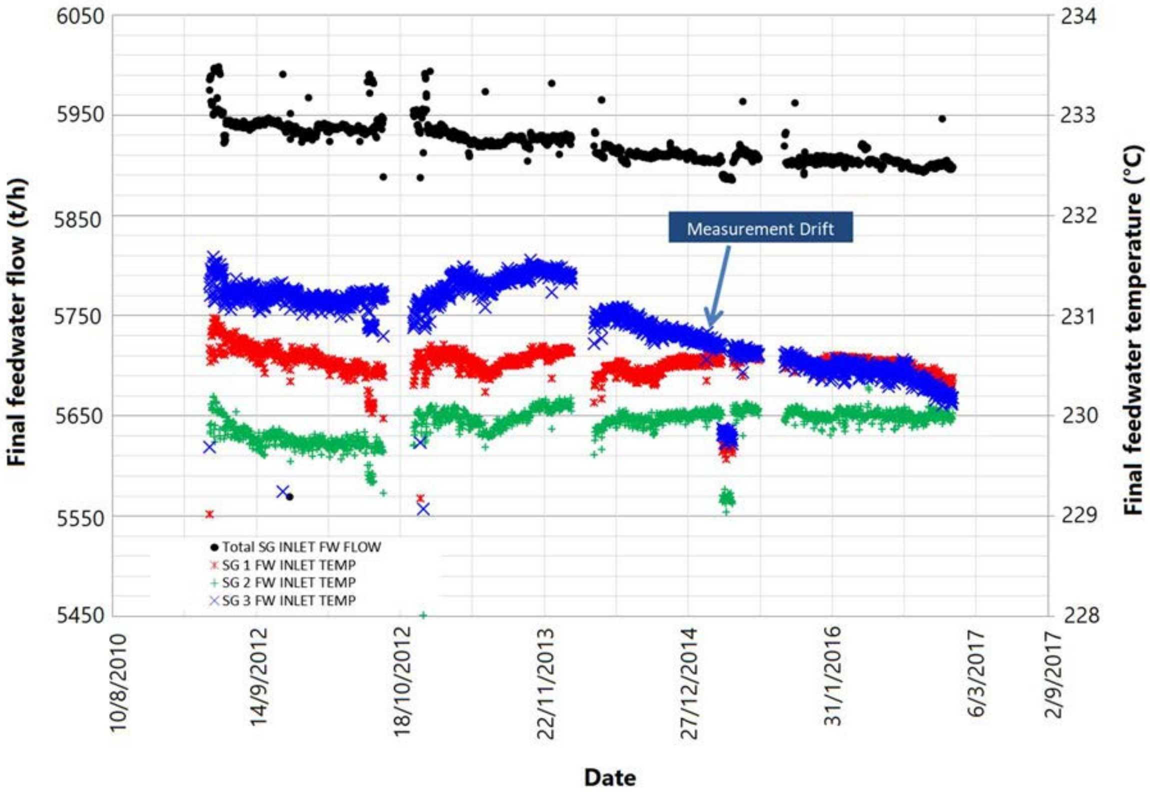


FIG. 108. Monitoring of final feedwater measurement drift

6.1.2. Cycle isolation losses

The cycle steam bypassing the HP and/or LP turbine directly reduces the mass flow rate of steam to work in the turbine steam path (to rotate the turbine rotor). In general, 1% of throttle (main) steam bypassing the steam turbine reduces the electrical power output by 1%.

Any turbine extraction steam, condensate or feedwater flow bypassing its design destination like feedwater heater or FWPT also reduces the electrical power output. This is because eventually theses bypassing flows will increase the turbine extraction steam flow rate. The impact of these energy losses (cycle isolation losses) on the electrical power output can be evaluated through simulation of the plant thermodynamic modelling if their locations and mass flow rates are quantified.

Unfortunately, it is not easy to identify these cycle losses because the valve or steam trap leakage at a drain line to the condenser cannot be seen. If the leakage is to the atmosphere via a vent or relief valve, it would be visible and therefore can be easily identified and repaired. The valve or steam trap leakages to the condenser can be undetected for a significant amount of time unless preventive or corrective actions are implemented to identify and track these leakages. Accordingly, in order to optimize the turbine cycle performance, it is required to establish a cycle isolation programme as explained in Section 3.4 of this publication. It is necessary to periodically monitor valves and steam traps for leakage and malfunctioning.

Infrared (IR) thermometers and acoustic meters are used for the cycle isolation check. Generally, an IR thermometer is used to measure the bare pipe temperature downstream of the suspected valve after making a ~50 mm hole through the aluminium cladding of the pipe. However, in case of a drain line with a bypass valve as shown in Figure 110, an acoustic meter is also required to identify whether the valve leaks through the main line or bypass line. The acoustic meter is a very useful tool to detect leakage on valves where upstream and downstream temperatures are almost same such as the feedwater pump minimum flow line. A detection example is shown in Figures 110 – 111 with the tools for cycle isolation check shown in Figure 109.

FIG. 109. Tools for cycle isolation check – Infrared thermometer (left); acoustic meter (right)

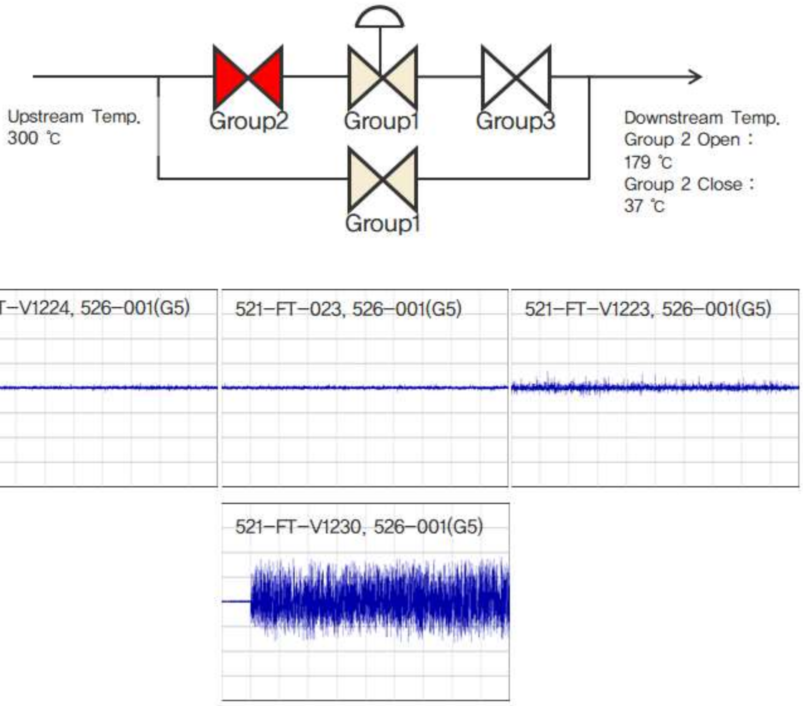


FIG. 110. Check for valve leakage at drain valve train

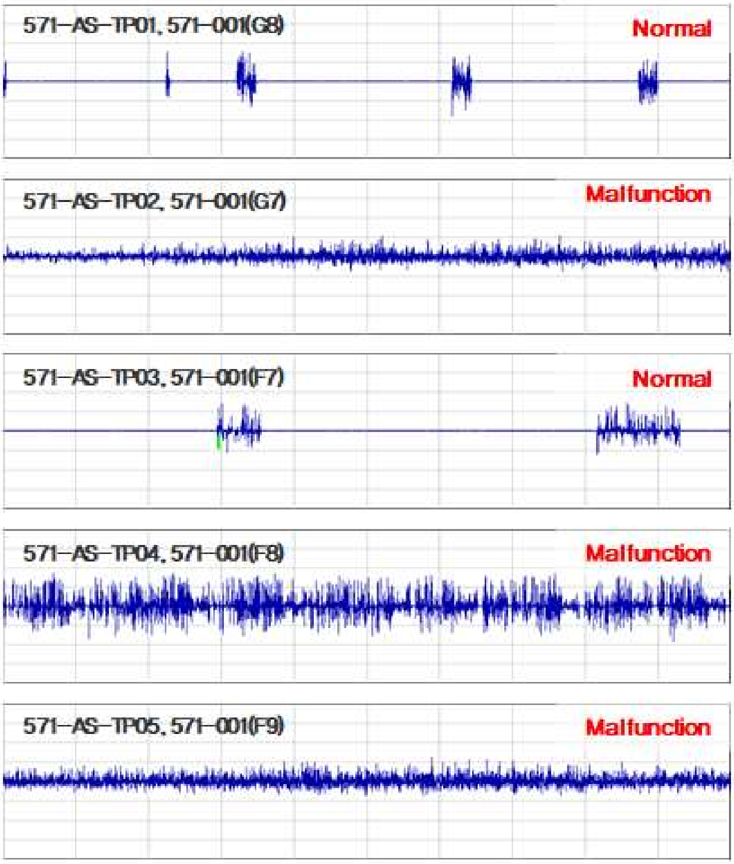


FIG. 111. Check for steam trap malfunctioning

6.1.3. Problem solving of turbine cycle components

Performance deterioration of turbine cycle components in the NPP is much less than that of the fossil plant. However, problems on the NPP turbine cycle components obviously occur, and accordingly performance indicators of key components as discussed in Section 5 need to be continuously or at least periodically monitored.

Any abnormal change in turbine cycle component performance indicators needs to be traced for determination of the root cause and corrective actions taken to recover the loss. In [29] detailed guidelines for these activities are provided.

The Δ kW in Table 22 is obtained from simulation of the thermodynamic modelling of a 1000 MW rating PWR unit. This table can be used to estimate the general impact of key component performance indicators on the electrical power output. The Δ kW in this table will be applicable for the BWR and CANDU plants with similar configurations.

TABLE 22. IMPACT OF KEY COMPONENT PERFORMANCE ON THE ELECTRICAL OUTPUT

Performance parameters	Units	Reference	Current	Δ Change	Δ kW
Steam turbine					
HP turbine steam path efficiency	%	85.00	85.85	0.85	2978
LP turbine steam path efficiency (base ELEP)	%	90.00	90.90	0.90	6839

TABLE 22. IMPACT OF KEY COMPONENT PERFORMANCE ON THE ELECTRICAL OUTPUT (cont.)

Performance parameters		Units	Reference	Current	Δ Change	Δ kW
MSRs						
Moisture separator effectiveness		%	97.00	98.00	1.00	311
LP reheater TTD		°C	12.00	11.00	-1.00	77
HP reheater TTD		°C	12.00	11.00	-1.00	156
Cycle steam ΔP		%	3.00	2.00	-1.00	1063
Feedwater heaters						
Feedwater heater 7	TTD	°C	2.70	1.70	-1.00	413
	DCA	°C	5.60	3.60	-2.00	64
Feedwater heater 6	TTD	°C	2.70	1.70	-1.00	245
	DCA	°C	5.60	3.60	-2.00	84
Feedwater heater 5	TTD	°C	2.70	1.70	-1.00	199
	DCA	°C	5.60	3.60	-2.00	318
Feedwater heater 3	TTD	°C	2.70	1.70	-1.00	145
	DCA	°C	5.60	3.60	-2.00	10
Feedwater heater 2	TTD	°C	2.70	1.70	-1.00	145
	DCA	°C	5.60	3.60	-2.00	50
Feedwater heater 1	TTD	°C	2.70	1.70	-1.00	249
	DCA	°C	5.60	3.60	-2.00	35
Feedwater pumping system (FWPT and FWP)						
FWPT driving steam flow in % of SG outlet steam flow		%	0.400	0.300	-0.10	987
Condenser						
Condenser pressure		mmHg	32.00	29.00	-3.00	588.04
		mmHg	34.00	31.00	-3.00	776.68
		mmHg	36.00	33.00	-3.00	971.27
		mmHg	38.00	35.00	-3.00	1168.47
		mmHg	40.00	37.00	-3.00	1365.33
		mmHg	42.00	39.00	-3.00	1559.24
		mmHg	44.00	41.00	-3.00	1747.94
		mmHg	46.00	43.00	-3.00	1929.48

6.1.4. Reduced steam generator outlet pressure

Changes in SG outlet pressure affect the turbine cycle performance in several ways. Intuitive understanding of steam properties may give a hint that the reduced SG outlet pressure reduces the available energy delivered to the steam turbine. But the actual behaviour in the nuclear turbine cycle, which is operating in the wet steam region, is somewhat different.

With fixed SG outlet moisture content in the SG thermal power calculation, the reduced steam pressure increases the calculated steam enthalpy. It also decreases the steam flow with increase of delta enthalpy across the SG. The reduced steam pressure reduces the MSR outlet (LP turbine inlet) steam temperature, while in this case increasing the throttle steam flow to the HP turbine. This is due to decreased heating steam flow supply to the reheater.

Throughout a combination of these cycle effects, if the unit is operated at constant SG thermal power, a 1% reduction steam pressure will cause approximately 0.1% or even less reduction in the electrical power output due to reduced throttling losses at the turbine control valve. The reduction in electrical power is compensated with increase of the specific volume of the throttle steam. Lower pressure causes an increase in volumetric flow of the steam at the HP turbine inlet. As a result of an increase in the turbine control valve position, throttling losses at the valve will be reduced. Meanwhile, the impact of the reduced steam pressure on the electrical power output will be much higher for units that operating in load cycling modes.

In a PWR, the HP turbine control valve positions are adjusted to maintain 100% power at a specified T_{avg} and SG outlet pressure. In a BWR, the reactor dome pressure is fixed and the turbine control valves vary to maintain the specified dome pressure. Lower dome pressure will cause the turbine control valve(s) to throttle down in order to cause dome pressure to rise back to its set point. This will result in a reduced throttle flow, reactor power, and unit output. To compensate, the operator will raise reactor recirculation flows or withdraw control rods to maintain 100% RTP power. Conversely, higher dome pressure will cause the control valve to throttle open, increasing mass flow through the HP turbine and requiring the operator to reduce recirculation flow or insert control rods.

A reduction in SG outlet pressure may occur for the following reasons:

- Fouling of the inside or outside of the tubes;
- Tube plugging;
- T_{avg} reduction;
- Power uprates.

The reduced SG outlet pressure has insignificant impact on the electrical power output before it consumes the 100% flow capacity of the turbine. However, if the HP turbine control valves are wide open, further reduction of SG outlet pressure will limit the RTP. That drastically reduces the maximum electrical power output capacity of the unit. Compensating operational changes may be required in this case.

6.1.4.1. SG Fouling

Figure 112 below describes the relationship between the SG pressure and the overall fouling factor.

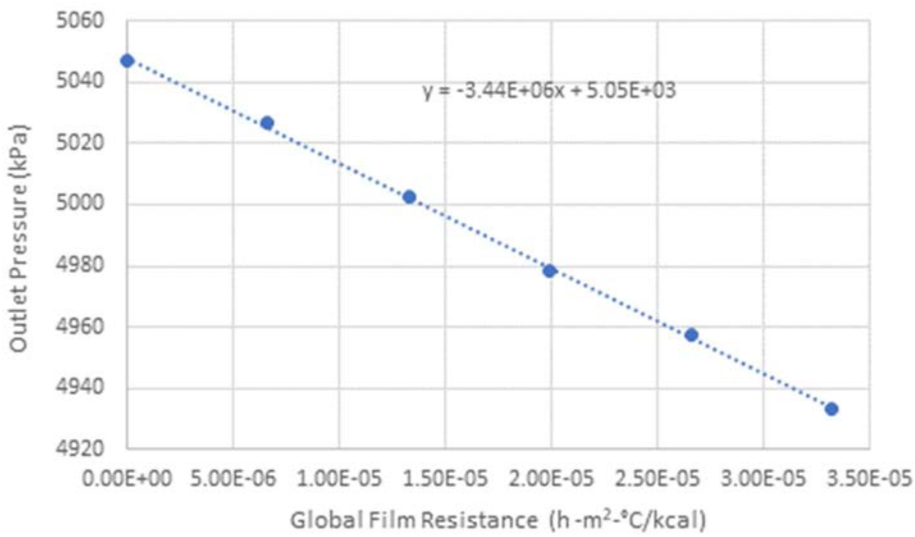


FIG. 112. Effect of fouling on SG pressure

In some cases, fouling and roughness on the outside of the tubes may be beneficial for effective heat transfer due to the increased area and nucleate boiling sites. Depending on the actual SG material and tube configuration steam pressure has shown a decrease after SG cleaning. This occurs due to a lack of nucleate boiling sites and a reduction in outside tube surface area. The restoration of a suitable surface can take years under modern water chemistry controls.

In many plants the steam pressure will decrease after an outage due to a change in the tube oxide layer (see Figures 113 – 114). The pressure will recover over time after the re-establishment of nucleate boiling sites with the build-up of the oxide layer.

FIG. 113. Steam generator pressure changes

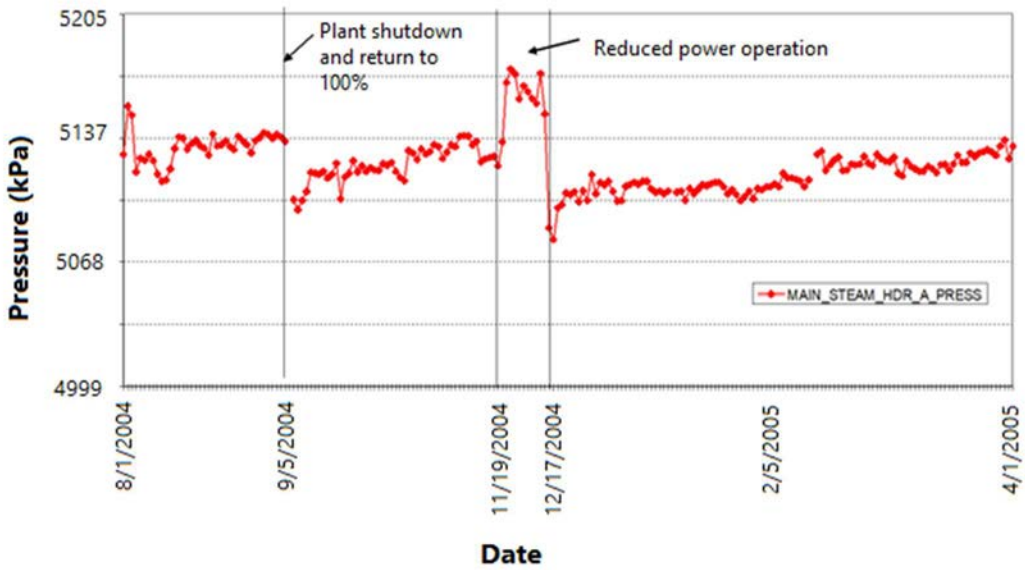


FIG. 114. *Steam header* pressure (PWR)

6.1.4.2. Tube plugging

Figure 115 below shows the effect of SG tube plugging on SG outlet pressure. It is shown to emphasize that the heat transfer area in fouling factor calculations needs to be corrected for plugged tubes.

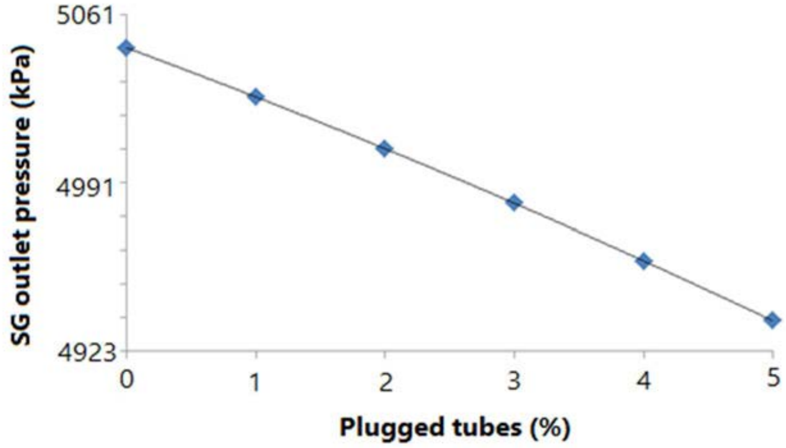


FIG. 115. Effect of SG tube plugging on SG outlet pressure

6.1.4.3. T_{avg} reduction

SG outlet pressure decreases as the primary coolant T_{avg} decreases. T_{avg} may be reduced in the process of reducing T_{hot} or T_{cold} depending on the type of SG. Primary temperature is sometimes lowered in order to keep the tube metal temperature below a critical threshold temperature where corrosion and subsequent tube leaks are more likely to occur. More information on primary loop temperature reduction is available in numerous EPRI papers, which have been written on this subject.

Since T_{avg} is simply the average of T_{hot} and T_{cold}, the calculated value can change due to problems with the T_{hot} or T_{cold} temperature measurement. The bulk mean T_{hot} is difficult to measure due to stratification of the hot water leaving the reactor core ('hot leg streaming'), large diameter of the pipe and natural convective circulation of the flow entering the SG.

T_{hot} is sometimes measured in a 'resistance temperature detector bypass loop'. This condition is also affected by the power distribution in the reactor, which changes over core life. In this method, part of the water is extracted from the main pipe and bypassed through a smaller pipe, which contains the T_{hot} measuring instruments. Some concerns have been raised recently about this technique and many plants have relocated their T_{hot} sensors to the main pipes where they directly measure T_{hot}.

This change in measurement location may yield a different indication of T_{hot} and result in changes in calculated T_{avg} even if the true T_{hot} has not changed. This change in indicated T_{avg} (see Figure 117) may result in a change in the T_{avg} control point and result in a change in SG outlet pressure (see Figure 116). Some plants have investigated the use of ultrasonic methods to more accurately measure T_{hot} which can better determine the bulk temperature.

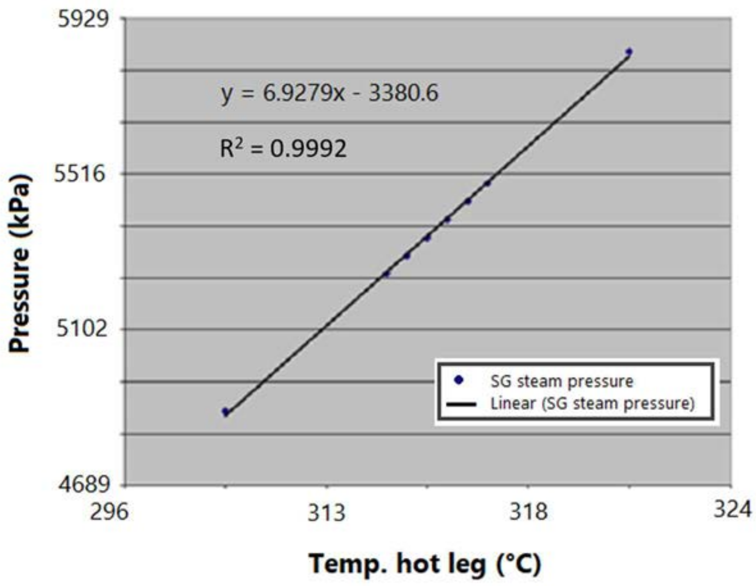


FIG. 116. SG outlet steam pressure vs primary hot leg temperature

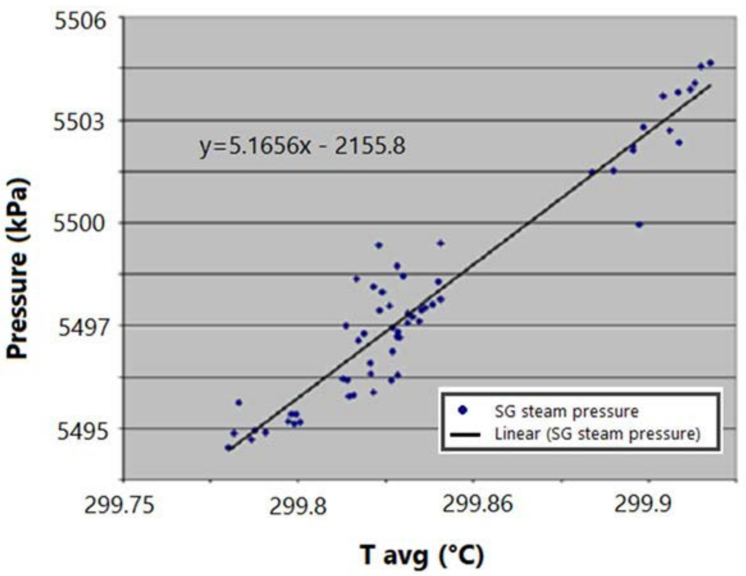


FIG. 117. T_{avg} vs SG outlet steam pressure

6.1.4.4. Compensative operation methods for the reduced RTP

Figure 118 below shows the relationship between primary average temperature T_{avg} and the equivalent throttle flow ratio (ETFR). The ETFR is the % flow passing capability of the turbine governor (control) valves.

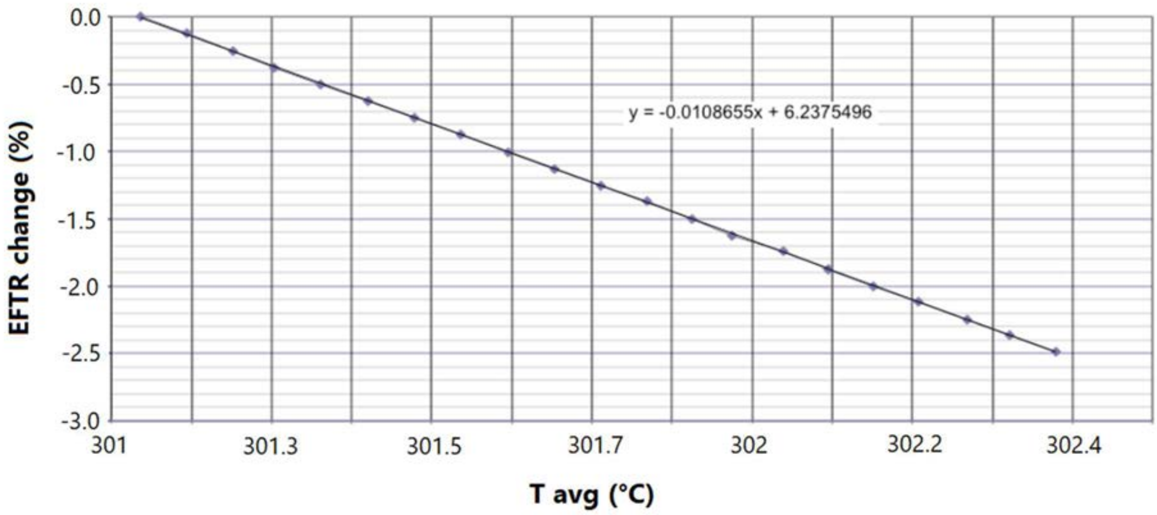


FIG. 118. Relationships between T_{avg} and ETFR

Continuously operating the plant at 100% RTP requires the capability of the turbine cycle to receive the energy produced by the reactor and the steam flow produced by the SGs. The mass flow rate produced by the SGs is a function of the enthalpy rise across the SGs at a constant 100% reactor power. The ability of the turbine cycle to receive the mass flow depends on the density of the steam and the total volumetric flow rate into the HP turbine control valves.

Once the volumetric flow reaches the point where all HP turbine control valves are wide open (VWO), any further decrease in density would require a corresponding decrease in mass flow rate into the HP turbine.

If the extractions upstream of the HP turbine remained constant, the main steam mass flow rate from the SGs would need to be reduced by way of reducing reactor power output. Figure 119 below illustrates this point.

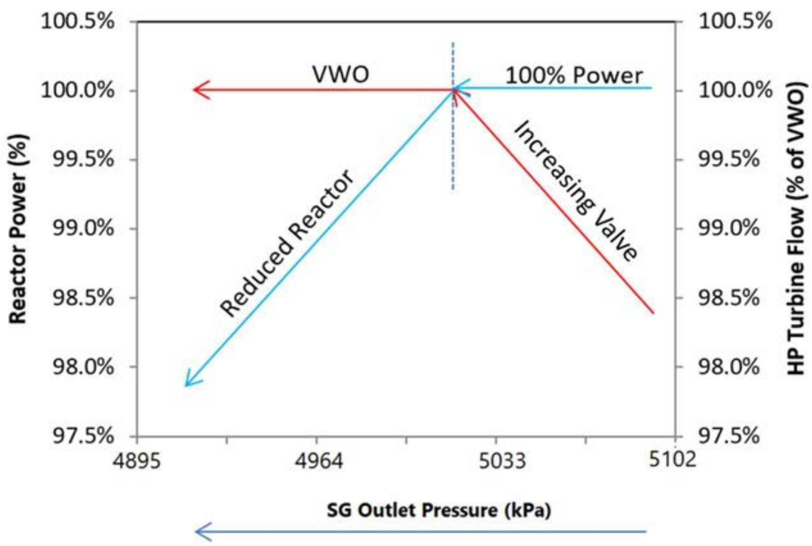


FIG. 119. Limitation of RTP due to SG pressure reductions

The reduction in RTP described above could be avoided if the volumetric flow rate into the HP turbine could be reduced or maintained constant. This can be done either by reducing the mass flow rate into the turbine or by increasing the density of the steam entering the turbine. Some of these changes could be achieved by changes in operation of existing equipment while others may require changes in design.

The following are some possible methods to compensate for the reduced RTP. Most of these methods are aimed at obtaining the turbine flow capacity margin with sacrifice of the turbine cycle performance. Alternatively, reduced RTP will be recovered, which will eventually increase the electrical power output.

(a) Evaluate accuracy of existing T_{cold}, T_{avg} and T_{hot} values

Figure 120 shows that two of the four SG T_{avg} for the example unit varied over time while the other two stayed essentially constant. Since T_{avg} was probably controlled using Boron dilution during these fuel cycles, the change in T_{avg} is supposed to be consistent for all four SGs. The variations may be due to instrumentation problems, sensor locations, flow patterns leaving the reactor vessel or SGs, or control system averaging and auctioneering methods. Currently the example plant operation is limited based on the highest recorded T_{avg} indication. The actual T_{avg} indication is lower and thus the overall primary temperature is lower resulting in the lower steam pressure.

An imbalance in the measured hot leg temperature can have two causes. Either the difference is due to an imbalance in the SG power or it is due to the thermo-hydraulic effect known as streaming. Also, temperature stratification in the hot legs, or layering, causes the fluid temperature not being uniform over a cross section of the hot leg. However, this latter effect is mitigated by use of sampling scoops providing an average hot leg temperature sample.

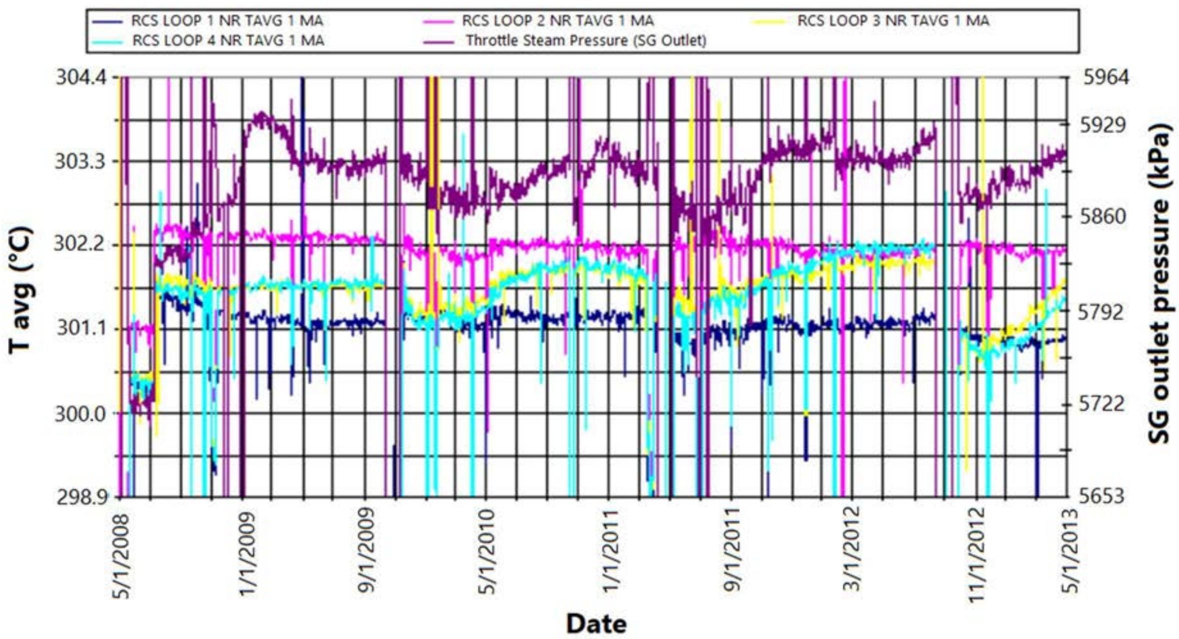


FIG. 120. Variation of T_{avg} over time

Outlet ports are provided in the scoops to direct the sampled fluid past the resistance temperature detectors. Therefore, this latter effect is not considered a significant or quantifiable effect, as compared to excessive loop streaming.

The reactor coolant temperature performance is graphed in Figure 121. The graph includes the four loop T_{avg} values, T_{ref}, auctioneered high T_{avg} (Loop 22) and average reactor coolant temperature. T_{cold} loop trends are graphed in Figure 122 and T_{hot} trends are shown in Figure 123. The trends clearly show the high degree of streaming with Loop 22 consistently having the highest temperature streaming of the four loops.

Figure 124 provides an example of the relationship between the shifts in temperature and SG pressure. While not all SG pressure changes correspond to changes in temperature, due to other conditions affecting SG pressure, it is clear that the steam pressure is a function of the changes in temperature due to streaming as described above. The streaming, in conjunction with the auctioneering high average temperature control, results in a higher indexed temperature than actual reactor coolant average temperature and causes operations to control the plant at a lower steam pressure. The magnitude of the streaming is determined below by plant data trending.

To change to average temperature T_{avg} control from auctioneered high T_{avg} to average T_{avg} would provide some additional operating margin. It has the added benefit of not requiring a change to the technical specification limit. This option would include the following:

- Safety analysis for the effects of using average T_{avg} vs high average temperature auctioneered T_{avg} on reactor control rod control, pressurizer level control and steam dump control.
- Fuels evaluations on fuel rod design, core design and thermal hydraulics.
- Review of the control system failure analysis.

This control conversion provides a more accurate determination of the reactor coolant average temperature, and thus removes the current penalty associated with high streaming.

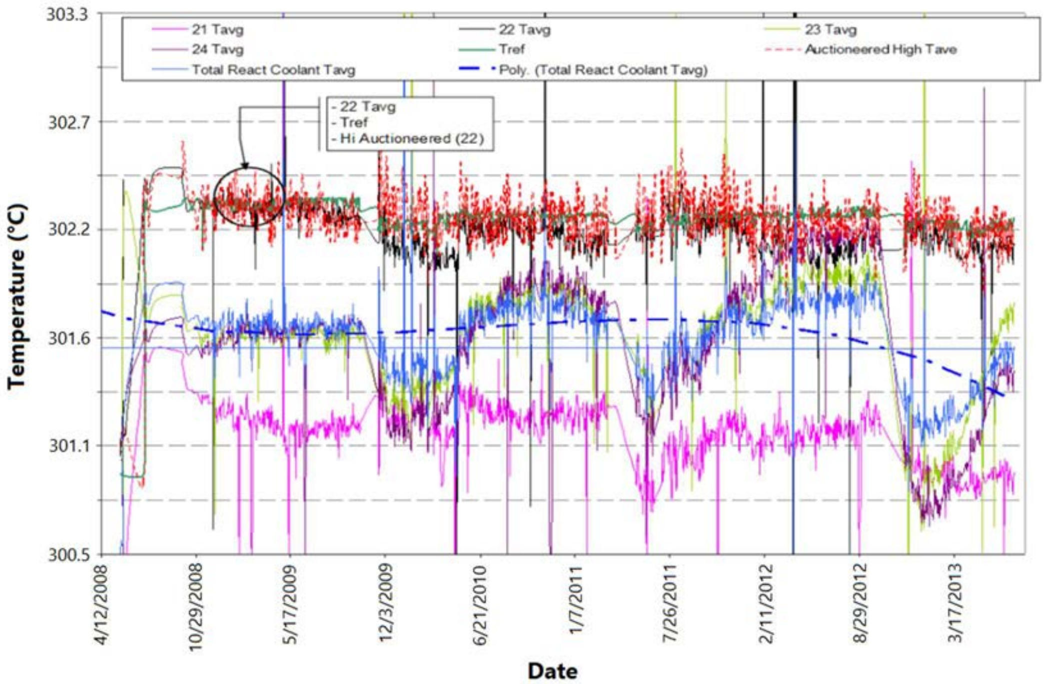


FIG. 121. T_{avg} streaming

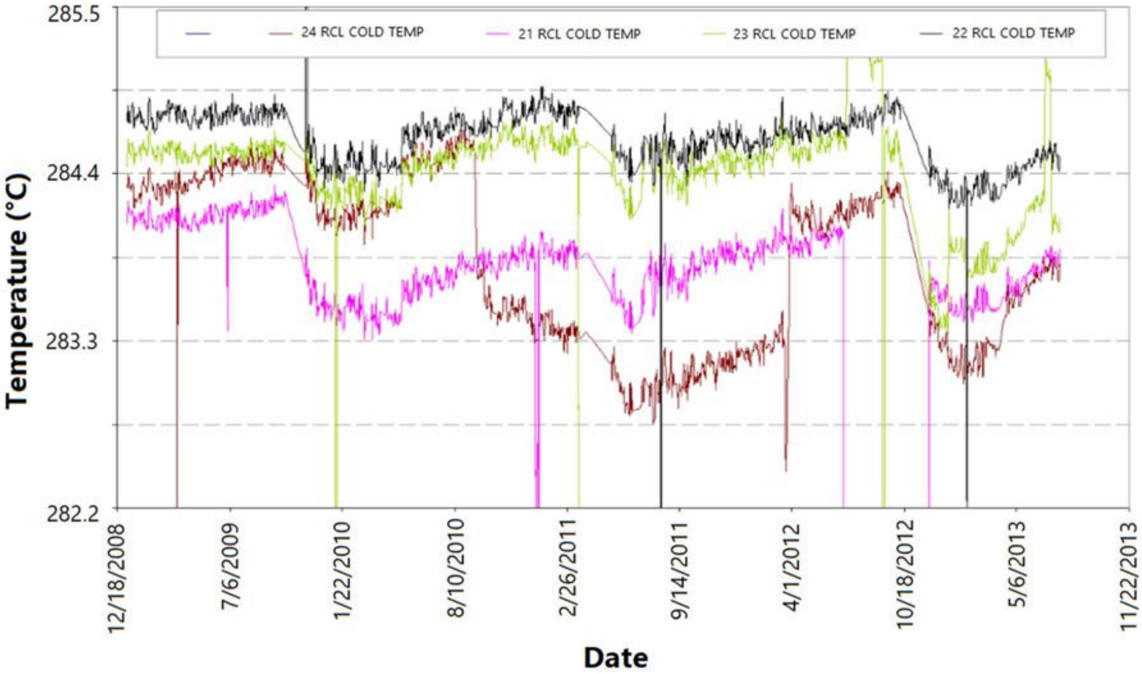


FIG. 122. Tcold streaming

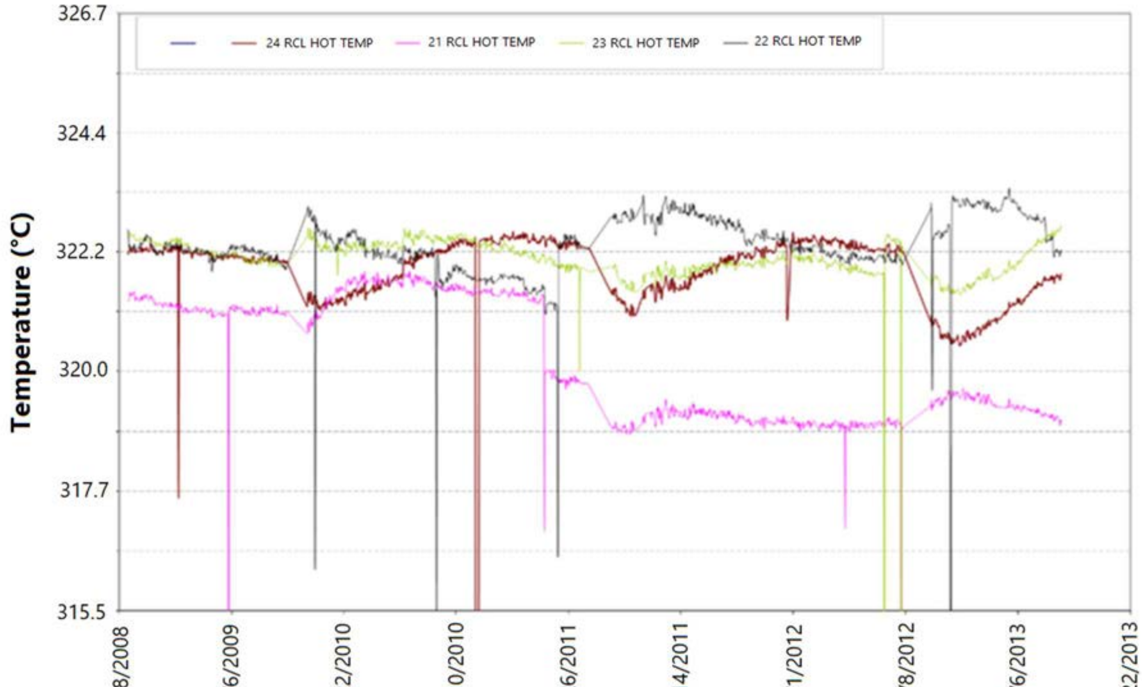


FIG. 123. T_{hot} streaming

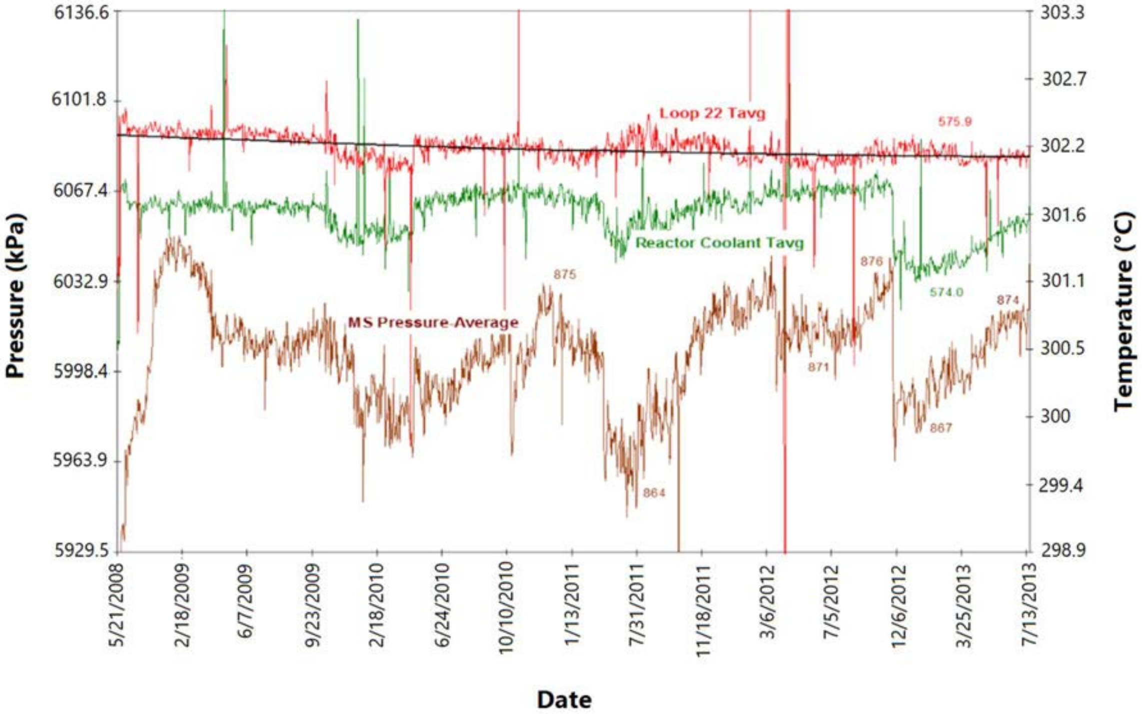


FIG. 124. MS pressure impact

(b) Increase MSR (HP reheater) excessive steam vent flows (4-pass tube arrangement)

The heating steam supply to the second stage reheaters comes from upstream of the HP turbine throttle valves. This flow includes excess steam used to ventilate the tubes and prevent condensate from accumulating in the tubes. The excess steam vent flow is regulated using throttling valves to obtain flow rate required to achieve a certain amount of subcooling.

Although normally adjusted to obtain a slight amount of subcooling, this valve could be utilized to allow greater excess amounts of steam to pass through the reheater and out to the HP feedwater heater. This excess steam flow would increase the total extraction flow to the second stage reheaters and effectively bypass the HP turbine. The excess scavenging steam normally goes to the No. 6 heaters. The energy in this excess flow would assist in the heat transfer to the feedwater in the No. 6 heaters and reduce the extraction flow required from the HP turbine extraction to the heaters.

The effect on MSR reheater performance is expected to be minimal, with only the previous heat transfer for subcooling the reheater condensate from saturated liquid to subcooled liquid being lost. There would be an increase in velocity through the tubes and possible heater drain tank issues which would require further study.

The typical excessive scavenging steam flow rate for a Westinghouse MSR reheater is 2%. Using the heat balance as a reference, the normal second stage reheater excessive scavenging steam flow would be 0.11% of throttle flow. If this excessive scavenging flow was increased to 10%, the flow would increase to 0.53% of throttle flow. This would represent a 0.42% increase in extraction flow to the second stage reheater and reduce throttle mass flow rate by 0.42%.

(c) Throttle first stage reheater flows to cause the second stage reheaters to increase extraction flows

Another way to increase flow to the second stage reheater is to reduce the amount of heat transfer that occurs in the upstream first stage reheater. This could be performed by the throttling of HP turbine extraction steam flow to the first stage reheater. The reduction in flow supplied to the first stage reheater would increase the TTD and lower the enthalpy of the steam leaving the first stage reheater. This would increase the load on the second stage reheater and result in an increase in flow to the second stage reheater.

Figure 125 shows the effect of reducing first stage reheater extraction steam flow on the HP ETFR. ETFR is the ratio of actual throttle flow (when corrected to design throttle pressures), divided by the design throttle flow at VWO. An ETFR of 1.0 or 100% would represent VWO. Numbers below 1.0 or 100% indicate the available flow margin before the unit would reach VWO.

The Figure 125 also shows the potential change in ETFR that would result from various first stage tube side flows. As shown, changing the extraction flow from 195 045 kg/h to 68 039 kg/h would reduce the ETFR by approximately 2%.

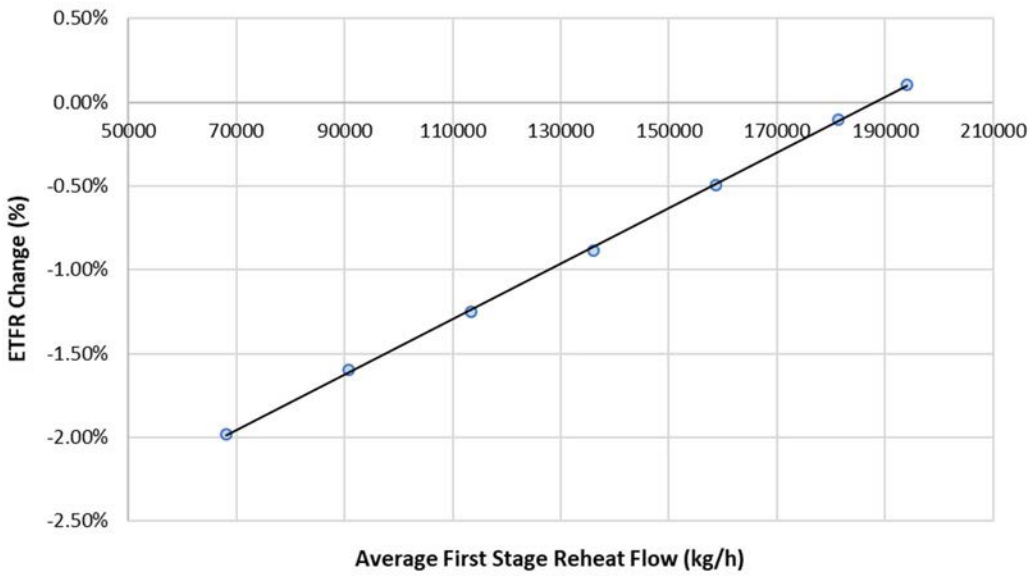


FIG. 125. First stage reheater flow vs ETFR

(d) Throttle extraction steam to top heaters

Another way to reduce mass flow into the HP turbine is to increase the enthalpy rise across the SGs by reducing the final feedwater temperature. Increasing the enthalpy rise across the SGs would require a corresponding decrease in mass flow rate in order to maintain the 100% RTP power. The enthalpy of the feedwater entering the SGs can be reduced by lowering the final feedwater temperature. This could be done by throttling the extraction flow to the top (highest pressure) heaters, thus increasing the pressure drop in the extraction line and lowering the pressure in the heaters. Since the TTD of the heaters would remain essentially constant, the final feedwater temperature would be reduced by approximately the same amount as the drop in saturated temperature at the new lower heater shell pressures.

Figure 126 shows the predicted effect of increasing extraction line pressure drops to the top heaters on the HP turbine ETFR, as simulated using PEPSE. As shown increases of pressure drop from 5% to 12% would reduce ETFR by approximately 1%.

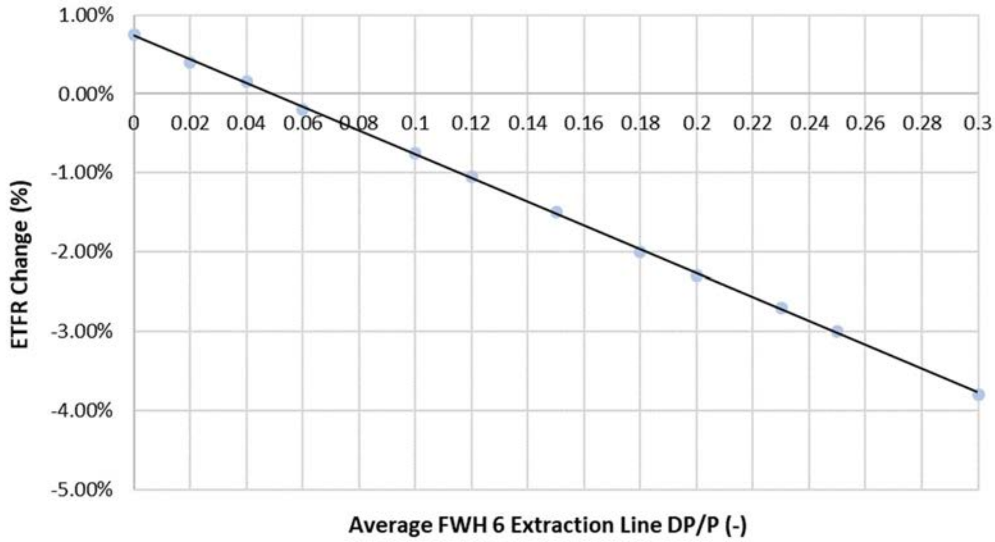


FIG. 126. Throttle extraction steam vs ETFR

(e) Bypass top FW heaters to lower FW temperature, increase SG enthalpy rise and reduce overall cycle flow.

Another way to reduce final feedwater temperature, increase SG enthalpy rise, and reduce HP turbine mass flow rate is to bypass the top feedwater heaters.

Figure 127 shows the predicted effect of bypassing various percentages of the total feedwater around the top feedwater heater. As can be seen in this figure, bypassing 20% of feedwater flow would reduce ETFR by approximately 1.2%.

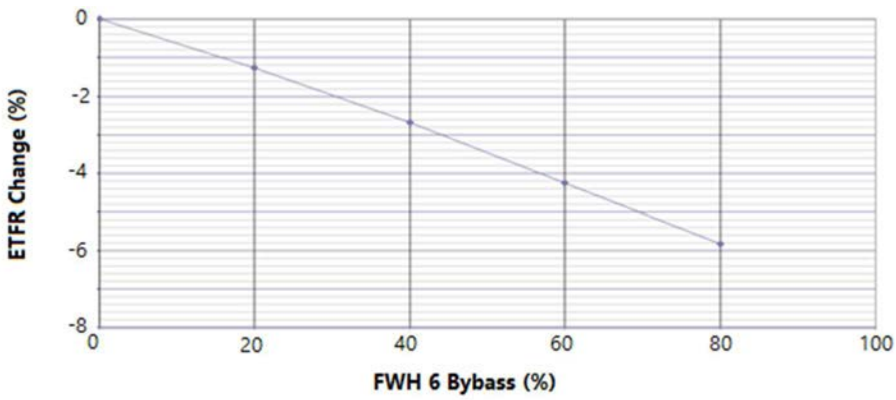


FIG. 127. Bypass FW heater vs ETFR

(f) Partially operate the feed pump turbines on HP steam at 100% RTP power

Many feed pump turbines start and operate on high pressure steam up to a certain point where the HP steam admission valves approach wide open and then the LP steam valve opens to continue picking up load. At unit 100% power, the feed pump turbines normally operate on LP steam only. The LP steam flow to the feed pump turbines are controlled to maintain speed and power supplied to the feed pumps as required for unit feedwater flow control.

Typical feed pump turbines have two admissions on the bottom of the turbine for HP steam admission and six admissions on the top for LP steam. This means that only six of eight admissions are being utilized at 100% power. Although the two admission areas are supplied with different pressures of steam, the temperatures are similar.

A possible method to reduce the flow into the HP turbine at 100% power would be to partially open the feed pump turbine HP steam stop and admission valves during 100% power operation. At a feed pump turbine LP steam flow rate of 72 000 kg/h according to the heat balance with 25% of the feed pump turbine steam requirements supplied with HP steam. This would utilize approximately 18 000 kg/h of main steam and reduce flow into the HP turbine by approximately 0.28%.

(g) Evaluate MSR performance in order to determine methods to increase extraction flows and improve overall performance.

In this example the existing design MSR second stage reheater TTD is 13.9 °C. The actual second stage TTD may be higher. Replacement of the second stage reheaters with new larger tube bundles has the potential to reduce the TTD from 13.9 °C to as low as 6.7 °C. If this could be achieved, the second stage heating steam flow extracted from upstream of the HP turbine could be increased from the existing design value of 352 389 kg/h to approximately 400 855 kg/h. This would increase the HP turbine flow margin by 0.76%.

There is also an improvement expected in overall unit efficiency due to the increase in hot reheat temperature supplied to the LP and feed pump turbines. Higher superheat temperatures entering the LP turbine would provide several improvements. Such as higher available energy and higher inlet volumetric flow delay the point where the steam crosses the Wilson line (becomes saturated) and possibly raise the quality at the LP turbine exhaust. All of these effects are supposed to increase LP turbine output and efficiency.

Modification of the MSR could also result in lower total system pressure drop between the HP turbine exhaust and LP turbine inlet, thus lowering the HP turbine exhaust pressure and providing more available energy across the HP turbine as well.

6.1.5. Optimization of feedwater heater performance

6.1.5.1. Optimization of liquid level

In the feedwater heater operation, it is important to maintain the liquid level at inlet of the drain cooling zone at the design level specified by the manufacturer.

If the liquid level is much higher than design, some additional heat transfer surface in the condensing zone may be flooded. This flooding may reduce the heat transfer capability and increase the TTD. If the liquid level is much lower than design, steam may enter the drain cooling zone and significantly increases the DCA.

Theoretically, the optimum feedwater heater level will be the point preventing the steam from entering the drain cooling enclosure and simultaneously preventing tubes in the condensing zone from being flooded. Figure 128 shows typical change of DCA and TTD with variance of the internal liquid level and the knee point.

Decreasing the liquid level to the knee point will maximize the feedwater heater performance, but the possibility of steam in-leakage into the drain cooling zone also increases. This damages tubes in the drain cooling zone.

A typical result of steam in-leakage is steam flashing inside the drain cooling zone and resultant oscillation of the heater drain temperature. It also significantly increases tube vibration in the drain cooler resulting in tube and baffle plate wear.

Figure 129 shows oscillation of the feedwater heater drain temperature when the liquid level is maintained too low and steam enters and flashes inside the drain cooling zone. Figure 130 shows the change of the heater drain temperature when the liquid level oscillates.

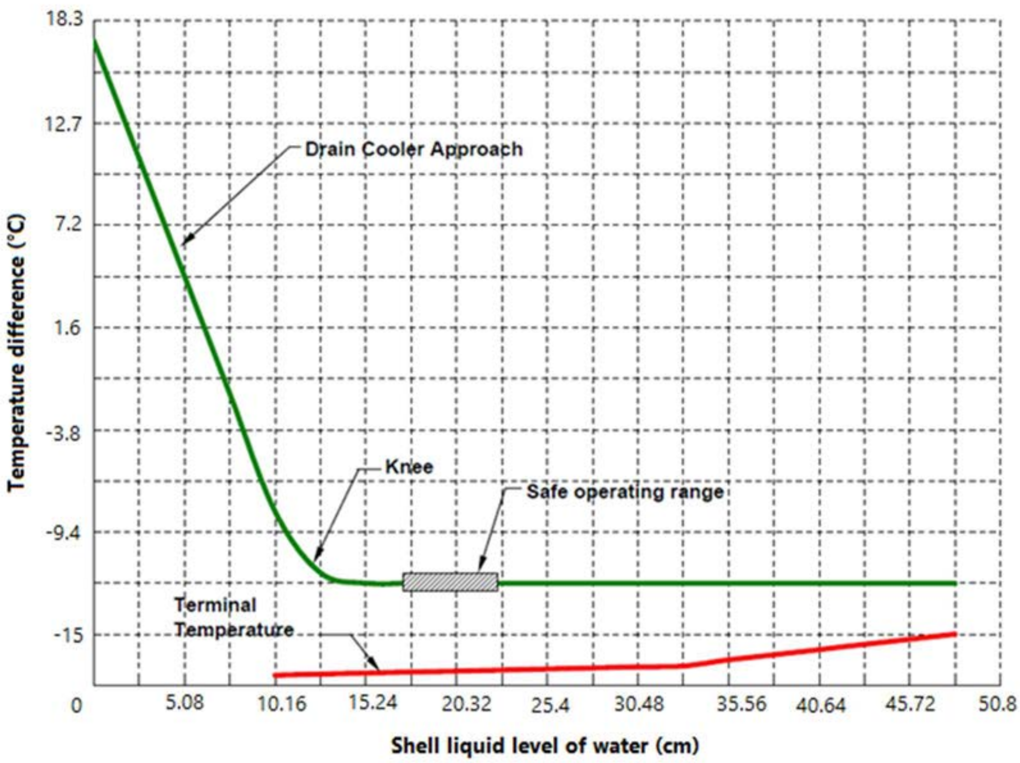


FIG. 128. Typical DCA and TTD vs. internal liquid level (courtesy ASME PTC 12.1 [4])

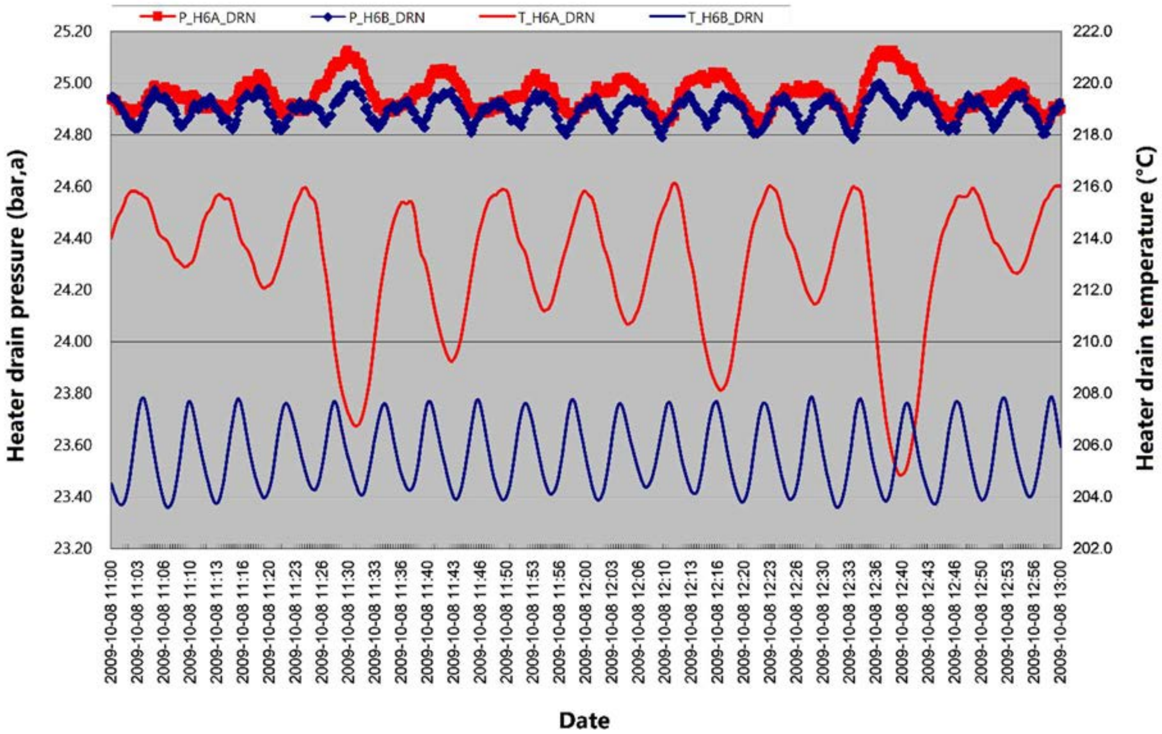
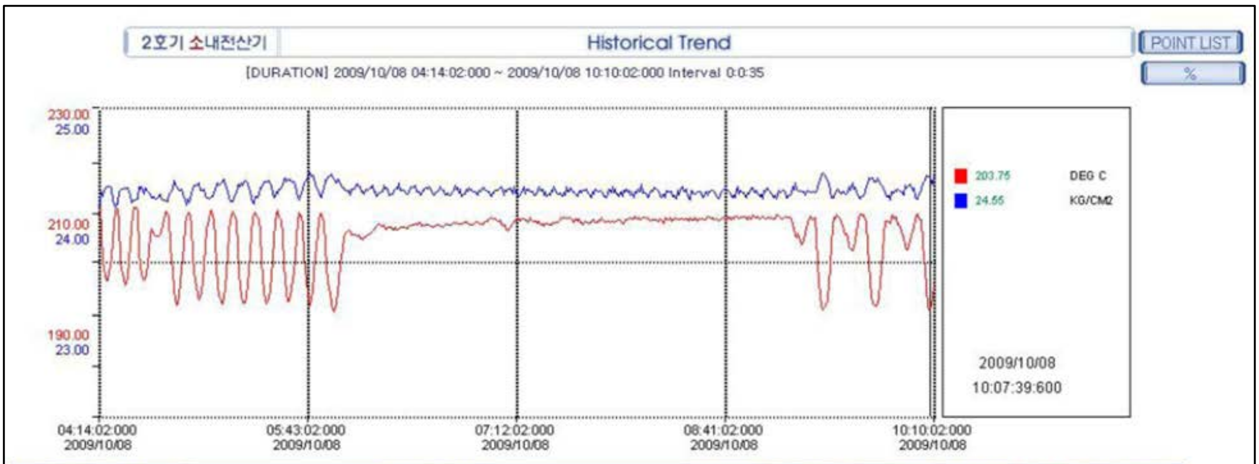


FIG. 129. Oscillation of feedwater heater drain temperature



	TYPE	PID	DESCRIPTION	AVG	MIN	MAX	UNIT	EMIN	EMAX
	AI	T2266A	HP HEATER 6A DRAIN T	210.45	203.57	217.33	DEG C	0.00	350.00
	AI	P2260A	SHELL PRESSURE AT 6A IP HEATER	24.46	24.35	24.58	KG/CM2	0.00	40.00

FIG. 130. Oscillation of heater drain temperature

The example feedwater heater had been operated at inadequate liquid level for a long-term period and during this time damage on the drain cooling zone became worse as shown in Figure 131. The peak to peak variations of the heater drain temperature had been gradually increased and in the long run all internals within the shell had to be replaced.

The lesson learned from this case is that in order to optimize the turbine cycle performance, rapid increase of DCA or oscillation of the heater drain temperature need to always be checked. Also, liquid level needs to be increase if this phenomenon occurs.

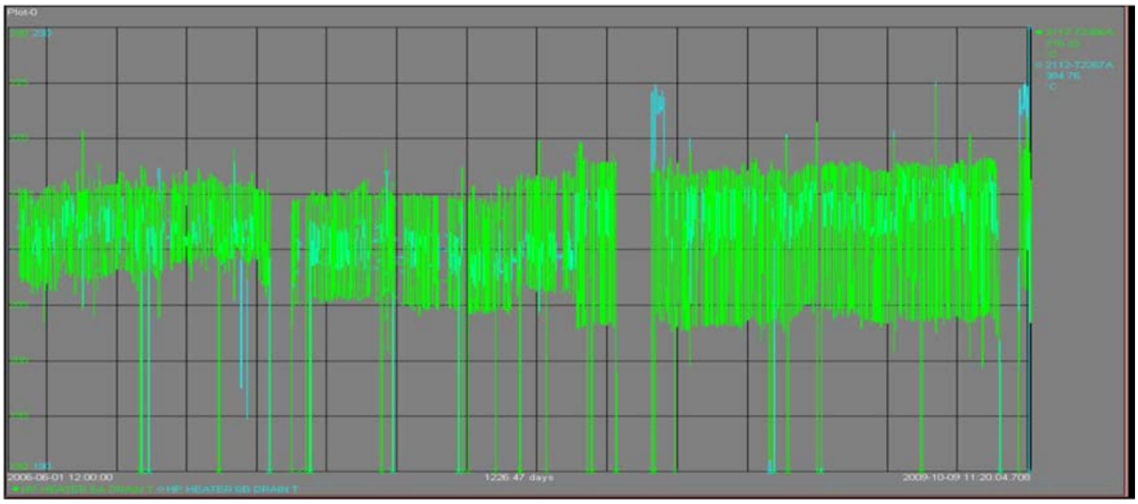


FIG. 131. Trending of heater drain temperature for 5 years

The original feedwater heater operating level may not be suitable for continued use due to:

- Error/mistake in establishing the level;
- Tube plugging in lower rows increases ΔP before drains are sub-cooled (flashing);
- Flow increase – uprate, turbine changes, lower FW inlet temperature;
- Debris at drain cooler entrance increases ΔP before drains are sub-cooled;
- Unstable level control allowing level to fluctuate below minimum.

Improvement of liquid level control may improve heater performance by lowering the DCA. Correct level control is more important for heater reliability and to maximize service life. Level control improvement could include:

- Level test to determine optimum operating level.
- Increased operating level may lower elevated DCA and control valve cycling.
- Possibly require changes to normal control and alarm set points.
- Possibly require change to instrument elevation, pending adjustable range.
- Replacement or upgrade of worn, degraded, or obsolete control instruments.
- Insulating steam legs of level instruments.

6.1.5.2. Heat exchange coefficients approach

Knowing the design information listed in Section 5.9 allows more precise calculation. Under that condition, the following approach calculates the heat transfer coefficients of the heaters, instead of using the manufacturer's data to determine the outlet fluid temperatures. Some inlet conditions cannot be known by permanent sensors, so in addition to the measurements mentioned above, some hypothesis may need to be taken:

- The steam mass fraction in the inlet vapour pipe can be determined using the turbine's reference efficiency (see Section 5.4.2) and turbine exhaust pressures or by using a constant mass fraction obtained by a recent measurement (tracer technique).

— The incoming condensate flow from the upper stage of heaters needs to be measured. In case the emergency letdown line separates from the main letdown pipe downstream of that measurement, it needs to be carefully confirmed that no leakage through these valves is occurring.
— For HP feedwater heaters, two main feedwater lines drive the flow to the heaters. The distribution of the main feedwater flowrate between these two lines can eventually be measured by using ultra sonic flowmeter technology. This measurement is not compulsory since if all the other inputs are known, the flowrate can be calculated along with the heat transfer coefficients with NTU method.

Under these conditions, the NTU method as the one described in ASME PTC 12.1 [4] can be used or computed to determine separately:

— The heat transfer coefficient from the drain cooling zone;
— The heat transfer coefficient from the condensing zone;
— The estimated drain cooling zone exchange surface, which is directly linked to the estimated water level inside the heater.

The calculated coefficient can be compared to manufacturer's coefficients, or to the best values of the coefficients calculated over the heater's life cycle. In case of symmetric degradation of both drain cooling and condensing coefficients, fouling is assumed. And the electric loss can easily be calculated with a thermal calculation code yielding results shown in Figure 132.

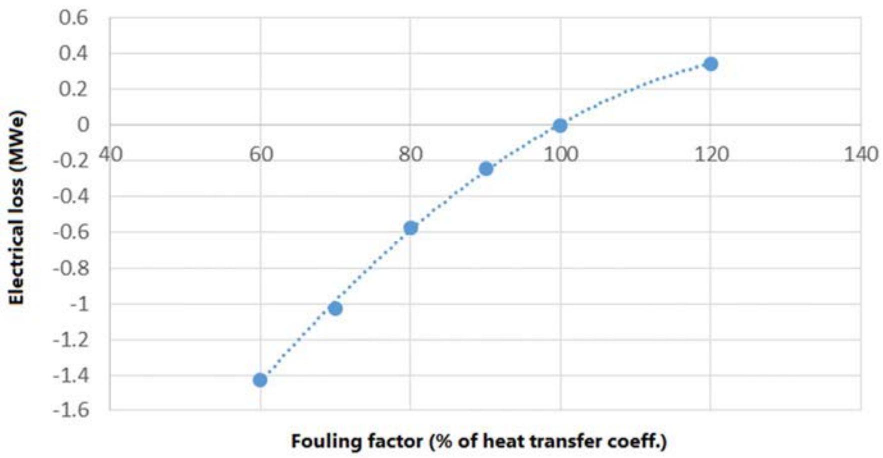


FIG. 132. Electrical losses vs fouling factor curve

Thus, the fouling factor of each heater can be monitored over the component's life as shown in Figure 133.

This approach uses relatively complex calculations. It relies on an important number of measurements and hypothesis, which causes the results to suffer from a high uncertainty.

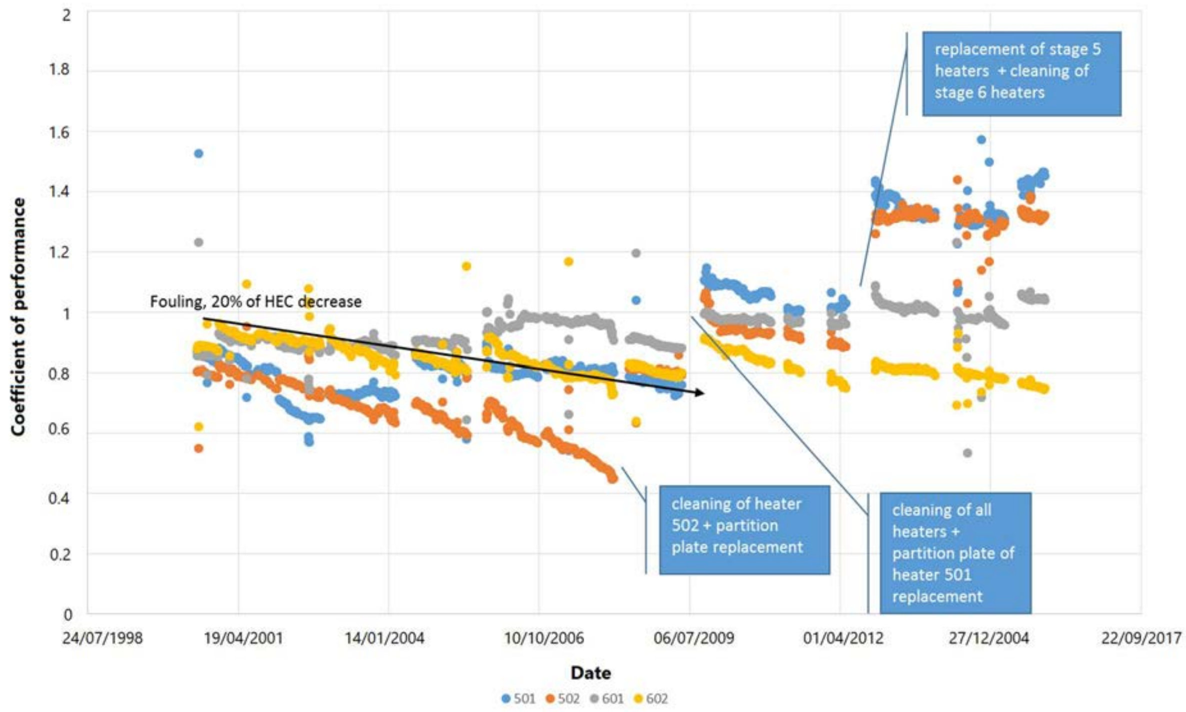


FIG. 133. Fouling factor for HP heaters

Though, a trend analysis over long periods of time gives a good overview of the heater's performance, the TTD and the DCA do not deviate significantly from their reference values. Knowing the fouling rate is essential to decide whether or not heat exchanger tube cleaning is justified:

- Performance gains from a HP heater tubes cleaning varies from 0 to 1 MW/heater depending on the heater's fouling factor. Most common gains are below 0.5 MW.
- Cost of a tube cleaning is also quite low: from 20 000 to 40 000 €/heater and can be realised in two to four days during a shutdown period.

Therefore, the return on investment time varies from less than four months if applied to a heavily fouled heater to several years if the heater is already clean.

6.1.6. Optimization of MSR excess steam vent flow (for 4-Pass tube arrangement)

Contrary to the feedwater heater tube arrangement, MSR has heating steam flows inside the reheater tubes and cycle steam flows on the outer surface. The heating steam supplied to the upper side of U-shaped tube (horizontal MSR) and condenses along the lower side tube.

This resulted in condensate slugs accumulating in the tubes which could then be sub-cooled by the colder cycle steam flowing outside the tubes, as shown in Figure 134.

Typical indications of this condensate plugging are flow and temperature oscillations which cause thermal expansion stresses on the tube bundle supports, differential expansion between tubes, and cyclic thermal stresses. These structural problems can lead to tube and support failures and subsequently reduce the turbine cycle performance.

In order to avoid these structural and thermodynamic problems, excess steam vent flow is always supplied to the reheater tube. The typical amount is 2%~10% of heating steam flow with the earlier 2-pass tube arrangement and 2-3% of heating steam flow with the 4-pass tube arrangement improvements.

The addition of new piping and a heating steam flow control device was followed by the 4-pass design, as shown in Figure 135. That allows optimizing the excess steam vent flow rate up to the limit to prevent the flow and temperature oscillations.

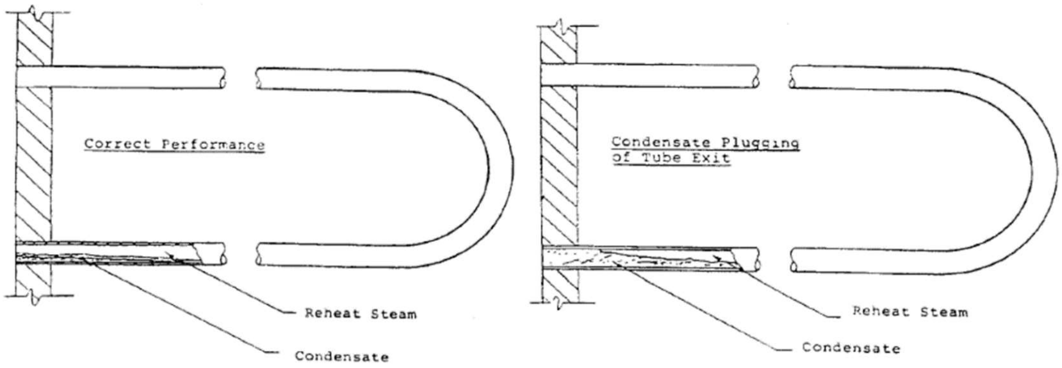


FIG. 134. Reheater tube inside condensation (two pass type reheater)

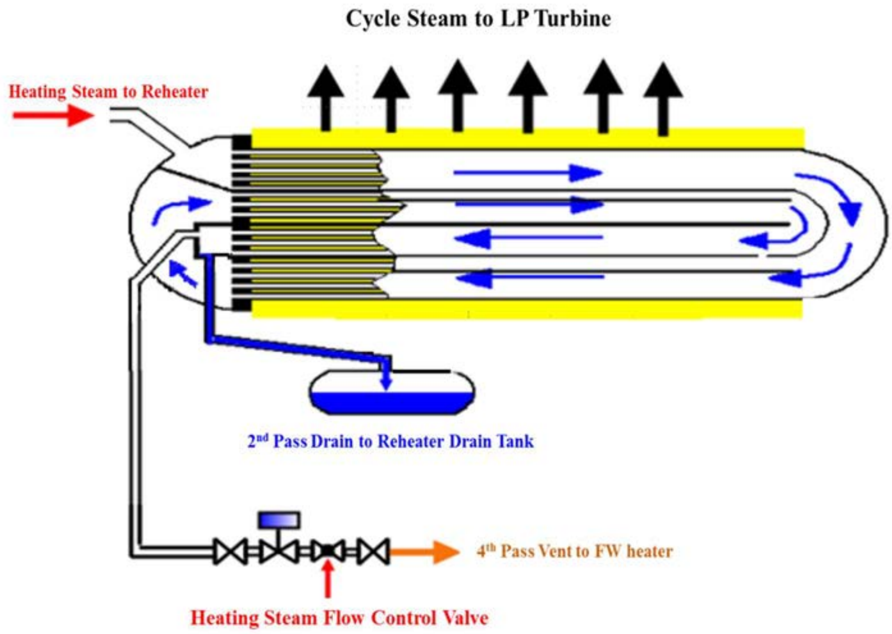


FIG. 135. Four pass type reheater

Figure 136 shows actual field experience of optimizing the excessive steam vent flow. The second stage reheater heating steam flow was measured using the temporary installed precision instrument and adjusted to the design flow rate. In case of the example unit the heating steam flow control valve was open too much as the valve position had been tuned with incorrect plant flow signal in DCS. After readjustment of the valve position with temporary precision test instrument, around 30 t/h of heating steam flow extracted from throttle flow was reduced and consequently the electrical power output was increased by 1 MW.

After optimization of the excess steam vent flow, the condensate slugs can be accumulated downstream of the tube or the heating steam flow control valve. Accordingly, the flow and temperature oscillation need to be always monitored and the heating steam flow control valve be readjusted up to the position which avoids these abnormalities.

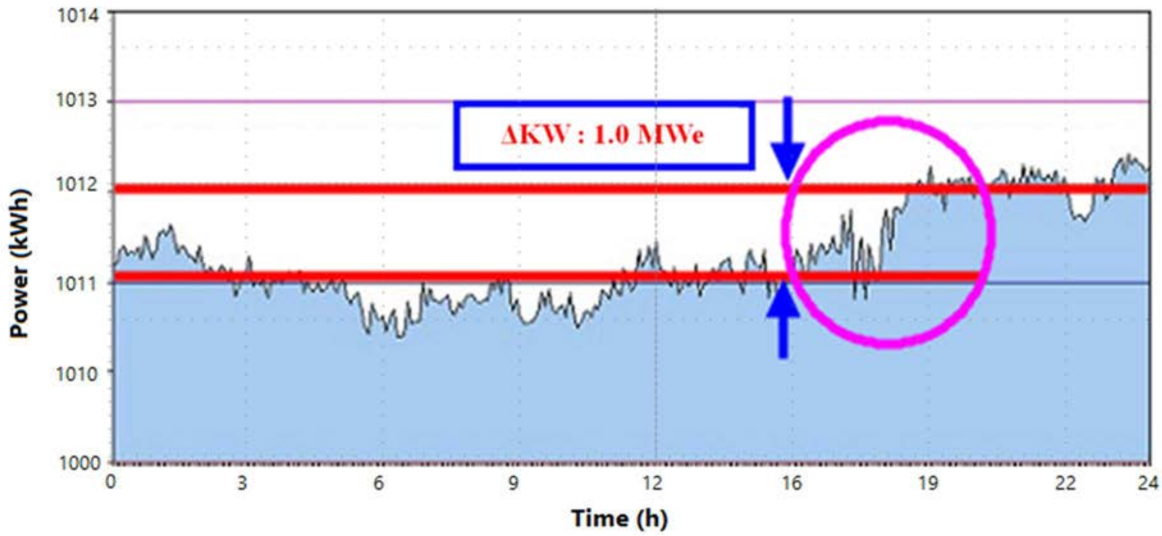


FIG. 136. Change of kW output from excessive heating steam optimization

6.1.7. Optimization of circulation water pump running program profile

A cooling water flow optimization is an important opportunity to increase the unit electrical output and improve steam turbine thermal efficiency. It is also called 'Steam turbine cold-end optimization'. The necessary requirement is that a cooling water flow needs to be variable.

The optimization is possible because the cooling water flow affects:

— The condenser pressure / steam enthalpy drop; higher cooling water flow rate causes lower condenser pressure / higher enthalpy drop = higher steam turbine output.
— The pump drive input; higher cooling water flow rate requires higher pump drive input.

From a certain cooling water flow rate called the 'reversal point' the pump drive input exceeds the increase of the turbine output. This 'reversal point' depends on:

— Reactor heat output (steam turbine input);
— Ambient air (wet bulb temperature, etc.) for cooling tower application;
— Seawater or river water temperature.

Thus, the objective of optimization is to define such cooling water flow rate in these boundary conditions for which the unit electrical power output (= the generator gross capacity reduced by the pump drive input) is the highest. This is not easily solved. A holistic point of view, which covers a mutual relationship between main equipment and deep understanding of the phenomena under consideration, is imperative.

Mathematical models make it possible to describe and understand the behaviour of cooling water circuit with adequate accuracy. The practical use of a mathematical model is to:

— Understand the equipment's performance characteristics, see Figure 137;
— Formulate the optimization task and its objectives in more detail;
— Apply the unit performance optimization;
— Validate result, see Figure 138.

The optimization is based on the highly accurate model. There are two general approaches to the mathematical model creation:

— First principle model (based on physical laws);
— Data driven model (also called empirical model; validated process data are crucial).

A data driven model is typically more accurate because it is derived directly from the actual behaviour of the real object. The optimization usually comprises the following interconnected steps (lettered from (i) to (ii)):

(i) Step 1: Modelling (data driven model building)
- The system definition;
- The experiment on the real object (unit); data collection;
- The mathematical model building and simulation.

(ii) Step 2: Model predictive control application and validation
- Cooling water flow control application;
- Behaviour assessment, results validation.

The rigorous assessment of the optimization benefit and resulting validation are based on a comparison of the two models as shown in Figure 138 which describes the unit performance before and after optimization:

— Mathematical model of the (cyber physical) system uses the old (original) control strategy.
— Mathematical model of the (cyber physical) system uses the new control strategy based on the mathematical model.

The benefit of optimization can be increased by covering:

— Process dynamics (water basin accumulation, etc.);
— Schedule of reactor power changing;
— Weather forecast.

The usual benefit of optimization is a 1 – 2 MW output increase for a 1000 MW unit. The higher benefit is related to the higher wet-bulb temperatures. Data reconciliation is advised for data preparation.

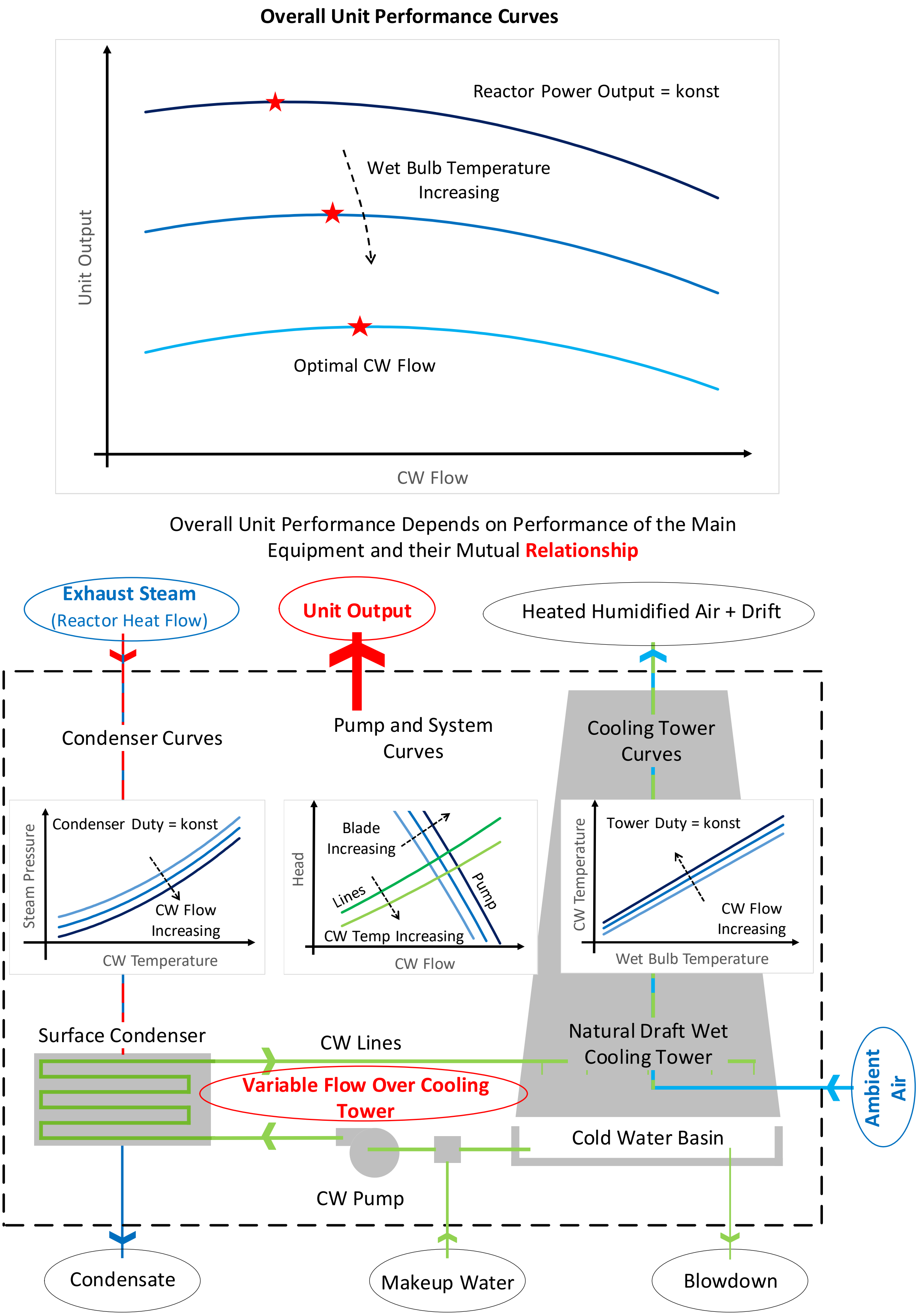


FIG. 137. Overall unit performance

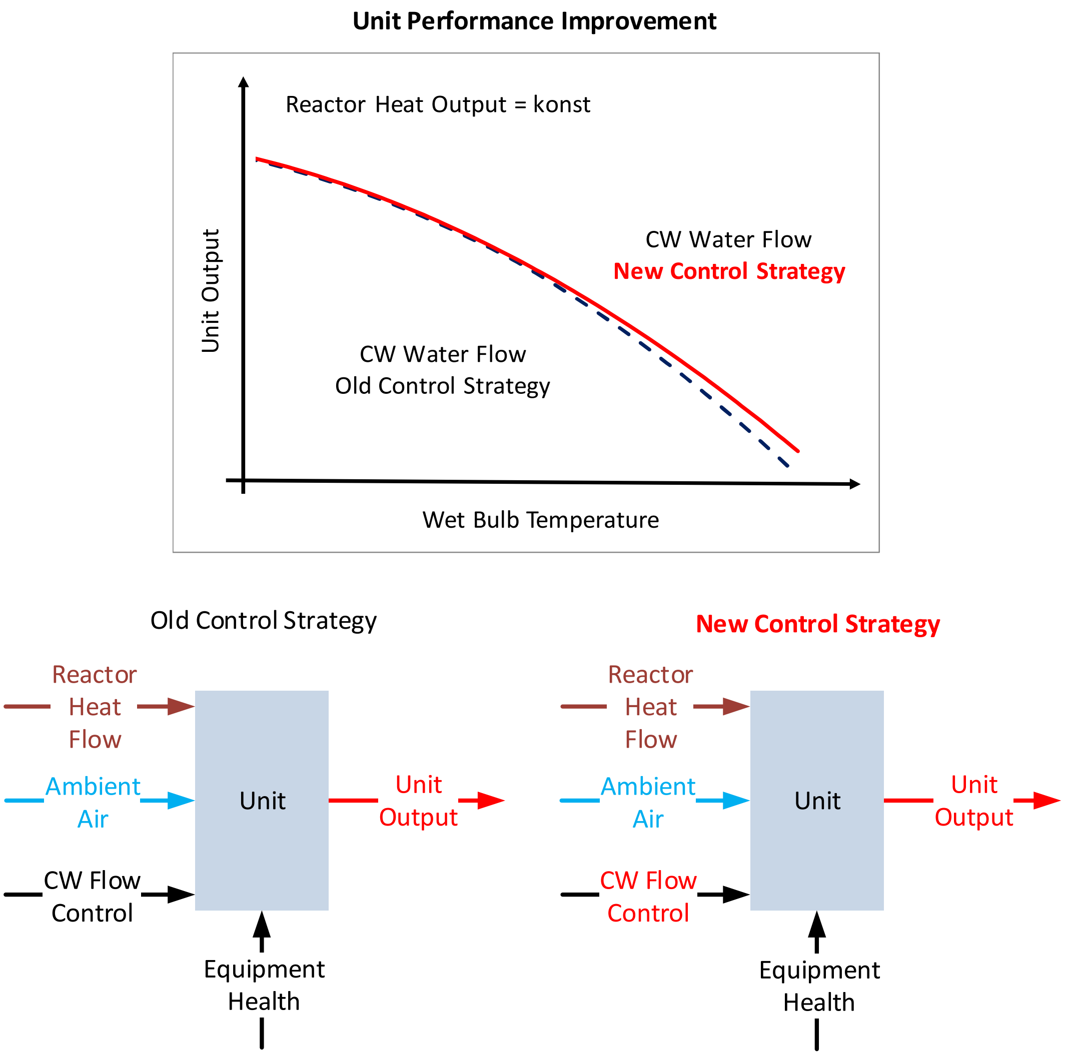


FIG. 138. Control strategy validation

6.2. MODIFICATION AND REPLACEMENT

Improving the turbine cycle performance beyond the benchmark for a new and clean condition may not be possible without modification or replacement of turbine cycle components. These modifications require thermo-economic analysis to justify the necessary investment. This subsection basically refers to [29] and [30] to provide guidelines for modification and replacement of turbine cycle key components.

6.2.1. Steam turbine

6.2.1.1. Turbine stage performance

Performance losses of a turbine stage (as shown in Figure 139) can be categorized into profile losses and leakage losses. The profile losses are composed of friction losses, secondary flow losses, non-uniform flow losses and flow separation losses. The leakage losses typically composed of tip seal leakage, root seal leakage and diaphragm packing (or nozzle packing) leakage. All of steam turbine suppliers are developing their own technology to minimize these losses and increase turbine stage efficiency.

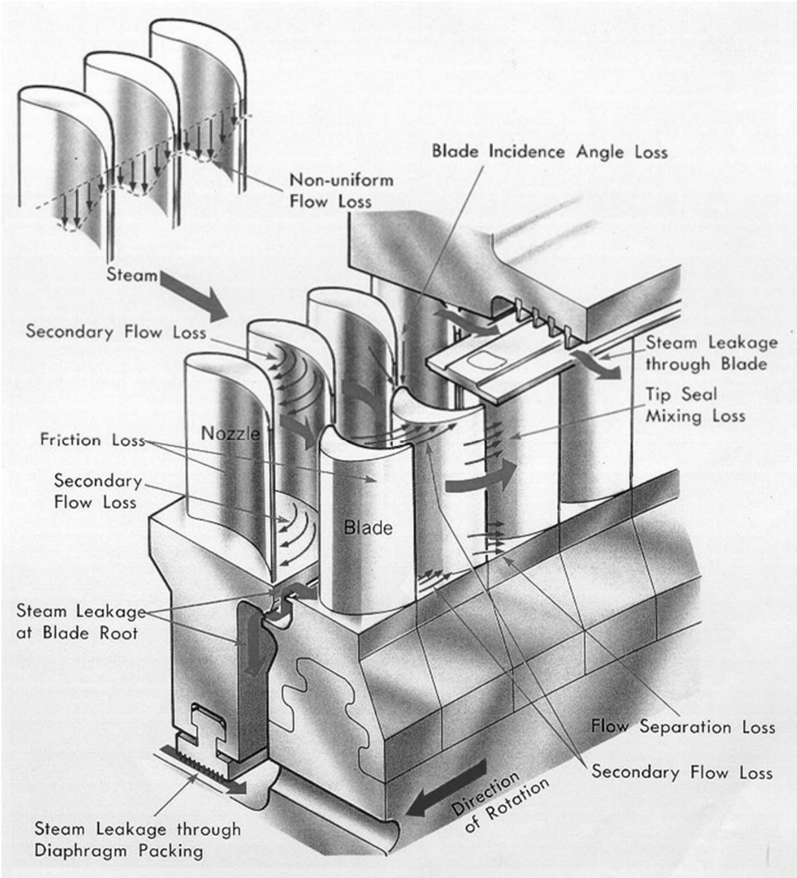


FIG. 139. Turbine stage performance losses (impulse turbine)

In the steam turbine replacement, it is common to replace the whole steam path of the HP and/or LP turbine section applying state of the art technology. The typical scope of replacement includes turbine inner casing with stationary blades and turbine rotor with moving blades. Intuitively, applying advanced leakage control technologies on the existing diaphragm (nozzle) packing of the HP turbine would be beneficial for performance improvement. However, it is not very effective because of the fixed sized of the balance holes (wheel or blade). The reduced diaphragm packing leakage will increase the blade root leakage to the balance hole to establish pressure balance. This will result in almost no change in the blade flow that is actually working to rotate the rotor (for impulse turbines only).

6.2.1.2. Full arc and partial arc

Most of nuclear turbines use more than one throttle valve. A four-valve design is common. There are two basic methods of admitting steam to the HP turbine first stage nozzles, full arc and partial arc. Full arc admits steam from all the control valves to a common chamber. Steam then flows from the chamber into all first stage nozzles located in a full arc in front of the first stage HP rotor blades. Partial arc admits steam into separate chambers, usually one for each control valve. Steam then flows from the chambers to a part of the first stage nozzle, each set of nozzles occupying a part of the full arc delivering steam to the first stage blades.

Partial arc turbines are designed to open the valves sequentially with load and to operate with the first three valves fully open and the fourth valve partially closed and throttling at full reactor power.

Some throttling at full load is needed for control. Full arc admission results in parallel steam flow through all first stage nozzles. Partial arc admission results in steam flow through some, but not all, of the nozzles.

As a result, especially at part load, more control valve throttling losses occur with full arc than with partial arc. However, full admission can give better HP turbine efficiency at VWO because this type has no inactive portions (dead arc) like in partial arcs (see Figure 140).

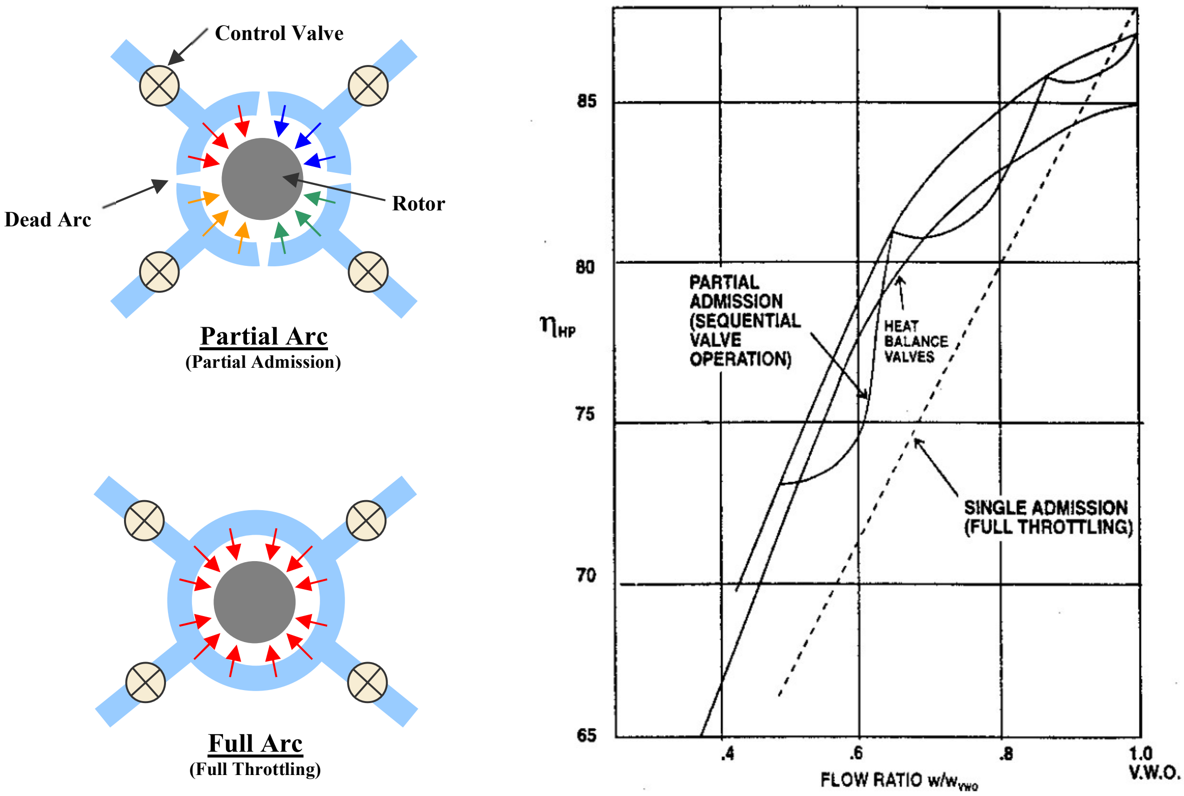


FIG. 140 HP Turbine Efficiency Characteristic – Full Arc vs. Partial Arc

6.2.1.3. Case study –steam turbine retrofit

Four project cases were studied for performance improvement. Gross electrical output increase, following the retrofit of HP and LP turbines by measuring and comparing the corrected gross electrical output at the generator terminals prior to and after the retrofit work. For each project, the baseline and verification tests were conducted in accordance with ASME PTC 6 [3] alternative test method. Test results are shown in Table 23 and Table 24.

TABLE 23. Δ KW OUTPUT AFTER REPLACEMENT OF HP TURBINE STEAM PATH AND INNER SHELL

	Baseline test	**Verification test**	**Δ kW output**
Plant A	1006.200 MW	1013.120 MW	6.920 MW (0.69%)

TABLE 24. Δ KW OUTPUT AFTER REPLACEMENT OF LP TURBINE STEAM PATH AND INNER SHELLS (3 CASINGS)

	Baseline test	**Verification test**	**Δ kW output**
Plant A	982.333 MW	1002.834 MW	+20.501 MW (2.09%)
Plant B	993.831 MW	1027.435 MW	+33.605 MW (3.38%)

6.2.2. Moisture separator reheater

6.2.2.1. Modification or design change

The following is a compilation of improvements to MSRs which require modifications. These modifications are discussed in the context of retrofits or changes within the existing shell. More extensive changes and improvements may be possible via replacement of the entire vessel.

Modifications to improve moisture separation include the following:

- Replace wire mesh separators with chevron separators.
- Replace single pocket chevron separators with double pocket design.
- Addition of perforated plates in front of chevrons to improve steam distribution.
- Internal steam manifolds to improve steam distribution to the chevrons.
- Addition of a deck plate to prevent moisture re-entrainment.
- Chevron vanes slanted or sloped outwards and towards the inlet cycle steam.

Improvements in the steam reheat function have been achieved by the following:

- Excess steam flow capability to reduce condensate oscillation and associated thermal stress cracking of reheater tubes (4th pass vent chamber arrangement).
- Tube bundle upgrades with improved materials and features to address failure modes, improve surface area, and reduce steam pressure drop. Tube materials have been changed to 439SS with integral fins (and higher fin density) to reduce SCC susceptibility and erosion damage while maintaining TTD capability. Where space permits, tube bundle width has been increased to distribute the steam over a larger cross-sectional flow area allowing TTD to be maintained with passage over fewer rows of tubes which reduces shell-side pressure drop.
- Seal strips and flow restricting bars to reduce flow bypass in the tube bundles.
- Improved seals installed around reheater bulkhead plates and side walls to prevent steam bypasses.

Figure 141 shows differences in the internals design between older and newer MSRs. Replacement of the entire MSR may be required or economically justified to achieve all of the internal design improvements.

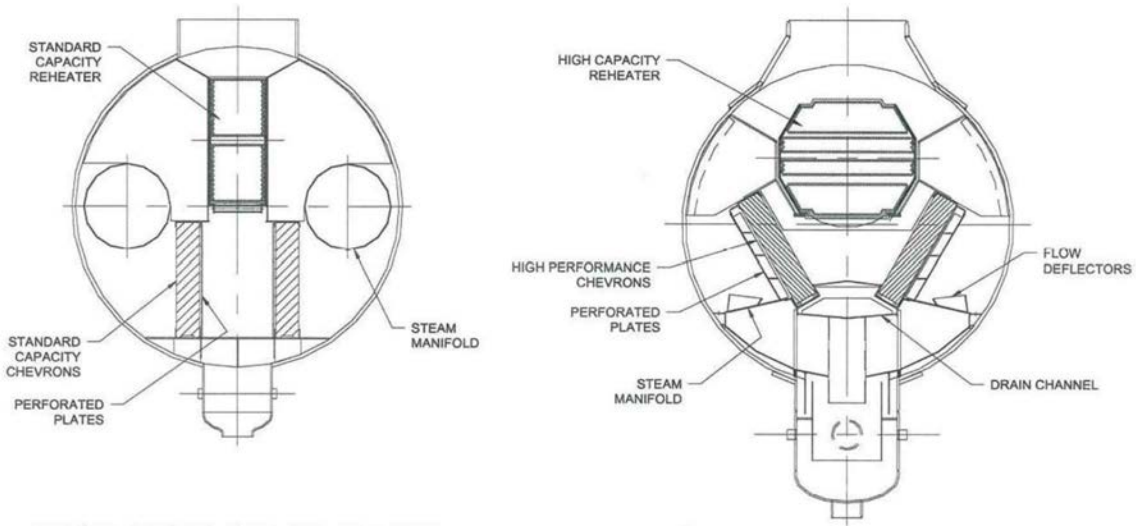


FIG. 141. Old (left) and new (right) MSR designs

6.2.2.2. Case study – Replacement with new design

The double-stage MSR was replaced with a new internal design changing single pocket chevron to double pocket design. The tube bundles were also upgraded from two pass design to four pass design as to optimize the excess steam flow. The baseline and verification tests were conducted in accordance with ASME PTC 12.4 [17] in order to determine the change in the MSR performance parameters. ASME PTC 6 [3] alternative test, including the MSR into the test boundary, was conducted simultaneously to compare the corrected gross electrical output at the generator terminals prior to and after the replacement as shown in Table 25.

The test result showed that the MSR performance was generally improved. An increase in the corrected gross electrical output was slightly above 3 MW which is much less than expected.

Even though the reheater TTD was improved, the reheater extracted more throttle steam which would normally be routed to the HP turbine. Moisture separator effectiveness was also increased with double chevron design. However, cycle steam pressure drop was also increased while reducing the available energy to the HP turbine. The 3 MW may be typical value of performance improvement from replacement of the MSR internals.

TABLE 25. Δ KW OUTPUT AFTER REPLACEMENT OF MSR INTERNALS WITH NEW DESIGN

	Unit	**OLD MSR** ⓐ	**NEW MSR** ⓑ	**Change** ⓑ-ⓐ
2nd reheater heating steam	kg/h	116 594	120 204	3610
1st reheater heating steam	kg/h	106 577	101 173	-5404
2nd reheater excess steam	kg/h	3600	1202	-2398
1st reheater excess steam	kg/h	3402	1012	-2390
2nd reheater TTD	℃	18.58	10.35	-8.23
1st reheater TTD	℃	13.86	8.62	-5.24

TABLE 25. Δ KW OUTPUT AFTER REPLACEMENT OF MSR INTERNALS WITH NEW DESIGN (cont.)

	Unit	OLD MSR ⓐ	NEW MSR ⓑ	Change ⓑ-ⓐ
Moisture separator effectiveness	%	95.30	99.96	4.66
Cycle steam pressure drop	%	0.3223	0.3648	0.0425

6.2.3. Feedwater heater

From the thermo-economic point of view, modification and replacement of a feedwater heater just to improve TTD and DCA is mostly not beneficial. These modifications are typically justified by plant reliability such as avoiding penitential tube failure, which may cause long term feedwater heater bypass operation. In this regard, [30] suggests conditions for run / repair / replacement decision as follows.

6.2.3.1. Run decision

A decision to return a heater to service can be made from the following conditions:

- The heater examinations revealed no defective indications, and no tube plugging was required. Visual examinations of the vessel internals showed no tubesheet ligament cracking or partition plate weld cracking. Any shell-side examination results showed no visible signs of tube support plate bowing. No presence of loose parts or foreign objects, no shell thinning, or no impingement plate damage. If the trends show that the heater thermal performance can be maintained for the expected life of the heater, continued operation is typically justified.
- The number of total plugged tubes does not exceed the plugging limit for the heater.

6.2.3.2. Repair decision

A decision to repair the heater is made to restore lost performance or extend the life of the heater. Some actions for repairing the heater are:

- Remove all tube plugs and retest the tubes by eddy current testing or other means. Only tubes that are defective are supposed to be plugged. This can return tubes to service and restore performance from previous proactive plugging.
- Install sleeves in tubes that were previously plugged. This can restore lost performance from previously plugged tubes and help prevent future forced outages.
- Perform diaphragm, pass partition, tubesheet, channel, and cover repairs to restore lost performance.
- Perform repairs to the tube-to-tubesheet joint to restore lost performance.
- Replace or weld repair sections of the heater shell with thinning caused by impingement plate erosion. This can extend the life of the heater.

6.2.3.3. Replace decision

If the failure mechanisms are widespread, the number of tubes plugged is at or above the limit that affects performance, and continued failures are supposed to affect the heater reliability, replacement is recommended.

Replacement options include retubing, rebundling, and complete replacement. Retubing a heater is replacing only the tubes.

Rebundling is replacing the tubes, tubesheets, support, and baffle plates in the existing heater shell. Complete replacement includes a new shell, tube bundle, tubesheet, and support and baffle plates.

The remaining life prediction for a heater is needed before a replacement recommendation can be evaluated.

6.2.3.4. Case study – Tube bundle replacement

Due to damage of the drain cooling zone caused by poor liquid level control and resultant steam in-leakage, the feedwater heater buddle was replaced. Pre- and post- replacement tests were conducted to check the feedwater heater performance improvement. Tests to confirm changes in the gross electrical output at the generator terminal were conducted and are demonstrated in Table 26. The test result shows that the DCA was highly reduced and recovered to normal performance. However, the increase of the electrical power output was marginal.

It is true that this change in a feedwater heater had little impact on the electrical output. However, if this damage had not been corrected, it may have progressed to future tube failure. The electrical power output losses caused by such feedwater heater operation will not be comparable.

TABLE 26. Δ KW OUTPUT AFTER REPLACEMENT OF FEEDWATER HEATER INTERNALS

	Unit	**OLD FW HTR** ⓐ	**NEW FW HTR** ⓑ	**Change** ⓑ-ⓐ	**Δ kW Output**
Feedwater heater#6 A	℃	1.45	2.36	0.92	-184 kW
	℃	23.93	9.49	-14.44	325 kW
Feedwater heater#6 B	℃	1.90	2.11	0.21	-40 kW
	℃	15.10	7.89	-7.21	154 kW
		sum			255 kW

6.2.4. Condenser

A variety of modifications or changes are possible to improve condenser performance and/or capability as follows.

6.2.4.1. Surface area recovery

Recovery of tube surface area can be achieved by replacing or returning to service plugged tubes.

- Partial re-tubing can be performed, although requires water box removal and adequate unrestricted space to handle the tube lengths.
- Sleeving or coating over tube defects may allow tubes to be returned to service.

- NDE, inspection, and pressure testing may allow tubes to be returned to service if they were precautionary or insurance plugged, such as in response to a leak from external damage when the condition of nearby tubes is unknown.

6.2.4.2. Tube cleanliness improvements

Improvements which maintain a higher condenser CLF will reduce efficiency losses and may avoid the need for power reductions during periods of high circulating water temperature.

- Improve debris barriers (e.g., traveling screens) to reduce macro-fouling load.
- Chemical treatment changes to reduce scaling, micro-fouling, or macro-fouling.
- Improve/add ball cleaning system.

6.2.4.3. Tube bundle replacement

Re-tubing or replacing the entire tube bundle recovers lost original surface area caused by tube plugging. The case of bundle replacement typically includes additional surface area from changes to tube layout to tube material changes to address past failure modes. This may involve improvements to the air removal configuration of the tube bundle.

The change to more robust tube materials such as titanium or highly alloyed stainless steels to avoid past failure modes results in use of less thermally conductive tube material when changing from admiralty brass or 90-10 copper-nickel tubes. An increase in biological micro-fouling may also result due to the absence of the natural bio-toxicity of the original copper alloy tube material. Thinner tube materials can be used to partially offset part of this reduction in thermal performance. However, much of the surface area recovery or addition with the replacement tube bundle may be needed to match performance with the original condenser. Condenser tube bundle design involves trade-offs between achievable operating pressure, tube surface area, overload capability and velocity constraints associated with the available volume in the shell.

6.2.4.4. Circulating water/cooling tower improvements

Condenser performance improvements can be achieved by projects which increase circulating water flow and/or decrease circulating water temperature. Improvements to the circulating water pumps may include:

- Rebuild/overhaul CW pumps to restore flow.
- Upgrade CW pump design to improve pump efficiency and performance.
- Pump bay or intake modifications to reduce pump degradation and/or avoid air entrainment problems.
- Upgrade CW pump motor with variable frequency controls to adjust pump speed to most efficiently meet plant needs.

Major projects to reduce circulating water temperature may include:

- Cooling tower upgrades to improve performance.
- Installation of helper towers to meet demand during peak season.
- Relocation of CW intake for once-through systems to deeper (i.e. colder) water intake, or to reduce thermal recirculation from CW discharge.

6.2.4.5. *Air removal upgrades*

Improved analysis of condenser air removal performance has revealed many condensers operate with poor air removal designs and/or ineffective air removal equipment causing a significant increase in condenser back pressure. Upgrades to correct or improve these conditions have yielded significant improvements to condenser back pressure, condensate chemistry, and air in-leakage removal capability. Improvements equivalent to 30% of the effective condenser surface area have been reported via retrofits to address the poor air removal design of the condenser tube bundle.

6.2.4.6. *Power uprates*

Power uprates increase the heat duty of the main condenser. The increased steam flow velocity may result in accelerated tube damage due to vibration and/or water droplet impingement erosion, although that is not the emphasis of this publication. The thermal performance capability of the condenser will have been evaluated as part of the uprate modification process. The evaluation needs to consider the historical range of CLF and circulating water temperatures to determine the projected condenser back pressure with the uprate heat load. The power uprate modifications need to include condenser replacement or efficiency improvements to avoid the need for power reductions during periods of peak circulating water temperatures.

Problems with the original condenser tube bundle can be greatly exacerbated by incremental increases in steam flow. Power uprate evaluations are often solely focused on the thermal design capability of the original condenser. They give inadequate consideration to its operating history for performance problems and failure modes which may be aggravated by the uprate. This type of limited uprate evaluation can result in a condenser which is thermally capable on paper but unreliable in operation.

6.3. SUMMARY

This section introduced preventive or corrective measures, including operation changes, to recover and optimize the NPP performance. Details of preventive and corrective activity for each component are beyond the scope of this section. EPRI report [29] can also be used in conjunction of this publication for field troubleshooting.

Guidelines for modification and replacement of turbine cycle components are also discussed in this section. Modification and replacement are not always beneficial from a thermo-economic point of view. These works would be rather justified by potential mechanical or structural problems in long term operation. Plant stability have priority over thermo-economic analysis such as payback period.

7. CONCLUSION

This publication provides various methodologies for tracking and trending NPP thermal performance. The benefits of each are described in the various sections of the publication. All methodologies described can be used to monitor and assess NPP thermal performance as described in this publication. The method(s) actually used by a particular utility will vary based on the following issues:

(a) Quantity of measurement system installed at the plant – how many measurements are connected to the plant computer system and can be accessed by thermal performance software.

(b) Quality of the measurements installed at the plant. Some of the methods are more amenable to lower quality measurements and some require higher quality measurements.

(c) Ability to use test instrumentation for periodic testing – having access, qualified instruments etc.

(d) Personnel available to maintain software or budget to have it maintained by vendors

(e) Management support for the monitoring of thermal performance.

(f) Budget available for the various tools described in this publication.

7.1. SUMMARY OF METHODS

(a) ASME PTC Testing

This methodology uses the ASME PTCs as a basis for periodically testing plant thermal performance. The basic concept of the code test is measuring performance parameter of a test target and then correcting it for affecting variables external to the test boundary. In other words, the object of the code tests is to determine the expected performance parameters when the external affecting variables are operated at the base reference conditions. This methodology can employ specific test instrumentation. Or it can use plant measurements if they are regularly calibrated and there is enough instrumentation to match the code requirements.

(b) Data validation and reconciliation

The goal of the DVR process is to correct any instrument biases and minimize the random error. Therefore, the other parameter measurements will be corrected by the process. This approach allows the use of typical instrumentation installed in the plant to accurately monitor plant thermal performance. It also provides the ability to obtain measurements based on calculations and know the uncertainty of these measurements. This methodology requires significant experience with the type of software utilized and a substantial up-front investment. It will provide many additional benefits for thermal plant monitoring such as pinpointing either component or instrument issues or high quality ‘pseudo’ measurements. That can be used to evaluate plant thermal performance.

(c) Data driven methodology

The data driven methodology is one that incorporates the use of sophisticated computer software using data reconciliation techniques. This is to achieve a high probability that the measurements have been corrected to the most likely value. This information is then used as input to models which produce performance indices. That can be evaluated to determine plant efficiency and the likely cause of reduced plant efficiency. This approach incorporates the DVR approach and adds a further element of performance models.

(d) Compare with components and plant model

This methodology consists of estimating the thermal power by means of using information from the BOP (or turbine cycle) and how it turns the energy into electricity. To achieve such a balance with a reliable degree of confidence, the use of a model as a ‘digital twin’ of the plant brings systematic application of formulas and efficient data treatment. Once correctly tuned, computed values are expected to provide a good representation of main physical quantities and, finally, an image of secondary circuit performance. While this methodology may require some software and dedicated time to develop, once established it will provide a robust capability going forward with minimal engineering time.

(e) Generic performance calculation or empirical relationships methodology

For an overall evaluation of plant performance some plants use a methodology based on empirical relationships between plant parameters and plant output. Based on these relationships the amount of generation can be compared to a 'target' value derived from corrections supplied by these empirical relationships. Corrections can be applied to the measured plant generation to account for known effects. Performing this 'accounting' can prevent an engineer from evaluating a problem that is not real. This methodology can use vendor curves, thermodynamic modelling or using historical plant data to determine the relationships. This methodology may be particularly useful for those utilities that have limited resources or do not have good design data for their equipment.

7.2. APPLICATION

The information, examples, equations and other guidance contained herein originate from a diverse collection of references, expertise and experience. The contributors and reviewers worked hard, for example, to ensure proper conversion of details provided in imperial to International System of Units (SI). This includes conversion of relevant constants in equations. Despite these efforts, readers are encouraged to verify equations, constants and related details using the references provided, similar standards or other thermodynamic, heat transfer or fluid dynamic technical resources as appropriate to ensure correctness prior to applying this work to the thermal power determination, performance measurement or monitoring of an operating NPP.

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ABBREVIATIONS

APR	advanced pattern recognition
ASME	American Society of Mechanical Engineers
BAHR	best achievable heat rate
BOP	balance of plant
BWR	boiling water rector
CANDU	Canada Deuterium Uranium
CF	capacity factor
CLF	cleanliness factor
DCA	drain cooler approach
DVR	data validation and reconciliation
EDF	Électricité de France
ELEP	expansion line end point
EPRI	Electric Power Research Institute
ETFR	equivalent throttle flow ratio
FWPT	feedwater pump turbine
HEI	Heat Exchange Institute
HP	high pressure
HR	heat rate
INPO	Institute of Nuclear Power Operations
IR	infrared
KHNP	Korea Hydraulic and Nuclear Power
KPI	key performance indicator
kWh	kilowatt-hour
lbm/h	pound mass per hour (0.0001259979 kg/s)
LORC	Loss-of-Reactivity-Control
LP	low pressure

LTO	long term operation
MSR	moisture separator reheater
MWe	megawatt electrical
NPP	nuclear power plant
ORT	operation at reduced temperature
PTC	performance test codes
PWR	pressurized water reactor
ROP	Regional Overpower Protection
RTP	reactor thermal power
STP	secondary thermal power
SG	steam generator
SRS	software requirement specifications
TPE	thermal performance engineer
TPI	thermal performance indicator
TTD	terminal temperature differences
UEEP	used energy end point

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