N AMERICAN NATIONAL STANDARD

ASME
PTC 12.4–1992

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Moisture Separator Reheaters

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PERFORMANCE
TEST
CODES

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
United Engineering Center
345 East 47th Street New York, N.Y. 10017

Moisture Separator Reheaters

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United Engineering Center 345 East 47th Street New York, N.Y. 10017

Date of Issuance: May 24, 1993

This Standard will be revised when the Society approves the issuance of a new edition. There will be no addenda or written interpretations of the requirements of this Standard issued to this edition.

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FOREWORD

(This Foreword is not part of ASME PTC 12.4-1992.)

Moisture Separator Reheaters (MSRs) were introduced to steam power cycles after the advent of commercial nuclear power. A moisture separator, with no reheat was first added to nuclear power cycles to minimize the low pressure (LP) turbine erosion caused by wet steam prevalent in those cycles and improve turbine cycle performance. Steam reheat was added later to reduce further the quantity of moisture in the steam passing through the LP turbine and to increase further the efficiency of the LP turbine.

The first MSRs were susceptible to many modes of failure. Great technological advances have occurred over the past 30 years with respect to MSR design and operation. These advances increased the reliability and enhanced the performance of the MSR which provided the momentum and justification for MSR upgrades.

During the 1970s and early 1980s an increasing number of utilities were involved in MSR upgrades which included replacing portions of or their entire MSRs. The ASME Board on Performance Test Code was notified in June 1984 that no code existed for the testing and analysis of MSRs. PTC-6 (1982) on steam turbines treated the MSR as an integral part of a turbine generator, which it is when purchased as a package. The Board authorized the formation of a new performance test code committee to develop a code for the treatment of the MSR as a separate component.

A new committee was formed and first met in December 1985. Numerous drafts were developed over the next 4 years, each more detailed than the previous. Upon the completion of appendices containing a set of sample calculations and a complete uncertainty analysis, the draft was released for the industry review in July of 1990. The comment resolution process, completed in April 1991, strengthened the document. The committee was balloted and approved the code draft in July 1991. The Board on Performance Test Codes approved the code in January 1992. This test code has been approved as an American National Standard by the ANSI Board of Standards Review on November 24, 1992.

PERSONNEL OF PERFORMANCE TEST CODE COMMITTEE No. 12.4 ON MOISTURE SEPARATOR REHEATERS

(The following is the roster of the Committee at the time of approval of this Standard.)

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In addition to the above personnel, the Committee is deeply indebted to Mr. Peter Bird, Mr. Al Smith, Mr. Clement Tam, and Mr. Richard Harwood for their contributions in the development of this Code.

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SECTION 0 — INTRODUCTION

0.1

A Moisture Separator Reheater (MSR) is a nuclear power plant component located between the high and low pressure turbines. Its purpose is to remove moisture and add superheat to the cycle steam before the steam enters the low pressure turbine. It consumes throttle steam, and may also consume high pressure extraction steam in the heating process. The MSR introduces an additional pressure drop in the turbine expansion while accomplishing these functions. The use of a properly designed and adequately performing MSR will result in a cycle heat rate improvement.

0.2

One of the purposes of this test Code is to consider the separate functions of moisture separation and either one or two stages of steam reheat. This procedure can be employed to combine the effects of the performance of the individual MSR components. Therefore, the test results will describe the performance of either individual MSR components or the entire MSR.

0.3

PTC 1-1991, the Code on General Instructions, should be studied thoroughly before formulating the procedures for testing an MSR. The Code on Definitions and Values, PTC 2-1980 (R1985), defines technical terms and numerical constants which are used throughout this Code. Unless otherwise specified, instrumentation should comply with the appropriate supplements of the PTC 19 Series of codes on Instruments and Apparatus. PTC 6-1976, Steam Turbines, should be consulted for isolation and verification methods.

SECTION 1 — OBJECT AND SCOPE

1.1 OBJECT

This Code provides the procedures, direction, and guidance for the accurate testing of Moisture Separator Reheaters (MSRs) which includes moisture separating and steam reheating components located between the high pressure and low pressure steam turbine. The purpose of the Code is to determine the performance of the MSR and to provide guidance in the evaluation of its performance effect on the turbine cycle heat rate with regard to:

(a) Moisture Separator Outlet Quality;

(b) Reheater Terminal Temperature Difference (TTD) per stage;

(c) Cycle Steam pressure drop across applicable component(s); and

(d) Excess heating steam flow.

1.2 SCOPE

Requirements are specified by this Code for application on MSR testing in the following areas:

(a) Pretest arrangements and agreements;

(b) Instrumentation types and accuracies;

(c) Instrumentation applications and methods of measurement;

(d) Testing and calculational techniques; and

(e) Information contained in the test report.

1.3 EXPECTED MEASUREMENT UNCERTAINTY

By satisfying the instrument accuracy criteria specified in Section 4 and complying with the balance of procedural requirements of this Code, a test will generally provide 95 percent or greater confidence that the measurement of the required performance parameters will yield results for which the bounds of the difference between the final test results and the true value is within $\pm$ 10.0 Btu/kW-hr.

Utilizing techniques specified in PTC 19.1, Measurement Uncertainty, the overall measurement uncertainty is based on the prescribed instrument accuracies and example precision indices for MSR testing. An outline of the calculations conducted to establish the expected overall measurement uncertainty value, noted above, is covered in Appendix B. Users of this Code should determine the quality of a Code test by performing a post test uncertainty analysis utilizing PTC 19.1.

SECTION 2 — DEFINITIONS AND DESCRIPTION OF TERMS

2.1 NOMENCLATURE

Variables used in this Code at MSR test point locations contain multiple terms and subscripts selected from the lists below:

(a) Term 1: Property or Value (capitalized)
- *DP* = Differential Pressure, psi, (kPa)
- *MSE* = Moisture Separation Effectiveness, %
- *H* = Enthalpy, Btu/lbm, (J/kg)
- *M* = Moisture Content, %
- *P* = Pressure, psia, (kPa)
- *PD* = Pressure Drop, psi, (kPa)
- *S* = Entropy, Btu/(lbm R), (J/(kg K))
- *T* = Temperature, °F, (K)
- *TTD* = Terminal Temperature Difference, °F, (K)
- *V* = Specific Volume, ft^3/lbm, (m^3/kg)
- *W* = Mass Flow Rate, lbm/h, (kg/s)
- *X* = Quality, %

(b) Term 2: Component Abbreviation (capitalized)
- *HP* = High Pressure Reheater
- *LP* = Low Pressure Reheater
- *HPT* = High Pressure Turbine
- *LPT* = Low Pressure Turbine
- *MS* = Moisture Separator System
- *SG* = Steam Generator
- *CN* = Condenser

(c) Term 3: Stream Abbreviation (capitalized)
- *CD* = Condensate
- *CS* = Cycle Steam
- *ES* = Excess Steam
- *FW* = Feedwater
- *HS* = Heating Steam
- *P2* = Reheater — 2nd Pass
- *P4* = Reheater — 4th Pass
- *TH* = Throttle

(d) Term 4: Location (optional) (capitalized)
- *I* = Inlet
- *O* = Outlet
- *V* = Vent
- *D* = Drain

(e) Term 5: Condition (lower case)
- *c* = Corrected
- *t* = Test
- *d* = Design
- *sat* = Saturated
- *f* = Saturated Liquid State
- *fg* = Difference between saturated liquid and saturated vapor states
- *g* = Saturated Vapor State
- *avg* = Average

Note: Any term may be followed with an alphanumeric identifier. Lack of an identifier indicates final, total, or average for multiple components. For example, HLPP4V = Enthalpy of LP reheater, fourth pass vent steam.

2.2 DEFINITIONS

Note: This Section provides the definitions for the standard terminology used in this Code. Unless otherwise specified, the definitions of PTC 2-1980 (R1985) apply.

cycle steam — the HP Turbine exhaust steam passing through the MSR shell, delivered to the LP Turbine

excess steam — non-condensing heating steam that clears the reheater of condensate to minimize subcooling, thermal distortion, and slug flow

heat rate, Btu/kW-hr — heat required to generate a unit of electrical energy

heating steam — steam supplied to reheater tubeside for the purpose of transferring its latent heat to the cycle steam

moisture carryover — moisture remaining in the cycle steam after the moisture separation system

moisture separation effectiveness — the ratio of the mass flow rate of moisture removed from the entering cycle steam to the mass flow rate of moisture entering the separator

moisture separator outlet quality, % — the thermodynamic quality of the steam at the outlet of the moisture separation section (expressed as percent).

(MSR) shell — the vessel containing the reheater(s) and the moisture separator section(s)

moisture separator system — the portion of the MSR that removes the moisture from the cycle steam

observation — a single measurement recording

pass (1-4) — the portion of the reheater section with respect to the number of times that heating steam flow is routed across the cycle steam flow path

pressure drop (psi) — the difference in static pressure across the MSR or its component

reference value — established from the design data and test conditions. Utilized in the calculation of MSR performance as the "expected" value to which a test value is compared.

reheater — the tube bundle portion(s) of the MSR used to transfer energy from the heating steam to the cycle steam

terminal temperature difference (°F) — the difference between heating steam saturation temperature at the reheater inlet and the cycle steam outlet temperature

test — a complete set of test runs

test run — a complete set of data taken over a continuous period of time

test value — the representative value of a physical parameter as determined by the utilization of test instrumentation and techniques

SECTION 3 — GUIDING PRINCIPLES

3.1 PREPARATION FOR THE TEST

3.1.1 Pretest Agreements. The parties to the test shall reach agreement on specific test objectives. Initial preparation should include familiarization and examination of the MSR systems and internals and testing apparatus by all parties involved. At least the following items shall be agreed upon prior to the test:

(a) unit operating conditions during the test (e.g., reactor power, steam generator pressure);

(b) reference values (or defining equations) for use in Table 5.1, for calculation of MSR performance;

(c) frequency of observations, method of recording data, number, and duration of test runs;

(d) system alignment and verification during the test;

(e) determination of parameters not measured (e.g., inlet cycle steam moisture content and heating steam inlet quality);

(f) test objectives;

(g) method of comparing test results to performance guarantee(s), including considerations for testing MSRs individually or simultaneously;

(h) provisions for maintaining stable test conditions;

(i) that MSR components, system piping, and internal structures have been installed as required or specified

(j) cleanliness conditions of the MSR (e.g., existence of fouling and debris);

(k) identification of any known damage or deficiency (e.g., missing tubes, plugged tubes, and broken welds);

(l) number, use, installation, and location of temperature, pressure, and flow sensors, including redundant measurements of critical test parameters;

(m) location and use of any station instrumentation for balance of plant (BOP) or auxiliary components testing;

(n) method of determining cycle steam, excess steam and drain flow rates (e.g., sensor design, location);

(o) radioactive tracer application techniques, including location of injection and sample taps;

(p) responsibility for licensing and handling radioactive tracers, if used;

(q) accountability of all extraneous and abnormal flows (see para. 3.3.4, System Alignment Requirements);

(r) adjustment of the excess steam flow rate, if adjustable;

(s) instrument accuracy and calibration;

(t) use of vendor thermal kit or previous precision turbine test data;

(u) time limits (see para. 3.4.5, Calibration of Instrumentation).

3.1.2 Acceptance Test Scheduling

Note: This paragraph may be disregarded if the MSR test is part of the initial turbine acceptance test.

The MSR should be in an as-new condition. An acceptance test should be conducted as soon as practicable but not later than 12 weeks after the initial operation of the new or modified MSR, providing no serious MSR operating difficulty has occurred. If station conditions or licensing limitations make it impossible to conduct the test within the prescribed time frame, then it may have to be postponed until immediately following an internal inspection. In lieu of an internal inspection, the condition may be considered as-new if the MSR's performance does not differ from that determined in the initial benchmark, observed in a trend of the MSR's performance parameters.

3.1.3 Performance Benchmark Determination. A performance benchmark should be established with plant instrumentation immediately after the MSRs are first placed in service at stable unit conditions, so that if the Code test is delayed by more than 12 weeks, there can be reasonable assurance that there has been no change in the MSR performance during the intervening period of operation by comparing measurements. This eliminates the need for an internal inspection prior to Code testing. Required measurements and information include:

(a) reactor power level;

(b) MSR cycle steam outlet temperature;

(c) reheater(s) heating steam pressure(s) and flow(s);

(d) MSR steam outlet pressure or LP Turbine bowl pressure;

(e) MSR cycle steam pressure drop.

3.2 GENERAL TEST REQUIREMENTS

3.2.1 Preliminary Test Runs. Preliminary test runs should be conducted for the purpose of:

(a) checking all instrumentation;

(b) orienting test personnel;

(c) establishing the required duration and frequency of observations for the actual test runs;

(d) making minor operational and test adjustments;

(e) determining whether the MSR(s) and plant are in a suitable condition for the test;

(f) establishing the operating conditions at which to conduct the test;

(g) achieving proper valve and system alignment;

(h) ensuring proper conditions can be met to perform the test;

(i) preliminary determination of test uncertainties;

(j) if tracer is used, determination of the required concentration levels, injection and sampling rates, and equilibrium/lag rates.

3.2.2 Responsibilities of Parties. The responsibilities of the parties to the test are:

(a) to ensure that the test report reflects if alternative methods within the guidelines of this Code are employed;

(b) to designate a mutually acceptable third party to direct and mediate disputes;

(c) to witness the test and verify that it is conducted in accordance with this Code and the pretest agreements. All data logs will be made available to all parties at the conclusion of each test run. Copies of the entire data collection will be distributed upon the conclusion of the test.

3.3 TEST OPERATING CONDITIONS

3.3.1 Operating Conditions. Test runs should be conducted under specified operating conditions, or as close to specified operating conditions as possible, in order to minimize corrections to the test results. The variation of any condition which may influence the results of the test run shall be minimized before the test run begins and maintained so during the test run. (Refer to paras. 3.3.3 and 3.3.4.) The testing should be commenced at 95 percent rated thermal power or above.

The licensee technical specifications, NRC regulations, and turbine manufacturer specifications should not be violated.

3.3.2 Constancy of Test Conditions. Prior to any test run, the unit shall be operated for a sufficient time to attain steady state conditions and shall be kept at steady state throughout the test run. Steady state conditions will have been attained when the unit operating conditions and permissible deviation criteria of Table 3.1 have been met.

When a tracer is used, the injection should commence sufficiently prior to the start of the test run to attain concentration equilibrium. As a guide, it may be conservatively expected that equilibrium is attained when a time period, equal to four times the calculated transport time through both the longest injection line and the longest sample line, has passed following the commencement of injection. All elements of the system (e.g., tanks) shall be considered. Concentration equilibrium is verified when the tracer concentration of two consecutive samples taken during the test run differ by no greater than three percent.

3.3.3 Deviations. Deviations of the variables in excess of the limits prescribed in Table 3.1, or as otherwise agreed upon, may occur during a test run. If such deviations are observed during a test run, the cause shall be eliminated or corrected and the test run continued, if possible, until all variables are within the specified limits for the duration of the test run. The test run may be extended in order to make up for the time and data lost during the correction of the cause of the deviation, but shall not exceed a two-hour total duration. If the cause of the deviations cannot be eliminated or corrected during the test run, or if deviations are discovered during the computation of results from a completed test run, that run shall be rejected in whole, or in part, and repeated as necessary after the cause of the deviations has been eliminated. Any rejected portions of the test run shall not be used in computing the overall averages. The results of that test run may then be acceptable, provided that the remaining valid periods aggregate to one hour or more and the quantity of readings obtained during the valid periods satisfies the criteria of Table 3.1.

TABLE 3.1
PERMISSIBLE DEVIATION OF VARIABLES

System Variable	Unit Operating Conditions	Permissible[1] Deviation During Each Test Run
Unit Conditions		
Thermal power	95% or greater	±0.5%
Main steam or steam generator pressure	±2% of expected	±1%
MSR Measurements		
Heating Steam Flow		±3%
Cycle Steam Outlet Temperature		±2°F
Reheater Drain Temperature		±2°F
Shell Pressure Drop		±5%
Heating Steam Pressure		±2%
Drains Flow		±5%
Excess Steam Flow		±10%
Cycle Steam Pressure		±1%

NOTE:
(1) Each observation of an operating condition during a test run shall not vary from the reported average for that operating condition during the complete run by more than the amount shown, except by mutual agreement between the parties to the test.

3.3.4 System Alignment Requirements

The MSR System should be aligned for normal operation as per plant/vendor procedures. In addition, the following should be addressed:

(a) In order to attain typical flow rates through the MSR shell and reheaters, all extraneous and abnormal flows which may significantly affect cycle steam flow or heating steam flow should be eliminated. Preparations shall be made prior to the test to eliminate or account for any extraneous flows.

(b) System alignment shall be made so that:

(1) no reheater or shell drains or vents are routed to the condenser unless that is their normal designed flow path;

(2) all reheater stop valves shall be fully opened;

(3) all flow element bypasses are isolated if flow element is to be used;

(4) differential pressure water legs shall be measured and compensated for;

(5) any significant abnormal flows to or from the MSR which cannot be isolated are measured, which may include sample flows and leaks to the atmosphere;

(6) all MSR shell and reheater bundle bypasses are isolated;

(7) pressure sensing lines are open, but not flowing;

(8) heating steam inlet check and control valves are fully open.

(c) The system alignment should be outlined and agreed upon by all parties prior to commencing the test. Refer to PTC 6-1976, Steam Turbines, for suggested isolation and isolation verification methods.

3.3.5 Special Test Precautions. Reheater venting and steam admission control during warm-up must follow manufacturer's recommendations and plant operating procedures to avoid structural damage.

3.4 TEST TECHNIQUES

3.4.1 Acceptability of Test Runs. A minimum of two test runs shall be conducted to ensure repeatability. A comparison of the test run results shall meet the following criteria:

(a) TTDs for each MSR should differ no more than 0.7°F;

(b) moisture separator outlet quality for each MSR should differ no more than 0.15 percent;

(c) MSR shell pressure drop for each MSR should differ no more than 0.1 psi;

(d) the total heat rate change due to MSR performance as calculated in Table 5.1 should differ no more than the resultant square root of the sum of the squares of the individual heat rate changes. These heat rate changes should be calculated with the above limits (a-c) as the deviation for each parameter and the unit specific sensitivity of deviation.

If the results of any two or more entire runs meet the criteria, the final test result will be the arithmetic average of the acceptable test runs. The results of any test run which does not meet this criteria shall be discarded.

3.4.2 Frequency of Observations. The minimum frequency of observations required is:

(a) all flow rates every one minute;

(b) temperature and pressure measurements every five minutes;

(c) all other measurements every ten minutes.

Each test run shall commence and end with a measurement (eg., for data requiring a frequency of readings every five minutes, 13 measurements shall be made during a one-hour test run).

3.4.3 Duration of Test Runs. This Code recommends a minimum steady-state test run of one hour duration. In any case, the length of the test period for which the readings are averaged shall be sufficient to reduce the effect of uncertainty of the final results to less than 2 Btu/kW-hr due to data scatter. Data scatter on the preliminary test run shall not produce results which differ by more than 2 Btu/kW-hr. If this is exceeded, the following should be considered:

(a) increase the frequency of readings;

(b) stabilize the unit or MSR test conditions;

(c) enhance the quality of the instrumentation;

(d) investigate the cause of the instability;

(e) verify instrument repeatability;

(f) replace suspect instrument(s).

3.4.4 Timing. Test period and observation times should be consistent. All machines and people recording observations shall commence, sample, and cease simultaneously.

3.4.5 Calibration of Instrumentation. All test instruments shall be calibrated before and after the test. The specific calibration data, duration, and procedure for each instrument shall be made available to the parties to the test. Instruments used for flow, temperature, pressure, differential pressures, and data acquisition shall be calibrated to standards traceable to the standards maintained by the National Institute of Standards and Technology. The calibrations shall be performed under the same conditions that the instrument will be employed in during the test. The instrument, once installed for the test, will be maintained in an artificial atmosphere approximating that in which it was calibrated, or the instrument software will self-compensate for the difference.

A time limit should be established by the parties to the test as to how long and under what conditions the test instruments shall be subjected to before the equipment is recalibrated. This is in the event that plant testing is delayed.

Installation of all test instrumentation shall comply with all applicable criteria of Section 4 of this Code. A few duplicate calibrated instruments should be readily available as spares.

3.4.6 Location of Test Points. Typical test point locations are shown in Fig. 3.1 for a single MSR that has two stages of reheat, each stage containing four passes. The test point locations may be different depending upon the actual MSR configuration, but the points described on Fig. 3.1 are required for an MSR with that configuration. Test instrumentation shall be configured equally on each MSR. The special considerations required in the selection of test instrumentation are described in Section 4 in each respective area.

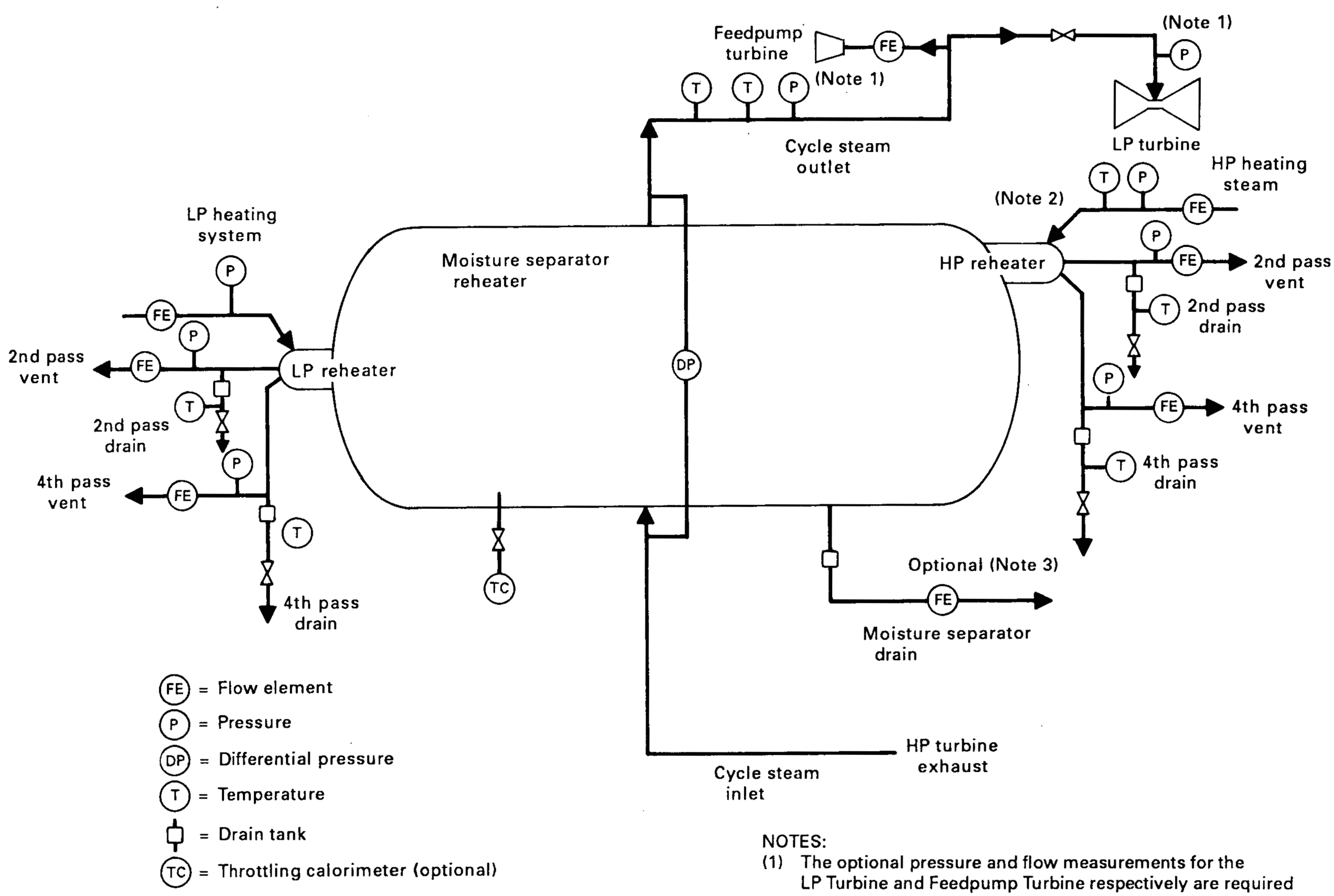
3.4.7 Method of Comparing Test Results. The method of comparing test results to the specified performance shall be agreed upon by all parties prior to the test. The two methods utilized for this comparison are:

3.4.7.1 Individual MSRs. The computed level of performance is compared to the specified basis for each MSR.

3.4.7.2 Lumped MSR System. The computed level of performance of each MSR is summed and an average performance level is compared with the reference values.

The method involved in the computation of the test results is described in Section 5 of this Code.

3.4.8 Thermodynamic Properties. Except with written agreement to the contrary, the "1967 ASME Steam Tables, Thermodynamic and Transport Properties of Steam and Its Enthalpy-Entropy Diagram (Mollier Chart)" shall be used for thermodynamic properties used in the calculation of test results. The 1977 edition of "ASME Steam Tables" should be used for transport properties. Where machine computation is employed, the computer shall be programmed in accordance with the 1967 International Formulation Committee Formulations for Industrial Use, which is included in the Appendix to the "1967 ASME Steam Tables."



NOTES:

(1) The optional pressure and flow measurements for the LP Turbine and Feedpump Turbine respectively are required for the alternate cycle steam flow calculation method using turbine flow factors.

(2) Temperature needed only if superheated.

(3) For use in shell drain flow method of determining moisture separator outlet quality.

FIG. 3.1 TYPICAL TEST POINT LOCATIONS

3.4.9 Tolerances. Tolerances to allow for testing inaccuracies, which might be directly applied to the final test results, are outside the scope of this Code. Such tolerances are chiefly of commercial significance and should be settled by agreement. The test results shall be reported as calculated from test observations with only such corrections as are provided within this Code.

SECTION 4 — INSTRUMENTATION AND METHODS OF MEASUREMENT

4.1 GENERAL CONSIDERATIONS

4.1.1 Introduction. This section presents the requirements for instrument methods, and precautions which shall be employed. The Supplements on Instruments and Apparatus (PTC 19 series) provide additional information concerning instruments and their use and should be consulted for specific details not included in this Code.

4.1.2 Duplicate Instrumentation. This section also specifies duplicate instrumentation for measuring certain parameters that are critical to the test results in order to reduce measurement uncertainty and increase reliability of the data. In addition, redundant instrumentation should be considered to detect trouble with sensors and to reduce measurement uncertainty of data.

4.1.3 Alternative Instrumentation. The parties to the test may agree to use advanced instrument systems, such as those using electronic devices or mass-flow techniques, as alternatives to the instrument requirements specified by this Code, provided that such systems have a demonstrated maximum measurement error equivalent to that required by this Code.

4.2 MEASUREMENT OF PRESSURE

4.2.1 Static Pressure. Static pressure measurements shall be made using calibrated gages or transducers having a maximum measurement error of 0.25 percent of expected reading.

4.2.2 Pressure Taps. The pressures measured in an MSR test are static pressures. Pressure tap holes for measuring such pressures shall be drilled at right angles to the inner surface of the pipe wall. The hole diameter shall be no smaller than 0.25 in. and no larger than 0.50 in. The inner rim of the hole should be free of burrs, leaving its edges sharp and square. The hole shall be straight and of uniform bore for a length of at least twice its diameter.

The pressure taps should be installed in a straight run of pipe as remote as possible from upstream elbows or obstructions. It is recommended that the tap be positioned on the side of a horizontal pipe.

In high velocity regions, abrupt area changes and piping losses should be properly accounted for or shown to be negligible. Typical locations where instruments may be affected are the MSR shell inlet, shell outlet, and heating steam inlet at the reheater hemi-head.

4.2.3 Connecting Piping. Connecting piping shall be not less than 0.375 in. inside diameter, or equivalent tubing, to avoid resistance damping inside the piping. When measuring steam pressure, a method ensuring proper water leg establishment and maintenance should be used. The use of a condensate pot or a two-foot horizontal tubing run are two such methods.

All subsequent tubing shall slope continuously downward to the level of the instrument, to prevent air or water pockets. The pressure measuring instrument should be located as close as possible to the connecting pipe taps in order to minimize water leg corrections. For additional guidance on connecting piping, see PTC 19.2, Pressure Measurement.

4.2.4 Heating Steam Pressure. The heating steam pressure should be measured at the same point from which the reference TTD is based. The pressure tap may be located in the reheater hemi-head. An alternate position for the tap is in a straight section of pipe, as close as practical to the hemi-head and downstream of any valves or orifices.

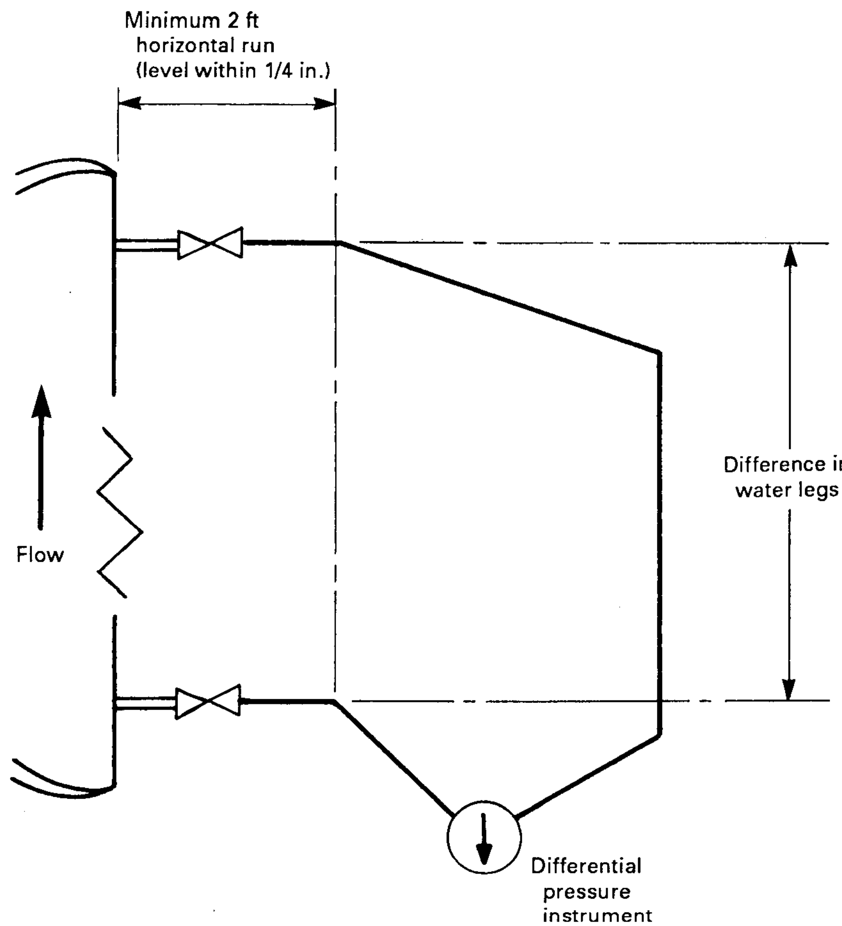


FIG. 4.1 WATER LEG DETERMINATION

4.3 MEASUREMENT OF DIFFERENTIAL PRESSURE

4.3.1 MSR Cycle Steam Pressure Drop. MSR cycle steam pressure drop should be measured using a differential pressure instrument with maximum measurement error of 0.25 percent of expected reading. Extreme care should be exercised in the application of this instrument due to the small pressure drop being measured and the elevation difference of the pressure taps. If nitrogen purge is not used, the static pressures caused by water legs in the sensor tubing connected to the MSR shell inlet and outlet can cause the differential pressure being measured by the differential pressure instrument to be negative. The differential pressure instrument shall be located at an elevation below both pressure taps. The difference in water legs, as shown in Fig. 4.1, should be added to the measured pressure drop.

The components over which the pressure drop is to be measured will be different for each specific application and shall be agreed upon by the parties to the test prior to starting the test. This will determine the locations of the pressure taps for this measurement, which may be in the shell or connecting piping.

For multi-nozzle MSR's, the overall cycle steam pressure drop should be determined by measuring the pressure drop from each inlet nozzle and averaging the results.

4.3.2 Component Modification or Replacement. Since MSR internal components are periodically modified or replaced, the pressure drop across the affected component shall be determined for overall performance computation (see Section 5). The direct measurement of the affected component pressure drop is difficult, but preferred.

An alternate method, in lieu of the component pressure drop measurement, is the measurement of the overall MSR cycle steam pressure drop before and after the modification or replacement. This change in the pressure drop can be used directly in the overall performance computation. This pressure drop is measured by pressure taps located upstream of the MSR in the cycle steam inlet piping and downstream of the MSR in the cycle steam outlet piping. The instrumentation, connections and calibrations should remain unchanged in both tests. The calibration of the instrument shall be checked prior to the latter test but shall not be adjusted if it is still within that required. If the calibration has drifted or if instrumentation failure occurs, another instrument shall be calibrated and used at the same location as the previous instrument. The alternate method is not valid in cases where significant component degradation has occurred prior to replacement or modification.

4.3.3 Accuracy of Flow Determination. Instruments used to measure differential pressure across flow metering devices should have a maximum measurement error of 0.25 percent of expected reading.

4.4 MEASUREMENT OF TEMPERATURE

The steam and condensate temperatures entering and exiting the MSR are required for the reheater energy balance calculation of moisture separator outlet quality and terminal temperature difference.

4.4.1 Instrumentation. Temperature measurements shall be made utilizing calibrated instrumentation having a maximum measurement error of $\pm 1\,°F$.

4.4.1.1 Sensing Instrument Requirements. In measuring temperatures around the MSR, the following fundamental requirements must be met:

(a) sensing instrument inherent bias must be quantified by proper calibration of each instrument;

(b) The sensing instrument or thermowell must be placed in a region which represents the flowing volume to be measured, and away from stagnant areas;

(c) the sensing instrument must not allow for any significant heat conduction through its wires, sheath, and external connections;

(d) avoid sources of radiation and conductive heat transfer.

4.4.2 Installation. If a thermowell is used, an error in measurement can result due to the inherent conduction of heat away from the temperature element by the thermowell material. For this reason, a thermowell of adequate length shall be used in conjunction with a means to ensure that the temperature element is in direct contact with the bottom of the thermowell. The thermowell should be cleaned of all foreign material and oxides.

The external head of the temperature element and pressure boundary couplings shall be completely insulated to minimize heat losses to atmosphere. The insulation thickness shall be at least as thick as the surrounding insulation, and of the same or equivalent thermal resistance value.

4.4.3 Measurement Points

4.4.3.1 Required Measurements. For a reheater energy balance and to calculate TTD, the following temperatures shall be recorded as shown in Fig. 3.1.

4.4.3.2 Cycle Steam Outlet Temperatures. The cycle steam outlet temperature should be measured at cross sections in the outlet pipe(s) at least six feet away from any massive, partly insulated piping structure, such as a valve. The temperature element should be located at least ten feet downstream of the MSR outlet nozzle. For multi-nozzle MSRs, without a common pipe junction, the temperature should be measured in the individual outlet pipes and averaged. At least two temperature elements should be installed in each cross section at approximately 90 degrees separation.

Cycle steam outlet temperature may be determined as the average temperature of several readings from a thermocouple grid, which contains several sensors in representative points above the bundle cross flow area. The thermocouple grid would consist of at least one sensor for each tube support sheet.

4.4.3.3 HP or LP Heating Steam Temperature Entering the Reheater. Measurement of the inlet pressure, from which inlet saturation temperature is determined, is the recommended method for determining heating steam inlet temperature to the reheater used for TTD calculation. This is because of the relative insensitivity of pressure on temperature (typically $1\,°F$ for every 7 psi in MSR heating steam application).

If superheat exists in this stream, a direct measurement of temperature is necessary to determine inlet enthalpy. The thermowell used for this measurement should be located downstream of the control valve as close as practical to the reheater inlet without adversely affecting the pressure or flow measurements.

4.4.3.4 Condensate Temperatures Exiting the Reheater. Condensate temperature should be measured in the drain line or in the respective drain tank, but upstream of any flow or level control valves. This will ensure the measurement of the average temperature of a well mixed condensate.

4.5 MEASUREMENT OF STEAM QUALITY

Saturated steam exists at various locations in the nuclear steam turbine cycle. The quality of the steam can be determined by using the following methods.

4.5.1 Reheater Energy Balance Method. The quality of cycle steam entering a reheater can be determined by the energy balance method which is described in para. 5.4.2.

4.5.2 Throttling Calorimeters. Throttling calorimeters are normally used when insufficient or inaccurate data precludes use of the reheater energy balance method or when a non-reheat cycle is to be tested.

The steam sample probe(s) should be installed at the outlet of the moisture separator system and located to obtain a representative sample. The probes should also be designed to obtain an isokinetic sample.

To measure the source pressure use the calorimeter sample line at a static flow condition.

The calorimeter temperature should be measured with a maximum error of $\pm 0.5°F$. If the calorimeter test temperature does not exceed the saturation temperature at atmospheric pressure, an expansion to a pressure below atmospheric is necessary. In order to maintain the throttling calorimeter as an adiabatic system, the calorimeter sample line and throttling calorimeter shall be extremely well insulated, and the sample line length should be minimized.

4.5.3 Constant Rate Tracer Injection. The mass flow rate of liquid in a two phase flow can be determined by the constant tracer injection method which is described in paras. 4.6.5.1, 4.6.5.2, and 4.6.5.3. The basic principle of this tracer application is that the tracer is soluble in liquid and essentially insoluble in the steam vapor.

The sampling flow rate must be set and maintained so that vapor entrainment and its consequent concentration does not occur in the sampling lines. The sampling flow rate may be set by increasing the sampling flow rate until the tracer concentration decreases dramatically, which indicates the dilution of the sample with entrained and condensed steam containing no tracer. Mechanical float or density type meters are another method used to set and maintain the correct sampling flow rate. The prevention of steam vapor entrainment is essential for an accurate determination of the liquid portion of the total steam flow.

The steam quality can then be determined from the ratio of the liquid flow rate to the total steam flow rate.

4.6 FLOW RATE DETERMINATIONS

ASME Supplement 19.5, Part 2 of "Fluid Meters-1971" provides detailed information relative to most of the flow techniques and flow elements herein recommended for this Code. The equations for the calculation of discharge coefficients for orifices, flow nozzles, and Venturi meters contained in the supplement have been superseded by those given in "Measurement of Fluid Flow in Pipes Using Orifice, Nozzles and Venturi: ASME MFC-3M-1989, and should be used. Some copies of the ASME 1971 supplement contain errata sheets showing some of the new equations for discharge coefficients. To avoid confusion as to which should be used, the following pertinent equations and their limits are given:

(a) Orifices: (Beta ratio between 0.20 and 0.75)
Pipe Reynolds number (R_D between 10^4 and 10^8)
Flange Taps
Pipe inside diameter D between 2.0 in. and 2.3 in.

$$C = 0.5959 + 0.0312\beta^{2.1} - 0.1840\beta^8 + \frac{0.390\beta^4}{1-\beta^4} - \frac{0.0337\beta^3}{D} + \frac{91.71\beta^{5/2}}{R_D^{3/4}} \qquad \text{(a)}$$

Pipe inside diameter D greater than 2.3 in.

$$C = 0.5959 + 0.0312\beta^{2.1} - 0.1840\beta^8 + \frac{0.0900\beta^4}{D(1-\beta^4)} - \frac{0.0037\beta^3}{D} + \frac{91.71\beta^{5/2}}{R_D^{3/4}} \qquad \text{(b)}$$

D and $D/2$ Taps (all pipe sizes)

$$C = 0.5959 + 0.0312\beta^{2.1} - 0.1840\beta^8 + \frac{0.390\beta^4}{1-\beta^4} - 0.01584\beta^3 + \frac{91.71\beta^{5/2}}{R_D^{3/4}} \qquad \text{(c)}$$

(b) Flow Nozzles
Wall Taps (Beta ratios between 0.20 and 0.70)
Throat Reynolds numbers (R_d from 10^4 to 10^6)

$$C = 0.9975 - \frac{6.53}{R_d^{1/2}} \tag{d}$$

For throat Reynolds numbers greater than 10^6

$$C = 0.9975 - \frac{0.1035}{R_d^{1/5}} \tag{e}$$

4.6.1 Cycle Steam Flow. The Final Feedwater Flow Method and the Flow Factor Method, which are discussed below, are the recommended methods for determining the total cycle steam flow through all the MSRs. In actual practice, cycle steam vapor and water flows are not equally distributed among all the MSRs; for the purposes of these calculations it is assumed that they are equally distributed. This necessary assumption permits results within an acceptable range of uncertainty.

4.6.1.1 Final Feedwater Flow Method (Preferred). The determination of the total cycle steam flow through all the MSRs is the cumulation of several flow measurements and the application of several known steam cycle parameters.

The basis of this cycle steam flow determination is the accurate measurement of final feedwater flow rate. All nuclear units are required to accurately determine reactor power via a heat balance, and therefore most have high quality venturi type flow elements installed in the final feedwater piping. The cycle steam flow measurement uncertainty will be reduced if this primary flow element is inspected prior to the test. Guidance for evaluating the measurement uncertainty of flow elements is given in PTC 6 Report on Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines. If a new feedwater flow element is being purchased, requirements for a high accuracy flow section are provided in detail in PTC 6.1-1984. Use of a high accuracy flow element will reduce the test measurement uncertainty but it is not required to achieve a cycle steam flow uncertainty of two percent.

During the test runs, steam generator blowdown should be isolated or measured. For BWR units, control rod drive flow shall be measured.

During the test runs, ensure that the high pressure portion of the turbine cycle is operating at or very near design/expected levels. This includes all feedwater heaters that are associated with the high pressure turbine, the feedwater pump turbines, and steam seal systems. Also ensure that the condition and performance of these systems remain constant through the test periods.

In order to determine cycle steam flow via a final feedwater flow measurement, all extraction and leakoff flows between the feedwater venturis and MSR outlet shall be accounted for. Feedwater heater extraction flows should be calculated by performing a heat balance calculation around each heater. Station instrumentation can be used to measure all conditions except extraction enthalpy for which the design or test values may be employed. Any other extraction or leakoff flow that is less than five percent of final feedwater flow should be taken from the best available source, i.e., vendor thermal kits, past precision turbine tests, or direct measurement.

Any extraction or leakoff flow that is five percent or greater of final feedwater flow shall be measured directly. Measurement techniques must be adequate to determine the flow within two percent. Heating steam flow and MSR drain flows independent of their size with respect to feedwater flow shall be measured utilizing the methods in paras. 4.6.2 through 4.6.5.

4.6.1.2 Flow Factor Method (Alternate). Total LP turbine inlet flow can be used to calculate the LP turbine bowl conditions and the fixed LP turbine flow factor. The flow factor (K_d) should be calculated using previous test data, or if not available, design should be used, in the following.

$$K_d = \frac{W_d}{\sqrt{P_d/V_d}}$$

The actual LP turbine inlet flow is calculated by inserting measured inlet pressure and specific volume.

$$W_t = K_d\sqrt{P_t/V_t}$$

Cycle steam outlet flow through the MSR shell can then be calculated by adding any measured flows routed to the feedpump turbines or other components. With this technique, measurement of moisture separator drain flow is unnecessary for the calculation of cycle steam outlet flow.

Due to the mixing of hot reheat streams, this procedure generally must use the average LP turbine bowl conditions and will yield only the total MSR cycle steam flow.

The accuracy of this method is dependent on the measurement uncertainty of the pressure and temperature ahead of the LP turbine but is most dependent on the accuracy of the turbine flow factor. The turbine flow factor is expected to be within five percent of the design value. However, the repeatability of the flow factor could be ± one percent provided the condition of the LP steam path has not changed. This method can be used to determine the cycle steam flow to within two percent provided data from a previously conducted turbine performance test is available to establish the turbine flow factor.

4.6.2 Heating Steam Flow. The heating steam flow rate may be determined by the measurement and addition of flows exiting the reheater or by direct measurement. The choice of method(s) should result in a flow measurement uncertainty not exceeding ± five percent.

The direct heating steam flow measurement is dependent upon the steam properties.

For two phase flow, a sharp edged orifice and the James equation[1] should be used subject to the beta ratio limitations given in para. 4.6.

$$W = 1890.07 \frac{CYd^2F_a}{\sqrt{1-\beta^4}} \left[\frac{\Delta P}{X^{1.5}(V_g - V_f) + V_f} \right]^{1/2}$$

where:

W = Flow rate, lbm/hr

C = Discharge coefficient calculated from equations (a), (b), or (c) of 4.6, as applicable

Y = Empirical expansion factor

$$Y = 1 - (0.41 + 0.35\beta^4)\left(\frac{\Delta P}{kp_1}\right)$$

p_1 = Total or stagnation pressure

k = Ratio of specific heats of a gas (ideal)

d = Throat diameter, in.

F_a = Orifice thermal expansion factor (refer to Fluid Meters, 1971)

β = Ratio of orifice diameter to inside pipe diameter

ΔP = Differential pressure across the orifice (psi)

X = Inlet quality, fraction

V_g = Specific volume of vapor phase (ft^3/lb)

V_f = Specific volume of liquid phase (ft^3/lb)

[1] James, Russell, Metering of Steam-Water Two Phase Flow by Sharp Edged Orifices, Proc. Instr. Mechanical Engineers, 1965-1966, Vol. 180, Pt. 1, No. 23, pp. 549-566

For superheated heating steam flow, a flow nozzle or orifice and the standard flow equation from PTC 19.5 Part 2-1971 should be used.

$$W = 1890.07 \frac{CYd^2F_a}{\sqrt{1-\beta^4}} \left[\frac{\Delta P}{V} \right]^{1/2}$$

A flow nozzle is the recommended flow element when the heating steam is superheated, in order to reduce the head loss and resulting heating steam temperature drop.

An alternative method is to calculate the heating steam flow by the addition of measured reheater drain and excess steam flows. This method could yield the least measurement uncertainty if it meets the conditions of paras. 4.6.4 and 4.6.5.

These measurements should be conducted with great care because the heating steam flow measurement is the largest contributor to the overall test uncertainty. A reasonable comparison to reheater drain flow is expected.

4.6.3 Excess Steam Flow. The measurement of excess steam flow may be conducted by conventional methods as outlined in para. 4.6.2, which require the installation of a flow element. An alternate method is to measure the pressure of the excess steam at the inlet to any flow control device which has a critical pressure ratio. The excess steam flow will be sonic at the throat and can be calculated from the upstream conditions and the throat area.

Another method is to use the design flow, in which case excess steam flow is no longer considered an MSR performance parameter. This method should only be considered if there is a reasonable assurance that the actual flow rate is close to the design value. The excess steam flow rate is still required for calculation of other MSR parameters and reference values.

The parties to the test must first consider the effect of this method in the determination of moisture separator outlet quality. Only after a thorough analysis and appreciation of the effect should this alternative be pursued.

4.6.4 Drain Flow Measurements Using Conventional Flow Elements. If full pipe flow exists in the moisture separator and reheater drains then a flow element can be used for the measurement of the flow. If the MSR drain system uses self-venting lines, flow elements cannot be used. A flow element between a drain tank and its level control valve is acceptable, provided the physics of cavitation is considered in its sizing and placement. Paragraph 4.58 of PTC 6-1976 recommends a large length of vertical piping extended from the drain to the flow element to avoid cavitation difficulties. The installation of a test flow element utilizing ultrasonic pulse generators and receiving transducers is acceptable when the total flow measurement uncertainty is $\pm$ two percent or less. If the addition of such piping or test flow elements is impractical, tracer techniques for liquid flow measurement should be employed.

4.6.5 Drain Flow Measurements Using Tracer Techniques. With reference to Fig. 3.1, if it is desired to measure the drain and vent flows around the MSR, tracer techniques may be used. This application involves the measurement of both all liquid flow (shell drains and HP and LP 2nd pass drains) and the steam quality of a two phase flow (HP and LP 4th pass vent and drains). For this purpose, the constant rate injection method is well suited.

4.6.5.1 Constant Rate Injection Method. A water-soluble tracer of concentration (C_{inj}) is injected at a constant rate (w_{inj}) into the water flow or vapor-water flow, as the case may be.

The concentration (C_w) is measured in the water phase downstream of the injection point after adequate mixing has taken place. For this condition, the following material balance can be written:

$$wC_o + w_{inj}\, C_{inj} = (w + w_{inj} + \Delta w)\, C_w$$

or

$$w = \frac{w_{inj}\,(C_{inj} - C_w) - \Delta_w C_w}{C_w - C_o}$$

where:

w = mass-flow rate of water in vapor-water mixture

C_o = initial concentration in the water-phase at the sampling point, before injection starts, due to natural amounts of tracer (background concentration)

Δw = change in water flow (condensation of water vapor due to injection of the cold-tracer solution).

In the cases where $C_w << C_{inj}$, $C_o << C_w$, and $\Delta w << w$ the above equation is reduced to:

$$w = w_{inj}\,(C_{inj}/C_w)$$

which gives the mass flow rate of water in the vapor mixture at the sampling point. If the moisture content in the steam is very low, then Δw is not negligible as compared to w, and the simplified equation can not be used.

Measuring flow rate and concentration of the tracer solution and maintaining a constant injection rate is comparatively simple. However, the tracer concentration in the water phase downstream of the point of injection can be accurately determined only if the tracer is well mixed and a representative sample of the liquid-phase can be obtained.

4.6.5.2 Injection and Sampling Requirements

(a) Injection Points. For the sample to be truly representative, the tracer must be homogeneously distributed in the water phase. Therefore, the injection point should be located after the drain tanks if the piping geometry allows adequate mixing. If not, it can be located before the drain tanks taking into account another volume that enlarges the time period equilibrium. The sample points should be located downstream of the tanks in either case (reference Figs. 4.2, 4.3, and 4.4). A long run of pipe with several elbows will promote mixing. Use of a spray nozzle for injecting may or may not be necessary.

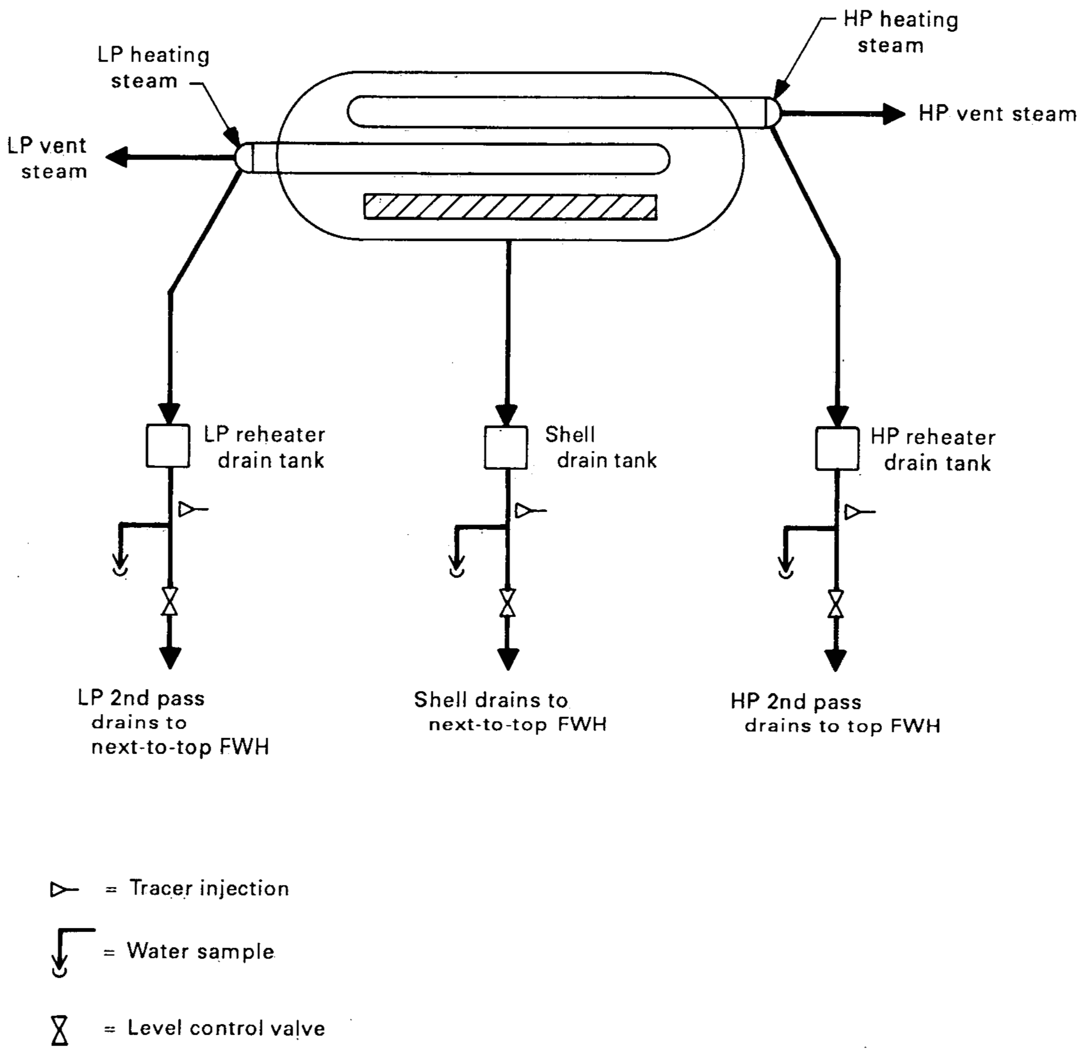
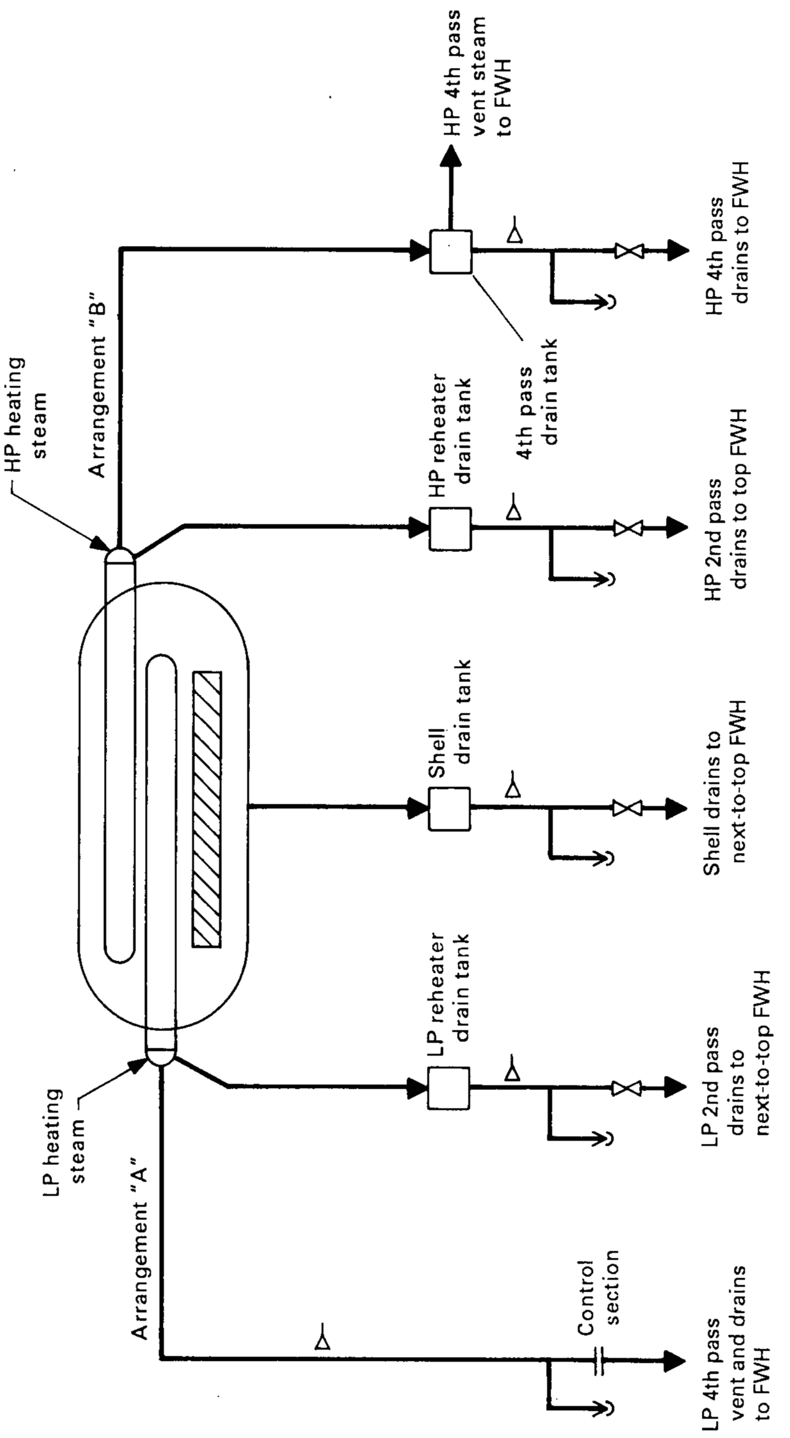


FIG. 4.2 INJECTION AND SAMPLING POINT LOCATIONS
MSR Two-Pass Arrangement



= Tracer injection

= Water sample

= Level control valve

NOTE:
4th pass arrangements "A" and "B" may apply to either HP or LP reheater bundles.

FIG. 4.3 INJECTION AND SAMPLING POINT LOCATIONS
MSR Four-Pass Arrangement

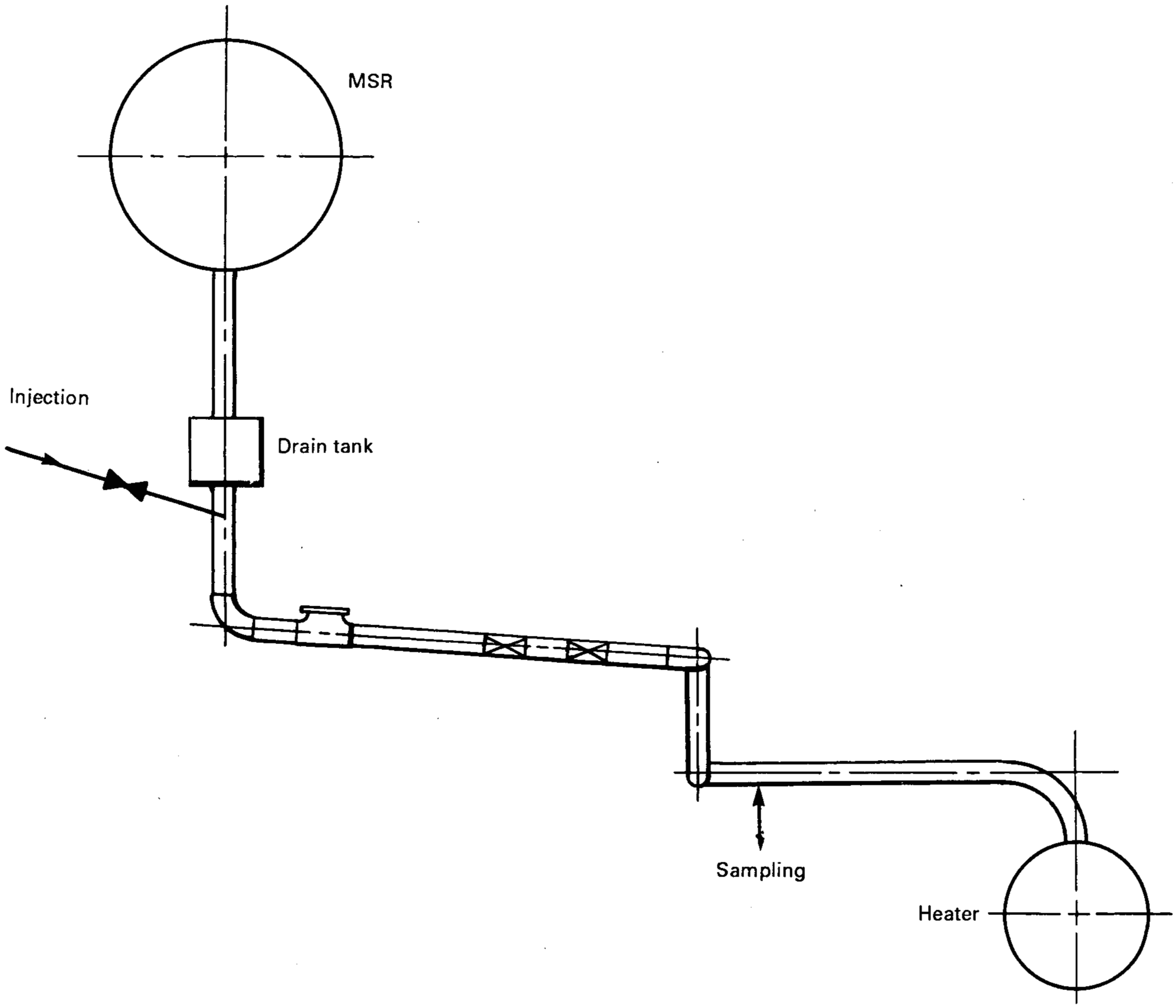


FIG. 4.4 TYPICAL INSTALLATION OF INJECTION AND SAMPLING POINTS

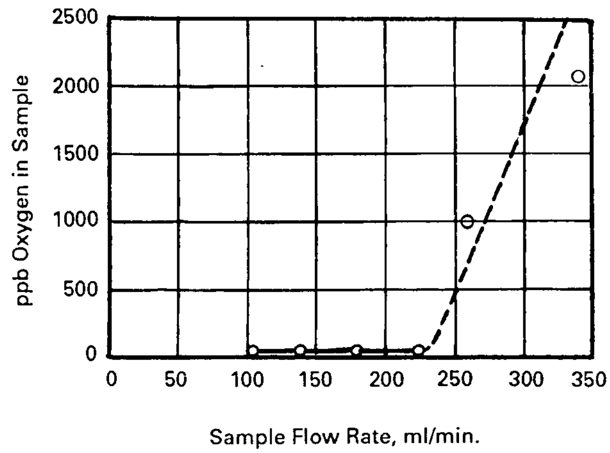


FIG. 4.5 OXYGEN CONTENT OF SAMPLE

(b) *Sampling Points.* Since any condensed vapor in the sample will falsify the result, care must be taken in selecting the location of the sampling tap. At the conditions and velocities normally found in the two-phase drain lines, the water is not homogeneously distributed over the cross section, but concentrated toward the pipe wall. This is a favorable distribution for water sampling, and a simple wall tap should prove satisfactory. However, advantage should be taken of gravitational or centrifugal forces by locating the tap on the bottom side of the pipe or on the outer radius at the exit of an elbow, and dead-flow regions should be avoided. The sample point must be located upstream of any level control valves. Figure 4.4 shows a typical installation of injection and sampling points.

(c) Sampling-Flow Rates. The sampling-flow rate must be adjusted so that entrainment and subsequent condensation of vapor is prevented. The maximum allowable sampling rate in BWR systems can be determined, for example, by analyzing the sampling stream for dissolved oxygen. Oxygen (20 to 30 ppm) is naturally present in the steam of boiling-water reactors as a result of radiolysis. The sampling-flow rate shall be determined prior to the conduct of the test. The oxygen content shall be measured for various sampling-flow rates and plotted, as shown in Fig. 4.5. The flow rate at which steam starts to entrain in the sample is evidenced by a sharp rise in oxygen content. The validity of using oxygen or other suitable tracer as a means of tracing the vapor fraction is based on the distribution of oxygen between the liquid and vapor phases. At pressures less than 500 psia (3450 kPa), oxygen is almost entirely in the vapor phase. In pressurized-water reactor plants which have no oxygen naturally present in the steam, the maximum sample flow rates can be determined by using a suitable tracer of low water solubility, such as Xenon-133, and measuring its concentration in the sample for various sample flow rates. When the Xenon-133 concentration increases, steam is being dragged in with the water sample.

Another method is to inject Sodium-24, a water soluble tracer, and measure its concentration in the sample. When the Sodium-24 concentration starts to decrease, steam is being condensed with the water sample.

4.6.5.3 Requirements for Tracers. For these methods to give accurate results, tracers shall meet these criteria:

(a) soluble in water but essentially insoluble in steam (less than 0.1 percent at the test steam conditions)

(b) nonvolatile

(c) stable at the conditions existing in the turbine cycle

(d) not absorbed on internal surfaces (provided that the water is not evaporated completely)

(e) mixed completely and homogeneously with all the water available at any instant

The choice of the tracer is determined by the criteria mentioned in the foregoing, plus considerations of the effects of the tracer on materials in the cycle, and possible hazards to operating personnel. Radioactive tracers are particularly well suited for application in nuclear power plants, where licensing requirements for possession and handling of radioactive materials present no particular problem. Tracer concentrations of one part per billion can be accurately measured using gamma-counting techniques. For steam cycles with very low radioactive background, the tracer activity concentration required for accurate testing is very small. However, the tracer should be a short lived isotope to eliminate long-term contamination problems. Since it is not practical to measure the concentration of the samples simultaneously, it will be necessary to apply a correction to the measured concentration of each sample to account for isotope decay. One of the tracers which meets these criteria is Sodium-24 with a 15-hour half-life. Flow rates measured with radioactive tracer should be considered to have an uncertainty of $\pm$ one percent.

4.7 MEASUREMENT OF WATER LEVELS

Measurement of water levels during the test is not a requirement of this Code (see para. 3.4.6); however, loss of water level or flooding of either a reheater drain tank or moisture separator drain tank during the test could introduce error into the performance evaluation. It is, therefore, recommended that drain tank level indicators be monitored once every 10 minutes to verify that the water level is maintained within the maximum and minimum normal operating range. Incremental level measurements are unnecessary.

SECTION 5 — COMPUTATION OF RESULTS

5.1 MSR PERFORMANCE COMPUTATIONS

Four factors contribute to determining the overall thermal hydraulic performance of an MSR and its effect on plant heat rate. The two most important performance parameters on heat rate effect are the pressure drop of the cycle steam as it passes through the MSR, and the moisture content, or quality, of the cycle steam leaving the moisture separator system. The other two, terminal temperature difference (TTD) and excess heating steam flow, have less effect on heat rate. TTD is traditionally a measure of a reheater tube bundle's heat transfer performance. It is desirable to have low cycle steam pressure drop, low moisture content in the cycle steam exiting the separator, and low TTD to optimize the benefit of an MSR's effect on plant heat rate. Also, excess steam flow should be held at as low a value as practical consistent with reheater design requirements to minimize a negative effect on plant heat rate.

5.2 COMPONENT PRESSURE DROP

MSR shell side pressure drop should be measured directly as a differential pressure with appropriate corrections taken for water legs.

5.3 TERMINAL TEMPERATURE DIFFERENCE

5.3.1 Introduction. After moisture separation and removal, another function of the MSR is to reheat the main cycle steam, performed in either one or two stages, before it enters the LP turbine. The method for evaluating the performance of this reheating function is by determination of terminal temperature difference (TTD) for each stage.

5.3.2 Calculation of HP Reheater TTD

$$TTDHP = THPHSI_{sat} - TCSO$$

where:

TTDHP = Terminal temperature difference for HP reheater, °F

$THPHSI_{sat}$ = Saturation temperature of heating steam entering HP reheater, °F

TCSO = Temperature of cycle steam at MSR outlet, °F

5.3.3 Calculation of LP Reheater TTD (If applicable)

$$TTDLP = TLPHSI_{sat} - TLPCSO$$

where:

TTDLP = Terminal temperature difference for LP reheater, °F

$TLPHSI_{sat}$ = Saturation temperature of heating steam entering LP reheater, °F

TLPCSO = Temperature of cycle steam at LP reheater outlet, °F

and where, TLPCSO is calculated as follows: Using the thermal energy balance method in para. 5.4.2, determine the cycle steam interstage enthalpy. Determine the temperature, TLPCSO, as a function of the cycle steam interstage pressure and enthalpy.

5.3.4 Adjustment of TTD Reference Values. The effect of turbine cycle parameters on TTD must be accounted for in order to correctly determine the reheater thermal performance capabilities. The reference values for TTD may be adjusted for the following parameters:

- heating steam pressure
- heating steam flow
- moisture separator outlet quality
- cycle steam pressure
- cycle steam flow

It is the responsibility of the parties involved to obtain a curve or mathematical relationship describing the effect of these parameters on TTD. It is possible that an acceptable range of a parameter may exist in which no adjustment is necessary.

5.4 MOISTURE SEPARATOR OUTLET QUALITY

5.4.1 Introduction. Another significant parameter involved in the MSR performance computations is the moisture separator outlet quality. As moisture separator outlet quality decreases, increasing amounts of reheater tube bundle surface area are required to evaporate or "boil off" the moisture exiting the separator. This results in a reduction in the effective heat transfer surface area of the reheater available for superheating the cycle steam, which further results in increased reheater TTDs and increased heating steam flow.

The moisture separator outlet quality can be measured directly by the methods described in paras. 4.5.1 and 4.5.2, or it can be calculated by the reheater energy balance method, described below.

5.4.2 Reheater Energy Balance Method. A calculation governed by the first law of thermodynamics, an energy balance, should be conducted around each reheater stage to determine the specific enthalpy of the cycle steam entering each reheater stage. The interstage cycle steam specific enthalpy is required for the determination of LP reheater TTD. All flows entering or exiting each reheater stage must be accounted for in this calculation. The specific enthalpy calculated is also equal to that of the moisture separator outlet cycle steam. The moisture separator outlet quality can then be determined from the specific enthalpy, pressure, and steam tables.

The reheater energy balance is applicable for single or two stage MSRs and allows for variations in excess steam and drain configurations.

5.4.3 Calorimeter Method. A throttling calorimeter uses an adiabatic expansion and the measurement of steam temperature and pressures to determine enthalpy and quality. (See para. 4.5 for application techniques.)

Moisture separator outlet quality is determined by the following equation:

$$XMSO = \left(\frac{H_2 - H_f}{H_{fg}}\right) 100$$

where:

$XMSO$ = moisture separator outlet quality, percent

H_2 = enthalpy of superheated steam at calorimeter pressure and temperature, Btu/lb

H_f = enthalpy of saturated liquid in mixture prior to throttling, Btu/lb

H_{fg} = enthalpy of vaporization corresponding to pressure or temperature of steam entering calorimeter, Btu/lb

Reference: ASME PTC 19.11-1970, Water and Steam in the Power Cycle

5.4.4 Differential Tracer Method. The steam quality cannot be directly measured on the short flow path from the moisture separator to the reheater bundle. The carried over moisture flow rate is determined with a two step method. In the first step the moisture separator shell drain flow (WMSD) is determined by the conventional tracer method. Then in the second step the tracer is injected in the cycle steam inlet pipe and the carried over moisture flow rate is determined by:

$$WCO = \frac{W_{inj}\,C_{inj2}}{C_{WMSD2}} - \frac{W_{inj}C_{inj1}}{C_{WMSD1}}$$

where the indexes 1 and 2 indicate the first and second step.

The moisture separator outlet quality is determined by:

$$XMSO = \left(\frac{WCSI_g}{WCSI_g + WCO}\right) 100$$

5.4.5 Shell Drain Flow Method. As an alternative method of moisture separator outlet quality determination, the total cycle steam flow at the inlet to the moisture separator ($WCSI_f$ and $WCSI_g$) may be determined from thermal kit information provided by the turbine manufacturer, and the moisture removed (WMSD) by the separator determined by direct measurement of shell drain flow.

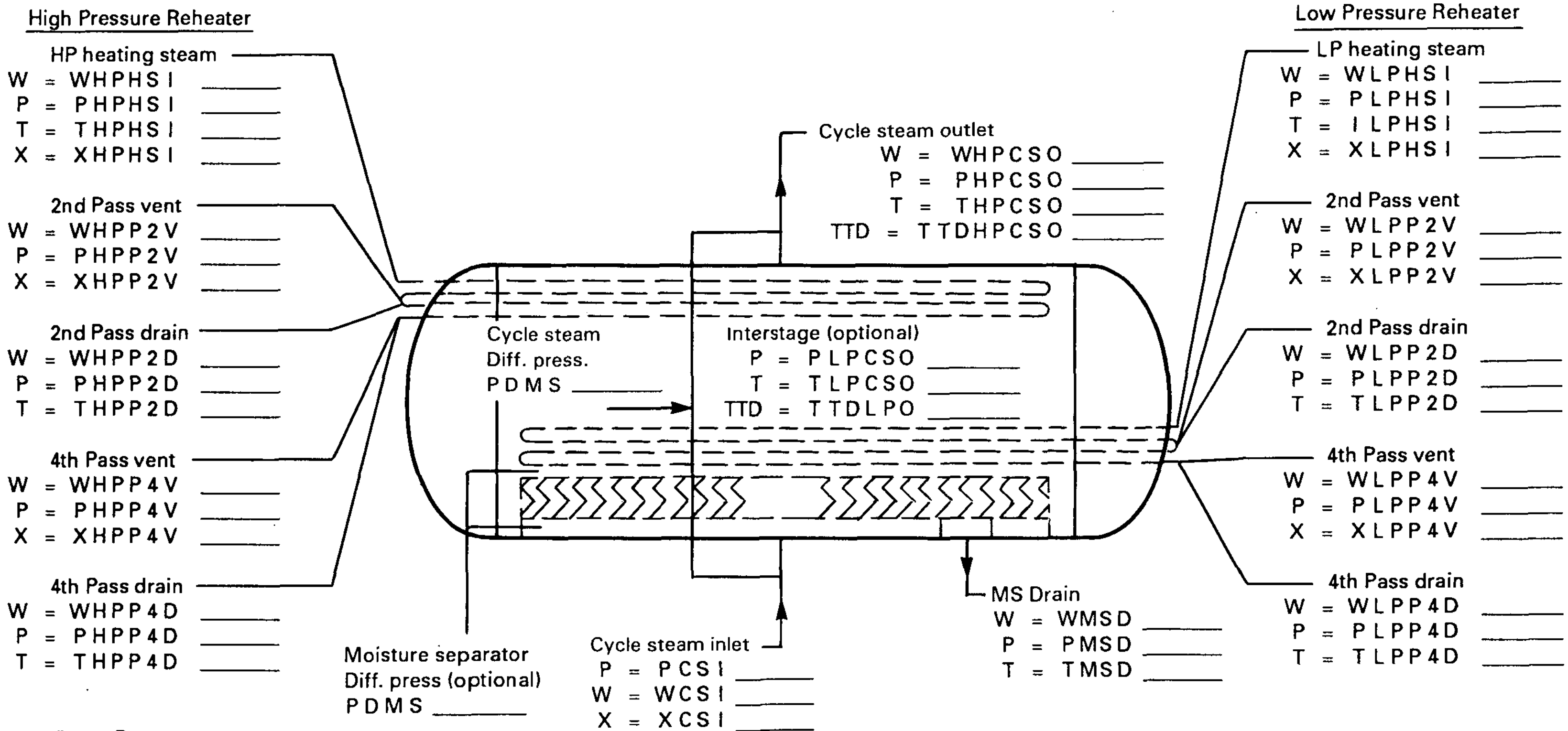
The moisture separator outlet quality is then expressed as:

$$XMSO = \left(\frac{WCSI_g}{WCSI - WMSD}\right) 100$$

where:

$XMSO$ = moisture separator outlet quality, percent

$WCSI_g$ = vapor portion of the cycle steam entering the moisture separator



P = Pressure
T = Temperature
W = Flow
X = Quality
TTD = Terminal temperature difference
PD = Pressure drop

GENERAL NOTES:

(1) Heating steam quality may be taken from heat balances or derived from other test data.
(2) This drawing is a typical representation. Variations and additional data may be required based on individual applications.
(3) Data measured calculated, or reference value agreed to by the parties to the test.

FIG. 5.1 MOISTURE SEPARATOR REHEATER
Typical Data Point Locations

TABLE 5.1
MSR PERFORMANCE COMPUTATIONS

Performance Parameter	Test Value	Reference Value **[Note (4)]**	Deviation	Sensitivity of Deviation **[Note (1)]**	Heat Rate Change (Btu/kW-hr)
Shell Side Pressure Drop **[Note (2)]**					
Terminal Temperatue Difference **[Note (3)]**					
Moisture Separator Outlet Quality					
Excess Steam Flow					
				Total Heat Rate Change ________	(Btu/kW-hr)

NOTES:
(1) From station specific curves, which must be agreed upon by the parties to the test before commencing the test.
(2) This parameter may be separated into the individual components, i.e., separator, tube bundle, vessel, upon specific test objectives.
(3) This parameter may be utilized per stage if required.
(4) Established in accordance with para. 5.5. These values (or the method of computation of the values) must be agreed to by the parties of the test before commencing the test. Specific reference values may be a function of test conditions i.e., unit load, specific flow rates or may be constants.

$WCSI$ = total inlet cycle steam flow
$WMSD$ = moisture separator drain flow

5.5 REFERENCE VALUES

Reference values are used in the calculation of overall MSR performance. Test values are compared to the reference values in order to calculate a deviation from which a heat rate change can be determined.

It is a shared responsibility of the parties to the test to agree upon the establishment of the reference values for a test. The specific values may be set constants, design parameters, expected values, or determined as functions of other test conditions.

5.6 SENSITIVITY OF DEVIATION FROM THE REFERENCE VALUE

The equations, contents, or graphical relationships that describe the sensitivity of the deviation of the MSR performance parameter from the reference value shall be developed and agreed upon before the test commences. In the development of these relationships, the effect on turbine control valve position (which may significantly affect the sensitivity of the deviation on heat rate) must be taken into account with respect to variations in the heating steam flow rate.

SECTION 6 — TEST REPORT

6.1 INTRODUCTION

This outline provides guidance for the reporting of tests on MSRs. Only the relevant items need apply in any particular case.

6.1.1 Brief Summary of Test

(a) Owner
(b) Designation of unit
(c) Name and location of plant
(d) MSR manufacturer
(e) Turbine manufacturer
(f) Object of test
(g) Date of test
(h) Brief report of test results and conclusions. A tabular or graphical presentation may be used to present essential findings.
(i) Brief history of operation of unit and MSRs since initial startup.

6.1.2 Discussion of Test

(a) Test procedure, or outline/summary of method
(b) Data acquisition methods
(c) Instrumentation summary
(d) System alignment procedures
(e) Methods of flow measurement
(f) Calculation of moisture separator outlet quality
(g) Any other pertinent information
(h) Test personnel, their affiliations, and their involvement in the test, i.e. Name/Company/Job Designator

6.1.3 Tabulation of Test Conditions

(a) Run number
(b) Brief description
(c) Date
(d) Time

6.1.4 Tabulation of Test Observations for each MSR shell and reheater stage (after application of all calibration corrections)

(a) Cycle steam outlet pressure
(b) Cycle steam outlet temperature
(c) Cycle steam flow
(d) Shell pressure drop
(e) Heating steam pressure
(f) Heating steam temperature
(g) Heating steam flow
(h) Reheater drain pressure
(i) Reheater drain temperature
(j) Reheater terminal temperature difference
(k) Reheater tube side pressure drop
(l) Moisture separator outlet quality
(m) Excess steam flow

6.1.5 Tabulation of Unit Operating Conditions

(a) Reactor thermal power level
(b) Gross generator output
(c) Condenser pressure

6.1.6 Reference Operating Conditions for each MSR shell and reheater stage

(a) Heating steam pressure
(b) Heating steam temperature (or moisture content, if saturated)
(c) Cycle steam inlet pressure
(d) Cycle steam inlet moisture content
(e) Cycle steam flow

6.1.7 MSR Performance Computations

(a) Tabulation of MSR performance computations including effects on heat rate
(b) Supporting calculations for reference values

6.1.8 Graphical Presentation (all values corrected to specified operating conditions)

(a) Correction factors, if determined by test
(b) Any other pertinent data

6.1.9 Overall Uncertainty of Test Results

APPENDICES

Figures

Tables

APPENDIX A — SAMPLE CALCULATION

A.1 TEST DATA (See Fig. A.1)

Feedwater Flow, W_{fw}	=	11467300 lb/hr
HP Reheater Heating Steam Flow, WHPHS	=	485200 lb/hr
LP Reheater Heating Steam Flow, WLPHS	=	265300 lb/hr
Feedpump Turbine Extraction Flow, WFPT	=	132521 lb/hr
Condensate Flow, WCOND	=	8353990 lb/hr
Steam Generator Shell Pressure, PSG	=	820 psia
Feedpump Suction Pressure, PFPI	=	250 psia
Feedpump Discharge Pressure, PFPO	=	1015 psia
HP Reheater Heating Steam Inlet Pressure, PHPHSI	=	776 psia
LP Reheater Heating Steam Inlet Pressure, PLPHSI	=	382 psia
MSR Cycle Steam Inlet Pressure, PCSI	=	216.0 psia
LP Turbine Inlet Pressure, PLPTI	=	194.0 psia
Throttle Pressure, PTH	=	785 psia
Heater 5 Shell Pressure, PH5S	=	195 psia
Heater 6 Shell Pressure, PH6S	=	365 psia
MSR Cycle Steam Outlet Temperature, TCSO	=	462.1°F
Heater 5 Condensate Inlet Temperature, TH5CI	=	301.6°F
Heater 5 Condensate Outlet Temperature, TH5CO	=	378.9°F
Heater 5 Drain Temperature, TH5D	=	310.0°F
Heater 6 Condensate Inlet Temperature, TH6CI	=	385.2°F
Heater 6 Condensate Outlet Temperature, TH6CO	=	429.5°F
Heater 6 Drain Temperature, TH5D	=	395.2°F
MSR HP Reheater Excess Steam Flow, WHPES	=	11834 lb/hr
MSR LP Reheater Excess Steam Flow, WLPES	=	7727 lb/hr

A.1.1 Design Data (See Fig. A.2)

A.2 CALCULATION OF THERMAL POWER

(a) Calculate Main Steam Enthalpy

Test Steam Generator Pressure	=	850 psia (PSGO)
Design Steam Generator Moisture Carryover	=	0.25%

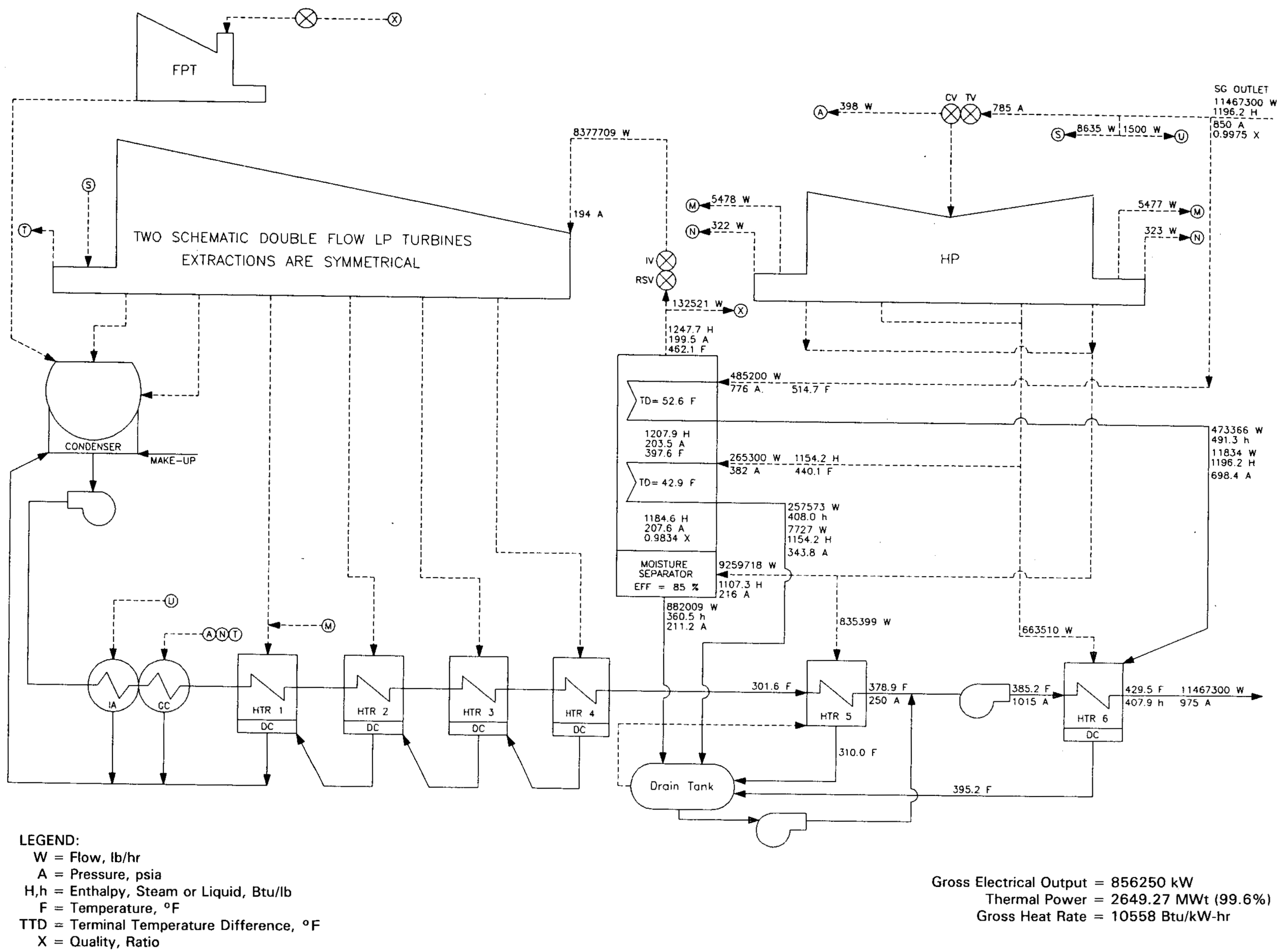


FIG. A.1 TEST CASE

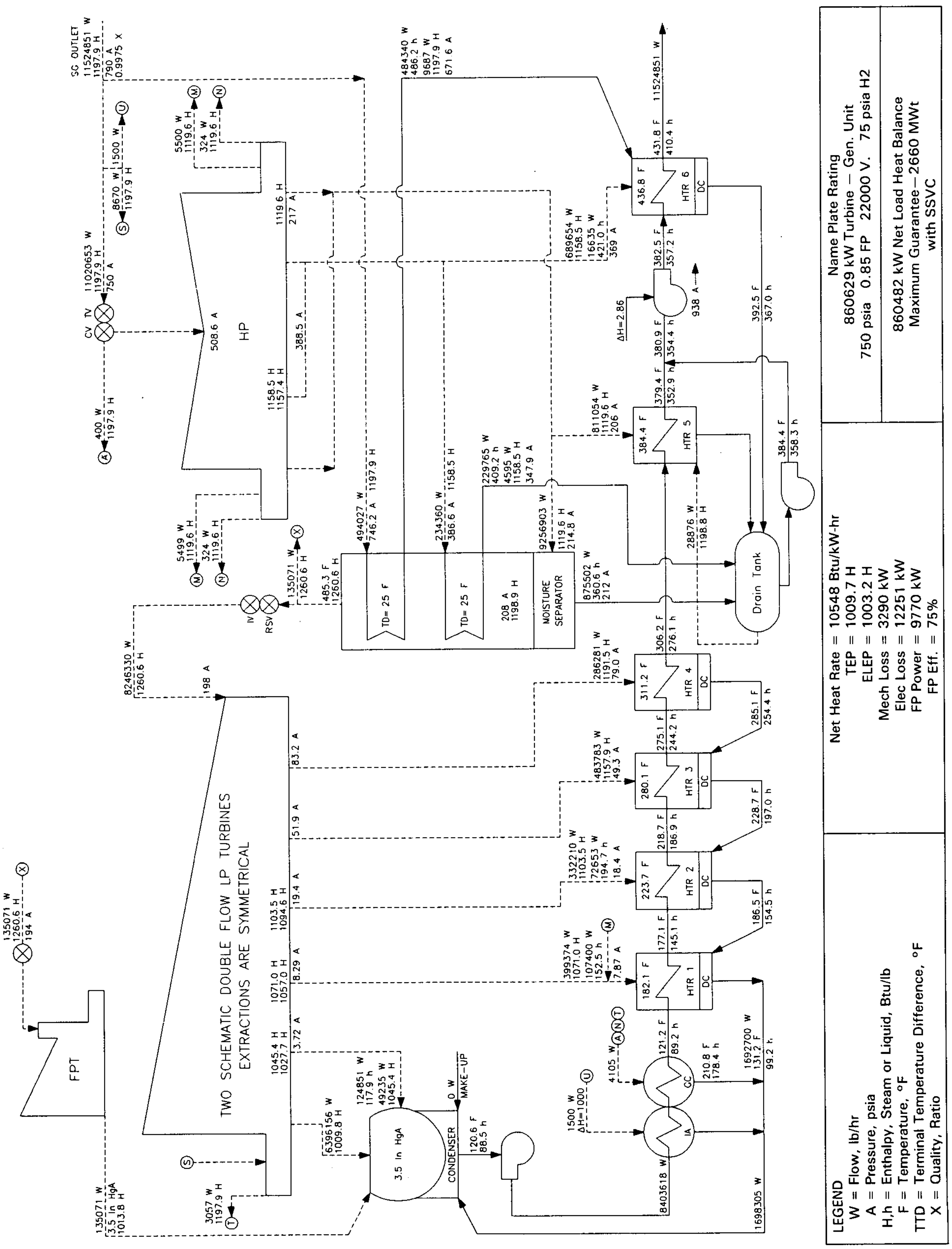


FIG. A.2 DESIGN CASE

therefore, from Steam Tables:

Throttle Enthalpy	=	1196.2 Btu/lb (HTH)

(b) *Calculate Feedwater Enthalpy*

Feedpump Discharge Pressure	=	1015 psia (PFPO)
Assumed Heater 6 Pressure Drop	=	40 psia
Estimated Feedwater Pressure	=	975 psia
Feedwater Temperature	=	429.5°F (TH6CO)

therefore, from Steam Tables:

Feedwater Enthalpy	=	407.9 Btu/lb (HFW)

(c) *Calculate Thermal Power*

$$
\begin{aligned}
\text{QSG} &= \text{WFW} \times (\text{HTH} - \text{HFW}) \\
&= 11467300 \times (1196.2 - 407.9) \\
&= 9.03967\text{E}9 \text{ Btu/hr} \times 1 / 3412141.63 \text{ Btu/MWt-hr} \\
&= 2649.27 \text{ MWt}
\end{aligned}
$$

Note: Steam Generator Blowdown is isolated.

(d) *Percent of Rated Thermal Power*

$$\text{QTP} = (2649.27 / 2660) \times 100 = 99.60\%$$

A.3 DETERMINATION OF MSR CYCLE STEAM OUTLET FLOW

A.3.1 Using Feedwater Flow Method. 5% of Final FW Flow = 573365 lb/hr, so both HP and LP heating steam flows and the HP Turbine Extraction Steam flows must be determined by testing, or directly measured. All other flow rates will be taken from the best available source.

(a) *Determine Heater 6 Extraction Flow*

(1) Use design HP Turbine extraction enthalpy, HH6SI = 1158.5 Btu/lb.

(2) Use the "ASME Steam Tables" for the following enthalpies:

HP Reheater Drain Enthalpy, HHPHSD	=	h (698.4 psia, sat liq) = 491.3 Btu/lb
HP Excess Steam Enthalpy, HHPES	=	HHPHSI = 1196.2 Btu/lb
Heater 6 Drain Enthalpy, HH6D	=	h (365 psia, 395.2°F) = 370.1 Btu/lb
Heater 6 FW Inlet Enthalpy, HH6FW1	=	h (1015 psia, 382.5°F) = 356.14 Btu/lb
Final FW enthalpy	=	Heater 6 FW Outlet Enthalpy, HFW = h (975 psia, 429.5°F) = 407.9 Btu/lb

(3) Conduct an energy balance around heater 6 and solve for extraction steam flow, WH6X

$$
\begin{aligned}
\text{WH6X} &= [\text{WFW (HFW} - \text{HH6FWI}) + \text{WHPHS} \times \text{HH6D} - \text{WHPES} \times \text{HHPES} - (\text{WHPHS} - \text{WHPES}) \times \text{HHPHSD}] / (\text{HH6SI} - \text{HH6D}) \\
&= [11467300\ (407.9-356.14) + 485200 \times 370.1 - 11834 \times 1196.2 - (485200 - 11834) \times 491.3] / (1158.5 - 370.1) \\
&= 667681 \text{ lb/hr}
\end{aligned}
$$

(b) *Determine heater 5 extraction flow*

(1) Using a 20 psi pressure drop through the tubes

$$
\begin{aligned}
\text{PH5CI} &= \text{PFPS} + 20 \\
&= 250 + 20 \\
&= 270 \text{ psia}
\end{aligned}
$$

(2) Estimate Extraction Inlet Enthalpy.

Test MSR Cycle Steam Inlet Pressure, PCSI = 216.0 psia.

Using design 1% pressure drop from the HP turbine exhaust to the MSR cycle steam inlet

HP Turbine Exhaust Pressure, PHPTO = 216.0 / (1 − 1% / 100) = 218.2 psia.

From the design HP turbine end point curve at 218.2 psia

HP Turbine Exhaust Enthalpy, HHPTO = 1107.3 Btu/lb.

Therefore, the Heater 5 Inlet Extraction Enthalpy,

HH5SI = 1107.3 Btu/lb.

(3) From "The ASME Steam Tables":

Heater 5 Condensate Inlet Enthalpy, HH5CI = h (270 psia, 301.6°F) 271.75 Btu/lb

Heater 5 Condensate Outlet Enthalpy, HH5CO = h (250 psia, 378.9 °F) = 352.5 Btu/lb

Heater 5 Drain Enthalpy, HH5D = h (195 psia, 310.0 °F) = 280.3 Btu/lb

LP Reheater Drain Enthalpy, HLPHSD = h (343.8 psia, sat liq) = 408.0 Btu/lb

LP Excess Steam Enthalpy, HLPES = HLPHSI equal to LP heating steam enthalpy = 1154.2 Btu/lb

(4) Conduct an energy balance around heater 5 and solve for extraction steam flow, WH5X.

WH5X = [WCOND (HH5CO − HH5CI) + WLPHS × HH5D − WLPES × HLPES − WLPHSD × HLPHSD] / (HH5SI − HH5D)

= [8353990 (352.5 − 271.75) + 265300 × 280.3 − 7727×1154.2 − 257573 × 408.0] / (1107.3 − 280.3)

= 767790 lb/hr

(c) Determine Cycle Steam Flow Rate

(1) Estimate Leakage Flows.

Assume all Leakage Flows are proportional to Thermal Power.

Leakage S, WLS = 8635 lb/hr

Leakage A, WLA = 398 lb/hr

Leakage M, WLM = 10955 lb/hr

Leakage N, WLN = 645 lb/hr

(2) Use Design SJAE Flow.

WSJAE = 1500 lb/hr

(3) Determine MSR Cycle Steam Inlet Flow.

WCSI = WFW − WHPHS − WLPHS − WLS − WLA − WLM − WLN − WSJAE − WH5X − WH6X

= 11467300 − 484200 − 265300 − 8635 − 398 − 10430 − 645 − 1500 − 767793 − 667681

= 9259718 lb/hr

(4) Cycle Steam Flow Determination Uncertainty Calculation. (See Table A.1.)

TABLE A.1
CYCLE STEAM FLOW DETERMINATION UNCERTAINTY CALCULATION

Flow Path	Flow Rate (lb/hr)	Individual Measurement Uncertainty (%)	Mass Flow Uncertainty (lb/hr)
Feedwater	11467300	±1.2	±137607.6
HP Heating Steam	485200	±5.0	±24260.0
LP Heating Steam	265300	±5.0	±13265.0
LPT Seal Steam	8635	±5.1	±431.8
CV Leakoff	398	±5.0	±19.9
HPT Seal Leakoff	10430	±5.0	±521.5
HPT Gland Leakoff	645	±5.0	±32.3
SJAE Supply	1500	±5.0	±75.0
Heater 5 Extraction	767793	±2.0	±15355.9
Heater 6 Extraction	667681	±2.0	±13353.6
Square Root Sum of the Squares			±141827.23
Inlet Cycle Steam	9259718		
Calculated Uncertainty		1.53%	

GENERAL NOTE: The resultant uncertainty is less than the 2.0% maximum required by code.

(5) Calculate MSR Shell Drain Flow Rate. (No measured value is available for this case.)

(a) For the first iteration use an 85% MS Effectiveness, MSE = 85%

(b) HP Turbine Exhaust Enthalpy. From Design HP Turbine End Point Curve:

$$\text{HCSI} = 1107.3 \text{ Btu/lb}$$

(c) Calculate Moisture Content Entering MSR Shell.

From Steam Tables:

$$\text{At PCSI} = 216.0 \text{ psia and HCSI} = 1107.3 \text{ Btu/hr}$$
$$\text{XCSI} = 0.89$$

therefore,

$$\begin{aligned} \text{MCSI} &= (1 - \text{XCSI}) \times \text{WCSI} \\ &= (1 - 0.89) \times 9259718 \\ &= 1018569 \text{ lb/hr} \end{aligned}$$

(d) MSR Shell Drain Flow

$$\begin{aligned} \text{WMSD} &= (\text{MSE} / 100) \times \text{MCSI} \\ &= (85 / 100) \times 1018569 \\ &= 865784 \text{ lb/hr} \end{aligned}$$

(6) MSR Cycle Steam Outlet Flow

$$\begin{aligned} \text{WCSO} &= \text{WCSI} - \text{WMSD} \\ &= 9259718 - 865784 \\ &= 8393934 \text{ lb/hr} \end{aligned}$$

A.3.2 Calculate MSR Cycle Steam Outlet Flow using LP Turbine Inlet Flow Factor method as a validation step:

(a) Calculate Design LP Turbine Inlet Flow Factor

Design LP Turbine Inlet Pressure, PLPTID = 198 psia
Design LP Turbine Inlet Enthalpy, HLPTID = 1260.6 Btu/lb

From Steam Tables:

Design LP Turbine Inlet Specific Volume, VLPTID = 2.6992 ft^3/lb

$$KLPT = WLPTI / \sqrt{PLPTI / VLPTI}$$
$$= 8246330 / \sqrt{198 / 2.6992}$$
$$= 962821$$

(b) *Calculate LP Turbine Inlet Conditions*

Test LP Turbine Inlet Pressure, PLPTI = 194.0 psia
Test MSR Cycle Steam Outlet Pressure, PCSO = 199.5 psia
Test MSR Cycle Steam Outlet Temperature, TCSO = 462.1°F

From Steam Tables:

MSR Cycle Steam Outlet Enthalpy, HCS0 = h (199.5 psia, 462.1°F) = 1247.7 Btu/lb
LP Turbine Inlet Enthalpy, HLPTI = HCSO = 1247.7 Btu/lb

From Steam Tables:

LP Turbine Inlet Specific Vol., VLPTI = 2.6724 ft3/lb

(c) *LP Turbine Inlet Flow*

$$WLPTI = KLPT \sqrt{PLPTI / VLPTI}$$
$$= 962821 \sqrt{194 / 2.6724}$$
$$= 8203434 \text{ lb/hr}$$

(d) *Calculate MSR Cycle Steam Outlet Flow by adding FW Pump Turbine drive steam*

$$WCSO = WLPTI + WFPT$$
$$= 8203434 + 160320$$
$$= 8363754 \text{ lb/hr}$$

The resultant flow is 0.36% less than the flow Determined via feedwater flow method, which is well within the expected error band, validating the first value.

A.4 CALCULATION OF MSR PRESSURE DROPS

A.4.1 Design Pressure Drop

Design MSR Shell Inlet Pressure, PCSID = $217 \times 0.99 = 214.8$ psia
Design LP Turbine Inlet Pressure, PLPTID = 198.0 psia

$$PDCSD = (PCSID - PLPTID) / PCSID \times 100$$
$$= (214.8 - 198.0) / 214.8 \times 100$$
$$= 7.82\%$$

A.4.2 Expected Pressure Drop

$$PDCSE = PDCSD = 7.82\%$$

A.4.3 Test Pressure Drop

MSR Shell Inlet Pressure, PCSI = 216.0 psia
LP Turbine Inlet Pressure, PLPTI = 194.0 psia

$$PDCS = (PCSI - PLPTI) / PCSI \times 100$$
$$= (216 - 194) / 216 \times 100$$
$$= 10.19\%$$

A.4.4 Estimate Intermediate Pressures

(a) Pressure Drop Ratios:

Ratio = (PDCS / PDCSD)
= 10.19 / 7.82
= 1.303

(b) MS Drain Pressure:

Design Pressure Drop, PDMSDD = 1.7%
PDMSD = PDMSDD × Ratio
= 1.7 × 1.303
= 2.22%
PDMSD = (100 − PDMSD) / 100 × PCSI
= (100 − 2.22) / 100 × 216
= 211.2 psia

(c) MS Outlet Pressure:

Design Pressure Drop from CS Inlet, PDMSOD = 3.0%
PDMSO = PDMSOD × Ratio
= 3.0 × 1.303
= 3.91%

PMSO = (100 − PDMSO) / 100 × PCSI
= (100 − 3.91) / 100 × 216
= 207.6 psia

(d) MSR Reheater Interstage Pressure:

Design Pressure Drop, PDLPCSD = 1.5%
PDLPCS = PDLPCSD × Ratio
= 1.5 × 1.303
= 1.95%

PLPCSO = (100 − PDLPCS) / 100 × PMSO
= (100 − 1.95) / 100 × 207.6
= 203.5 psia

PHPCSI = PLPCSO
= 203.5 psia

(e) MSR Shell Outlet Pressure:

Design Pressure Drop, PDHPCSD = 1.5%
PDHPCS = PDHPCSD × Ratio
= 1.5 × 1.303
= 1.95%

PCSO = (100 − PDHPCS) / 100 × PHPCSI
= (100 − 1.95) / 100 × 203.5
= 199.5 psia

(f) HP Reheater Drain Pressure:

Design Pressure Drop, PDHPHSD = 10%
PHPHSD = (100 − PDHPHSD) / 100 × PHPHSI
= (100 − 10) / 100 × 776.0
= 698.4 psia

(g) LP Reheater Drain Pressure:

Design Pressure Drop, PDLPHSD	=	10%
PLPHSD	=	(100 − PDLPHSD) / 100 × PLPHSI
	=	(100 − 10) / 100 × 382.0
	=	343.8 psia

A.5 CALCULATION OF HP REHEATER TTD

A.5.1 Calculate Heating Steam Temperature

HP Reheater HS Inlet Pressure, PHPHSI = 776.0 psia

Note: Assume no pressure drop across reheater inlet flange.

From Steam Tables:

THPHSI = 514.7°F

A.5.2 Terminal Temperature Difference

TTDHP	=	THPHSI − TCSO
	=	514.7 − 462.1
	=	52.6°F

A.6 CALCULATION OF LP REHEATER TTD

A.6.1 Calculate Heating Steam Temperature

LP Reheater HS Inlet Pressure, PLPHSI = 382 psia

Note: Assume no pressure drop across reheater inlet flange.

From Steam Tables:

TLPHSI = 440.1°F

A.6.2 Calculate HP Reheater CS Inlet Enthalpy by Energy Balance

(a) HP Reheater Heating Steam Inlet Enthalpy:

Assume zero heat loss from Throttle:

HHPHSI	=	HTH
	=	1196.2 Btu/lb

(b) HP Reheater Drain Enthalpy:

Assume Saturated Liquid at Drain Pressure:

From Steam Tables, at PHPHSD	=	698.4 psia:
HHPHSD	=	491.3 Btu/lb

(c) Excess Steam Flow:

Assume Excess Steam passes through at heating steam inlet enthalpy:

HHPES	=	HHPHSI
	=	1196.2 Btu/lb

(d) HP Reheater Inlet Enthalpy:

Heat Loss by Heating Steam	=	Heat Gain by Cycle Steam
(WHPHS − WHPES) × (HHPHSI − HHPHSD)	=	WCSO × (HCSO − HHPCSI)

Solving for HHPCSI:

$$
\begin{aligned}
HHPCSI &= HCSO - [(WHPHS - WHPES) \times (HHPHSI - HHPHSD) / WCSO] \\
&= 1247.7 - [(485200 - 11834) \times (1196.2 - 491.3) / 8393934] \\
&= 1208.0 \text{ Btu/lb}
\end{aligned}
$$

A.6.3 Calculate LP Reheater Cycle Steam Outlet Temp

$$
\begin{aligned}
HLPCSO &= HHPCSI \\
&= 1208.0 \text{ Btu/lb}
\end{aligned}
$$

From Steam Tables, at HLPCSO = 1208.0 and PLPCSO = 203.5:

$$TLPCSO = 397.8°F$$

A.6.4 Terminal Temperature Difference

$$
\begin{aligned}
TTDLP &= TLPHSI - TLPCSO \\
&= 440.1 - 397.8 \\
&= 42.3°F
\end{aligned}
$$

A.7 CALCULATION OF MOISTURE SEPARATOR OUTLET QUALITY

A.7.1 Calculate LP Reheater CS Inlet Enthalpy by Energy Balance

(a) Estimate LP Reheater Heating Steam Inlet Enthalpy:

LP Reheater Inlet Pressure, PLPHSI = 382.0 psia

Design Press. Drop from No.1 Extraction, PDX1D = 0.5%

Estimated No.1 Extraction Pressure, PX1 = PLPHSI / (100 − PX1D) × 100

= 382 / (100-0.5) × 100

= 382.9 psia

From Turbine Design Curve, at PX1 = 382.9 psia:

HX1 = 1154.2 Btu/lb

Assuming Zero Heat Loss from HP Extraction

$$
\begin{aligned}
HLPHSI &= HX1 \\
&= 1154.2 \text{ Btu/lb}
\end{aligned}
$$

(b) LP Reheater Drain Enthalpy:

Assume Saturated Liquid at Drain Pressure:

From Steam Tables, at PLPHSD = 343.8 psia

HLPHSD = 408.0 Btu/lb

(c) Excess Steam Flow:

Assume Excess Steam Blows through at heating steam inlet enthalpy:

$$
\begin{aligned}
HLPES &= HLPHSI \\
&= 1154.2 \text{ Btu/lb}
\end{aligned}
$$

(d) CS LP Reheater Inlet Enthalpy:

Heat Loss by Heating Steam = Heat Gain by Cycle Steam

$$(WLPHS - WLPES) \times (HLPHSI - HLPHSD) = WCS0\ (HLPCSO - HLPCSI)$$

Solving for HLPCSI:

$$\begin{aligned}
\text{HLPCSI} &= \text{HLPCSO} - [(\text{WLPHS} - \text{WLPES}) \times (\text{HLPHSI} \\
&\quad - \text{HLPHSD}) / \text{WCSO}] \\
&= 1208.0 - [(265300 - 7727) \times (1154.2 - \\
&\quad 408.0) / 8393934] \\
&= 1185.1 \text{ Btu/lb} \\
\text{HMSO} &= \text{HLPCSI} \\
&= 1185.1 \text{ Btu/lb}
\end{aligned}$$

A.7.2 Calculate Moisture Separator Outlet Quality

$$\begin{aligned}
\text{From Steam Tables, at PMSO} &= 207.6,\ \text{HMSO} = 1185.1: \\
\text{XMSO} &= 0.9837
\end{aligned}$$

A.7.3 Calculation Validation Step. The initial iteration used an estimated MS effectiveness which corresponded to an XMSO of:

$$\begin{aligned}
\text{XMSO} &= 1 - (\text{MCSO} - \text{WMSD}) / \text{WCSO} \\
&= 1 - (1018569 - 865784) / 8393934 \\
&= 0.9820
\end{aligned}$$

A.7.4 The difference between the initial XMSO estimate used to determine cycle steam flow and the calculated result is large enough to warrant a second iteration to determine the values for LP TTD and XMSO.

$$\begin{aligned}
\text{With XMSO} &= 0.9837 = 1 - \text{MCSO/WCSO} \\
&= 0.9837 = 1 - (\text{MCSO} \times \text{Carryover}) / (\text{WCSI} \\
&\quad - \text{MCSI} + \text{MCSI} \times \text{Carryover})
\end{aligned}$$

Solving for carryover and resubstituting:

$$\text{WCSO} = 8377709 \text{ lb/hr}$$

A.7.5 Recalculate LP Reheater TTD

$$\begin{aligned}
\text{HHPCSI} &= \text{HSCO} - [(\text{WHPHS} - \text{WHPES}) \times (\text{HHPHSI} \\
&\quad - \text{HHPHSD}) / \text{WCSO}] \\
&= 1247.7 - [(485200 - 11834) \times (1196.2 - \\
&\quad 491.3) / 8377709] \\
&= 1207.9 \text{ Btu/lb} \\
\text{From Steam Tables TLPCSO} &= \text{T}(1207.9 \text{ Btu/lb, } 203.5 \text{ psia}) \\
&= 397.6^\circ\text{F} \\
\text{TTDLP} &= \text{TLPHSI} - \text{TLPCSO} \\
&= 440.1 - 397.6 \\
&= 42.5^\circ\text{F}
\end{aligned}$$

A.7.6 Recalculate Moisture Separator Outlet Quality

$$\begin{aligned}
\text{HLPCSI} &= \text{HLPCS0} - [(\text{WLPHS} - \text{WLPES}) \times (\text{HLPHSI} \\
&\quad - \text{HLPHSD}) / \text{WCSO}] \\
&= 1207.9 - [(265300 - 7727) \times (1154.2 - \\
&\quad 408.0) / 8377709] \\
&= 1184.9 \text{ Btu/lb} \\
\text{From Steam Tables XMS0} &= \text{X} (1184.9 \text{ Btu/lb, } 207.5 \text{ psia}) \\
&= 0.9834
\end{aligned}$$

This value is 0.0003 less than the value determined in the previous iteration, which is acceptable. No further iterations are necessary.

TABLE A.2
MSR PERFORMANCE EFFECT ON HEAT RATE

Parameter	Design	Reference	Test	%ΔHRr	%ΔHRt	%ΔHR	ΔHR
Shell DP (%)	7.82	7.82	10.19	0	0.32	0.32	33.8
LP Reheater TTD (°F)	25.0	27.9	42.5	0.02	0.154	0.134	14.1
HP Reheater TTD (°F)	25.0	26.4	52.6	0.012	0.198	0.186	19.6
Moisture Separator Outlet Quality (%)	100	100	98.34	0	0.252	0.252	26.6
LP Reheater ES Flow (% of HS)	2.0	2.0	2.91	0	0.003	0.003	0.3
HP Reheater ES Flow (% of HS)	2.0	2.0	2.91	0	0.005	0.005	0.5
Total Heat Rate Change (Btu/kW-hr)							94.9

A.8 CALCULATE HEAT RATE EFFECTS OF MSR PERFORMANCE AND TABULATE IN TABLE A.2

A.8.1 Obtain expected TTD's from Manufacturer's curves at XMSO = 98.34%:

$$\begin{aligned} \text{TTDLPE} &= 27.9°\text{F (from Fig. A.3)} \\ \text{TTDHPE} &= 26.4°\text{F (from Fig. A.4)} \end{aligned}$$

A.8.2 Assume expected equals design performance for remaining parameters

A.8.3 Obtain $\%\Delta HR_r$ for reference values from sensitivity curves (Figs. A.5–A.10)

A.8.4 Obtain $\%\Delta HR_t$ for test values from sensitivity curves (Figs. A.5–A.10)

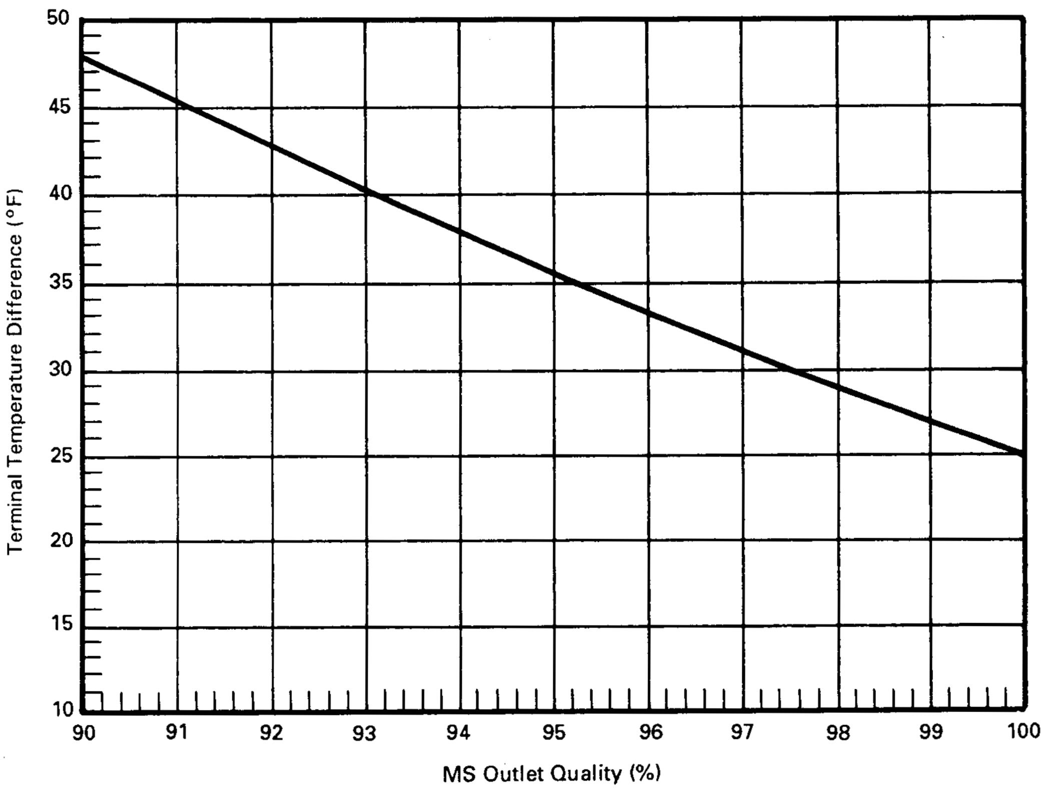
A.8.5 Calculate $\%\Delta HR = \%\Delta HR_t - \%\Delta HR_r$

A.8.6 Calculate $\Delta HR = \%\Delta HR \times HR_d$

$$\begin{aligned} \text{Where } HR_d &= \text{Design Heat Rate} \\ &= 10548 \text{ Btu/kW-hr} \end{aligned}$$

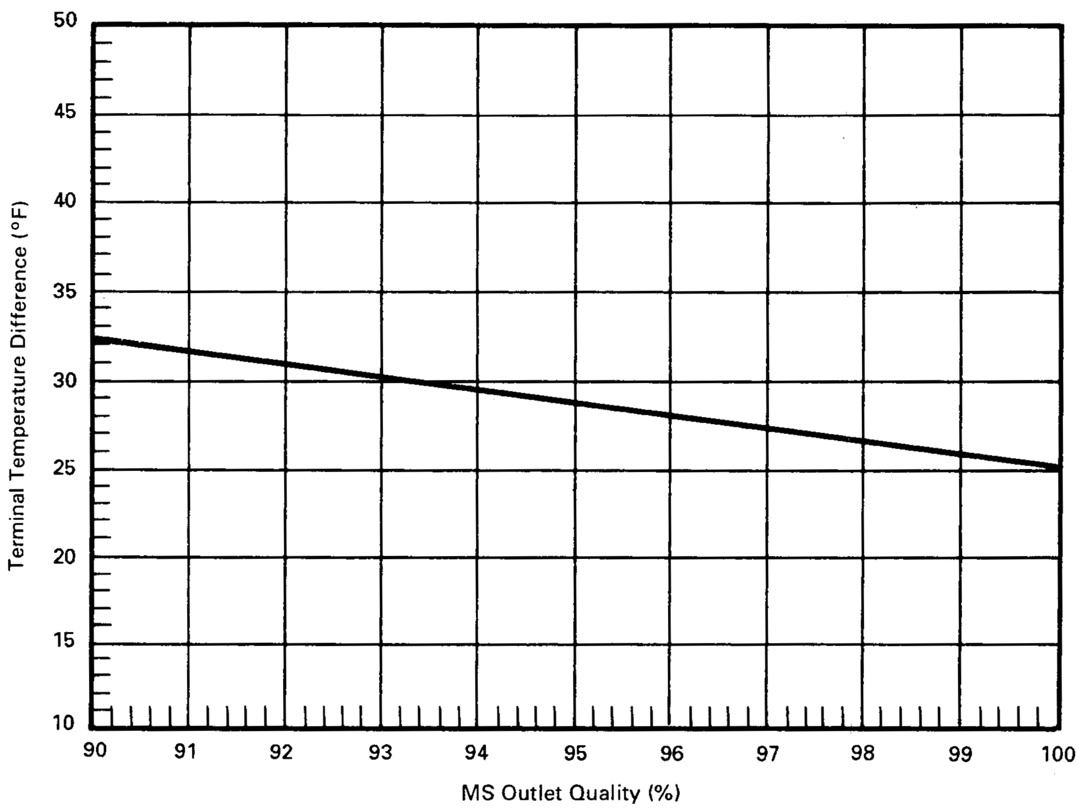
A.8.7 Sum of HR = 94.9 Btu/kW-hr

Therefore, deviation of test performance from expected MSR performance is causing an increase in Heat Rate of 94.9 Btu/kW-hr.



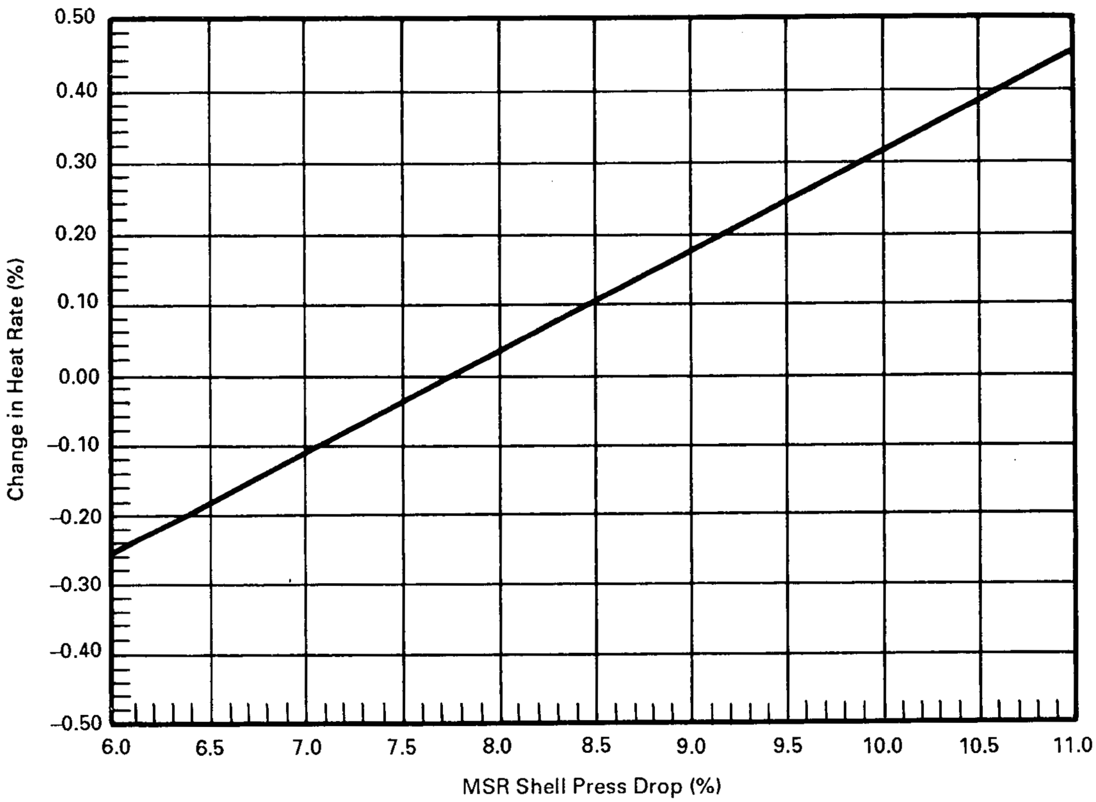
GENERAL NOTE:
This figure is for illustrative purposes only.
Specific unit information should be developed.

FIG. A.3 EXPECTED LP REHEATER TTD



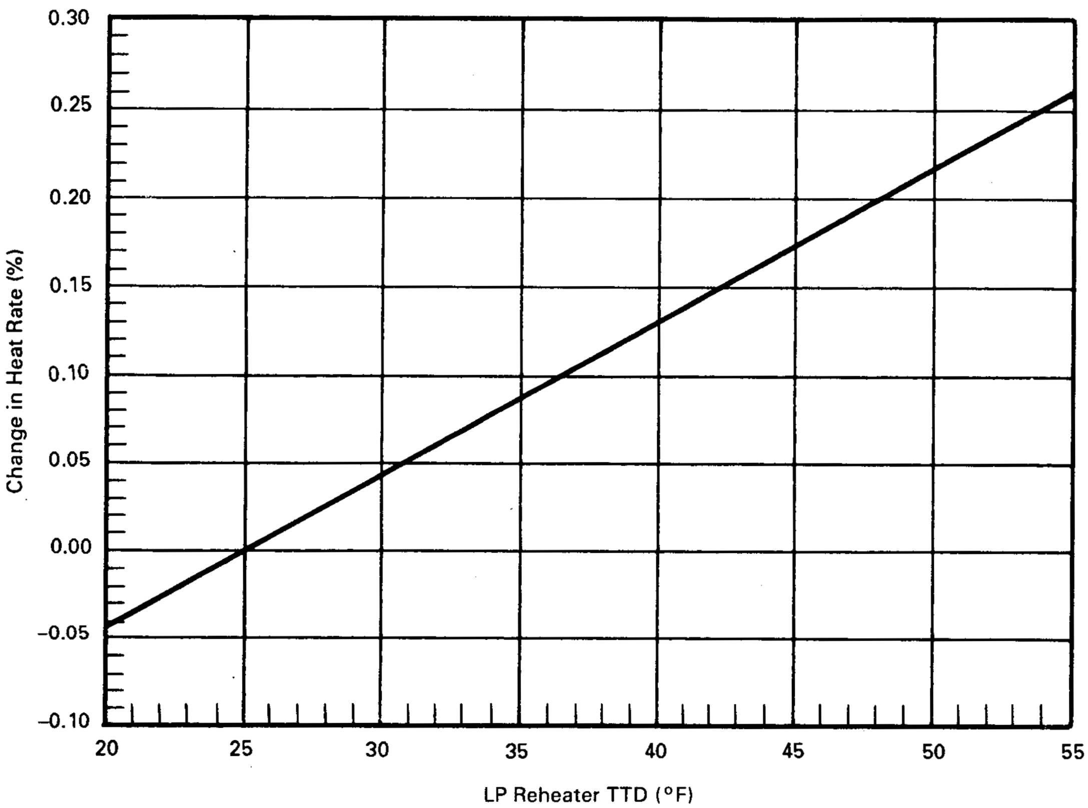
GENERAL NOTE:
This figure is for illustrative purposes only.
Specific unit information should be developed.

FIG. A.4 EXPECTED HP REHEATER TTD



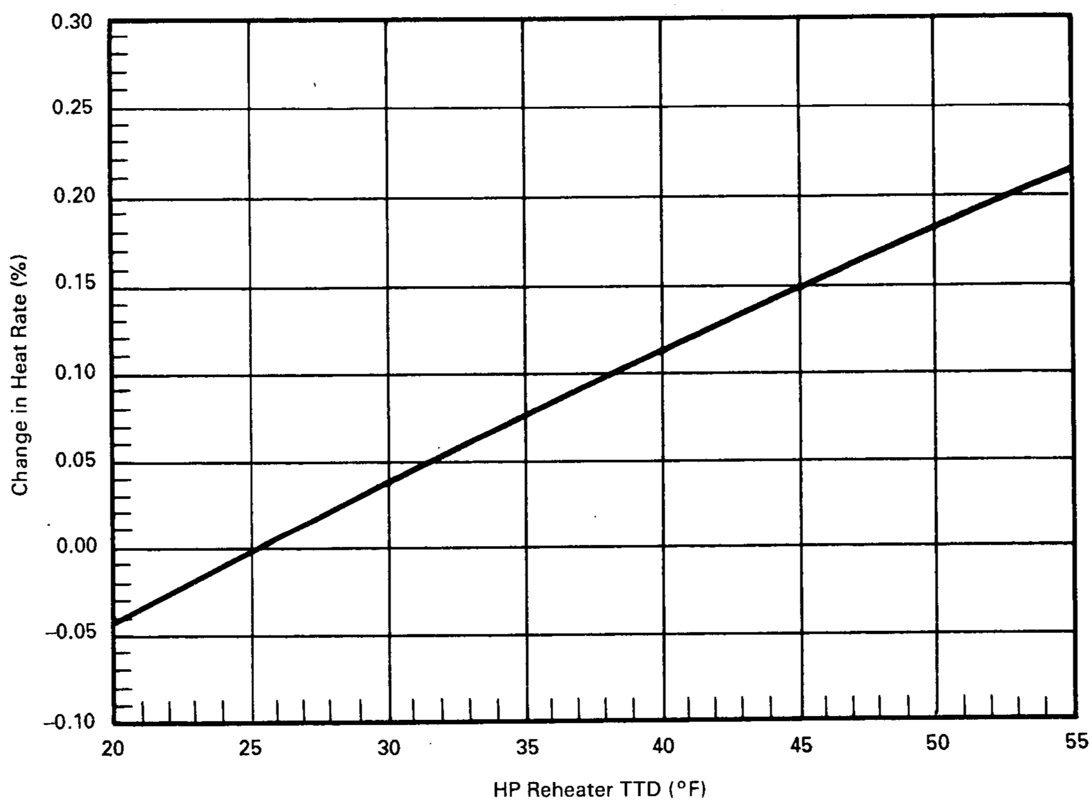
GENERAL NOTE:
This figure is for illustrative purposes only.
Specific unit information should be developed.

FIG. A.5 SENSITIVITY TO SHELL PRESSURE DROP



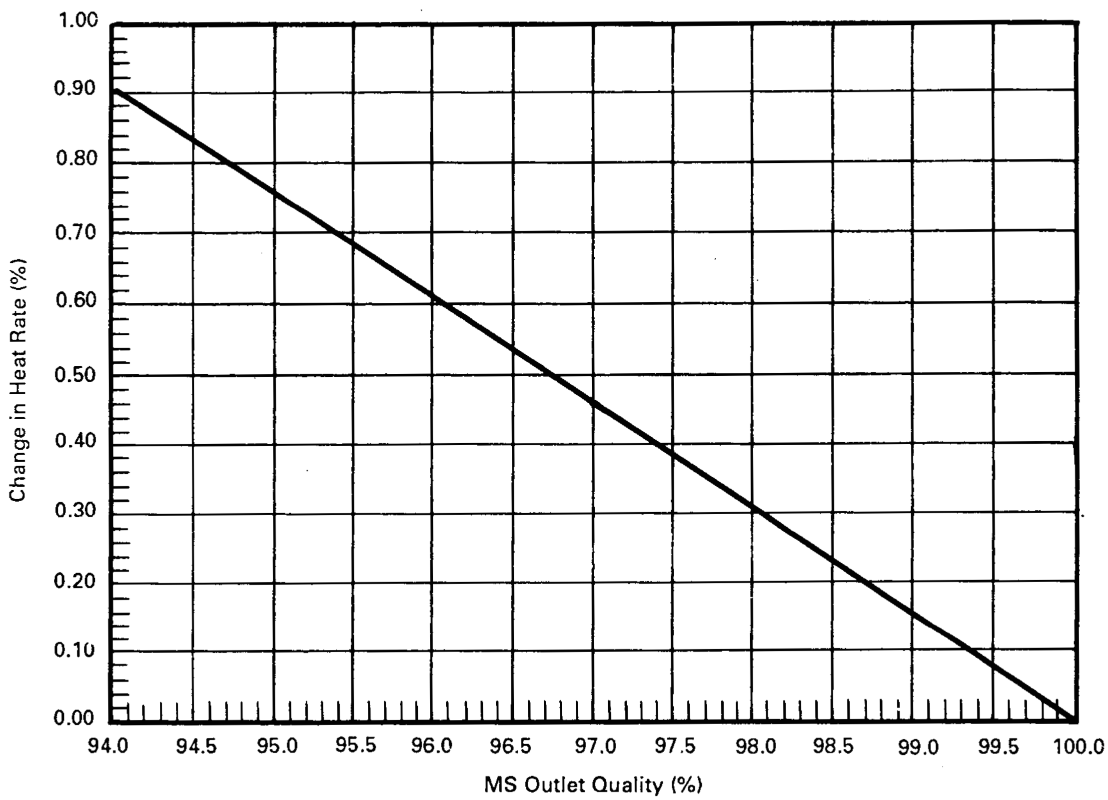
GENERAL NOTE:
This figure is for illustrative purposes only.
Specific unit information should be developed.

FIG. A.6 SENSITIVITY TO LP REHEATER TTD



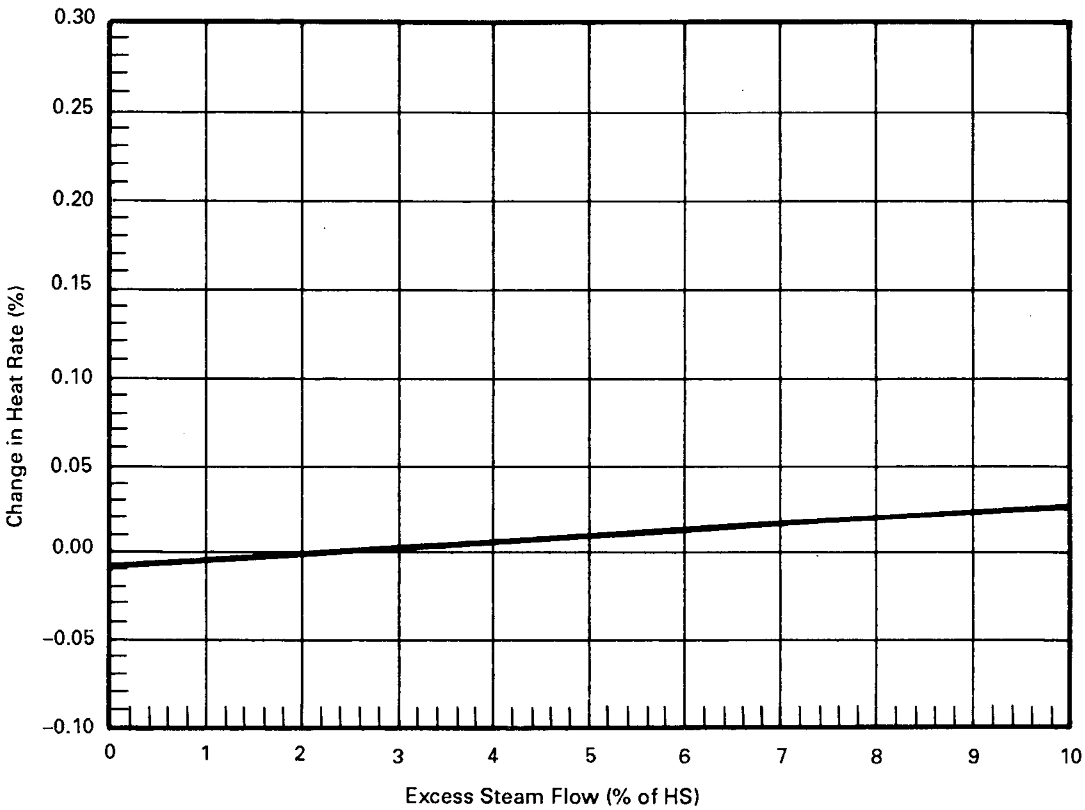
GENERAL NOTE:
This figure is for illustrative purposes only.
Specific unit information should be developed.

FIG. A.7 SENSITIVITY TO HP REHEATER TTD



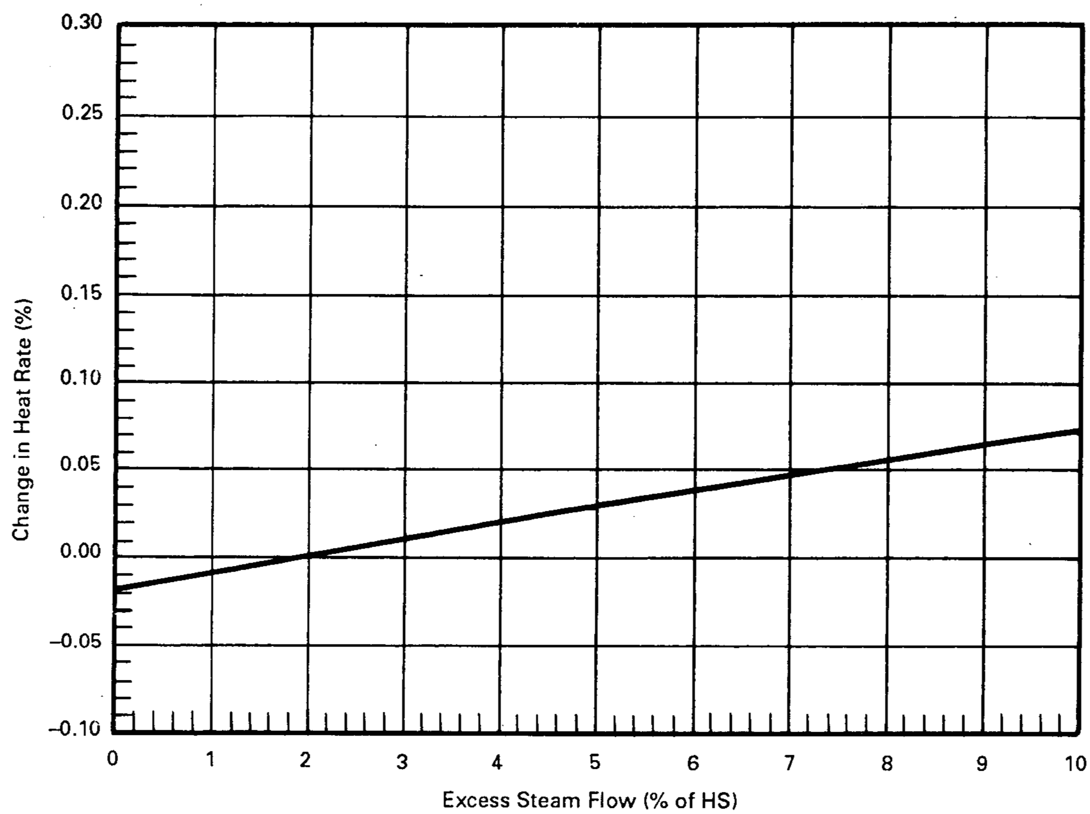
GENERAL NOTE:
This figure is for illustrative purposes only.
Specific unit information should be developed.

FIG. A.8 SENSITIVITY TO MS OUTLET QUALITY



GENERAL NOTE:
This figure is for illustrative purposes only.
Specific unit information should be developed.

FIG. A.9 SENSITIVITY TO LP REHEATER EXCESS STEAM FLOW



GENERAL NOTE:
This figure is for illustrative purposes only.
Specific unit information should be developed.

FIG. A.10 SENSITIVITY TO HP REHEATER EXCESS STEAM FLOW

APPENDIX B — MEASUREMENT UNCERTAINTY CALCULATIONS

TABLE B.1
TTD (HP)

Test Measurement	Effect on TTD	Actual Measurement Bias	Uncertainty of Test TTD (Bias)	Actual Measurement Precision	Uncertainty of Test TTD
Cycle Steam Outlet Temp	1.00°F/°F	± 1°F	1.00°F	± 0.35°F	0.350°F
HP Heating Steam Press	1.21°F/%	± 0.25%	0.303°F	± 0.44%	0.532°F
Square Root Sum of the Squares			1.04°F		0.637°F
Total Uncertainty			1.22°F		

TABLE B.2
CYCLE STEAM FLOW MEASUREMENT UNCERTAINTY

	Measurement Error	Sensitivity	Uncertainty in Cycle Steam Flow
Measured Feedwater Flow	1%	1.16% per 1%	1.16%
HP Heating Steam	5%	−0.22% per 5%	−0.22%
LP Heating Steam	5%	−0.12% per 5%	−0.12%
MS Drain Flow	10%	−0.95% per 10%	−0.95%
Packing Leakages	10%	−0.03% per 10%	−0.03%
Heater 6 Extraction Enthalpy	20 Btu	0.05% per 5 Btu	0.20%
Heater 5 Extraction Enthalpy	40 Btu	0.06% per 5 Btu	0.48%
Final Feedwater Enthalpy	2 Btu	−0.32% per 2 Btu	−0.32%
Enthalpy of Heater 6 Feedwater In	2 Btu	0.32% per 2 Btu	0.32%
Enthalpy of Heater 5 Feedwater Out	2 Btu	−0.24% per 2 Btu	−0.24%
Enthalpy of Heater 5 Feedwater In	2 Btu	0.24% per 2 Btu	0.24%
Enthalpy of Heater 6 Drain	2 Btu	0.00% per 2 Btu	0.00%
Enthalpy of Heater 5 Drain	2 Btu	−0.09% per 2 Btu	−0.09%
Total Uncertainty (RMS)		1.71%	

TABLE B.3
TTD (LP)

Test Measurement	Effect on TTD	Actual Measurement Bias	Uncertainty of Test TTD (Bias), °F	Actual Measurement Precision	Uncertainty of Test TTD Precision, °F
Cycle Steam Out Temperature	0.92°F/°F	±1°F	0.92	±0.354°F	0.326
Cycle Steam Out Pressure	0.40°F/%	±0.25%	0.10	±0.44%	0.176
Cycle Steam Flow	0.70°F/%	±2.0%	1.40	±0.032%	0.022
HP Heating Steam Flow	0.68°F/%	±5.0%	3.40	±0.288%	0.196
HP Heating Steam Pressure	0.18°F/%	±0.25%	0.045	±0.44%	0.079
HP Heating Steam Quality	0.86°F/%	±0.25%	0.215	N/A	0
HP Drain Temperature	0.10°F/°F	±1°F	0.10	±0.525°F	0.053
HP Excess Steam Flow	0.02°F/%	±10%	0.20	±0.632%	0.013
LP Heating Steam Pressure	0.99°F/%	±0.25%	0.248	±0.288%	0.285
Square Root Sum of the Squares			3.81		0.52
Total Uncertainty			3.85°F		

TABLE B.4
MOISTURE SEPARATOR OUTLET QUALITY (XMSO)

XMSO Test Measurement	Effect on XMSO	Actual Measurement Bias	Uncertainty of Test XMSO (Bias), %	Actual Measurement Precision	Uncertainty of Test (Precision), %
Cycle Steam Out Temperature	0.0655%/°F	± 1°F	0.0655	±0.354°F	0.0232
Cycle Steam Out Pressure	0.0249%/%	±0.25%	0.0062	±0.40%	0.0100
Cycle Steam Flow	0.1066%/%	±2.0%	0.2132	±0.032%	0.0034
HP Heating Steam Flow	0.0497%/%	±5.0%	0.2485	±0.288%	0.0143
HP Heating Steam Pressure	0.013%/%	±0.25%	0.0033	±0.44%	0.0057
HP Heating Steam Quality	0.061%/%	±0.25%	0.0153	N/A	0
HP Drain Temperature	0.0101%/°F	±1°F	0.0101	±0.525%°F	0.0053
HP Excess Steam Flow	0.0012%/%	±10%	0.0120	±0.632%	0.0008
LP Heating Steam Flow	0.0485%/%	±5%	0.2425	±0.288%	0.0140
LP Heating Steam Pressure	0.0006%/%	±0.25%	0.0002	±0.44%	0.0003
LP Heating Steam Quality	0.0452%/%	±1.5%	0.0678	N/A	0
LP Drain Temperature	0.0071%/°F	±1°F	0.0071	±0.525°F	0.0037
LP Excess Steam Flow	0.0008%/%	±10%	0.0080	±0.632%	0.0005
Square Root Sum of the Squares			0.42		0.03
Total Uncertainty			0.42%		

TABLE B.5
EXCESS STEAM FLOW

Treat Measurement	Uncertainty of Test ES Flow (Bias), (%)	Uncertainty of Test ES Flow (Precision), (%)
Excess Steam Flow	±10.0	±0.6
Total Uncertainty	10.0%	

TABLE B.6
MSR SHELL PRESSURE DROP

Test Measurement	Uncertainty of Test Pressure Drop (Bias), (PSID)	Uncertainty of Test Pressure Drop (Precision), (PSID)
Excess Steam Flow	±0.02	±0.12
Total Uncertainty	0.122 psid	

TABLE B.7
TOTAL OVERALL TEST UNCERTAINTY

Performance Parameter	Individual Uncertainty	Parameter Effect on Heat Rate	Uncertainty of Test Heat Rate, (Btu/kW-hr)
TTD (HP)	±1.22 °F	1.4 Btu/kW-hr/°F	1.71
TTD (LP)	±3.85 °F	0.4 Btu/kW-hr/°F	1.54
XMSO	±0.42 °F	23 Btu/kW-hr/%	9.66
Shell Pressure Drop	±0.122 PSID	6.6 Btu/kW-hr/psid	0.81
ES Flow	±10.0%	0.01 Btu/kW-hr/%	0.10
Square Root Sum of the Squares			9.96 Btu/kW-hr or 0.10%, Based upon a typical plant heat rate of 10,000 Btu/kW-hr

GENERAL NOTE:
The precision errors were calculated for the test measurements of interest by the formula:

$$\frac{t \times s}{\text{Square Root of (N)}}$$

Where:

s = Standard Deviation
t = Students t
N = Number of Readings

COMPLETE LISTING OF ASME PERFORMANCE TEST CODES

PERFORMANCE TEST CODES

While providing for exhaustive tests, these Codes are so drawn that selected parts may be used for tests of limited scope.

A complete list of all Performance Test Codes appears at the end of this book.

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ISBN # 0-7918-2205-2

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