Procedures for Routine Performance Tests of Steam Turbines

(Not Intended for Acceptance Testing)

ASME PTC 6S Report-1988

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PERFORMANCE TEST CODES

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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Please Note: ASME issues written replies to inquiries concerning interpretation of technical aspects of this document. PTC 6S Report-1988 is being issued with an automatic subscription service to the interpretations that will be issued to it up to the publication of the 1992 Edition.

FOREWORD

(This Foreword is not part of ASME PTC 6S Report-1988.)

Users of both large and small turbines have experienced an increasing need for procedures for routine turbine tests which trend performance with time. The use of full-scale ASME Performance Test Code procedures and instrumentation for this purpose is expensive and produces information and accuracy beyond that necessary for periodic monitoring. When ASME Performance Test Code Committee No. 6 was reorganized to revise PTC 6-1949, it was charged also with developing simplified procedures for periodic tests. Because of the routine nature of the tests, these procedures were to emphasize repeatability of results rather than absolute accuracy and thus provide a more economic means of monitoring performance trends.

This Report reflects the consensus of knowledgeable engineers and contains recommended procedures for collecting sufficiently accurate data to permit analyses of performance trends. Recommendations are given which include advance planning, cycle isolation, and suggested presentation of results. Emphasis is placed upon the use of accurate instrumentation, approaching measurement uncertainties required by the Code, for the measurement of critical variables that are part of the heat-rate equation. Other instrumentation is specified to produce results of good accuracy and of a high degree of repeatability. With the application of automatic data-logging and on-line computer systems to the plant cycle, the procedures presented in this Report, when applied to this end, should satisfy the needs of users of both large and small turbines.

Procedures recommended in this Report are not intended to produce abolute levels of performance. If absolute performance level is required, the ASME Test Code for Steam Turbines, PTC 6, 1976, reaffirmed 1985, or the Interim Test Code for an Alternative Procedure for Testing Steam Turbines, PTC 6.1, 1984, should be followed. For other levels of accuracy, where the test instrumentation varies from the Test Code specified procedure, the Report by PTC Committee No. 6 on "Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines," 1985 should be consulted.

Users of this Report are requested to comment and provide to the Committee supporting data obtained with these procedures. Such comment and repeatability data covering long-term and/or extensive experience will provide guidance for subsequent revisions of this Report. User suggestions and data should be submitted to the Secretary, ASME Performance Test Codes Committee, 345 East 47th Street, New York, New York 10017.

This Report was approved by the ASME Board on Performance Test Codes and adopted as a standard practice of the Society on May 8, 1988. It was approved as an American National Standard by the ANSI Board of Standards Review on September 8, 1988.

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(In addition to the above personnel, the Committee is indebted to Ms. E. Pitchford of Lower Colorado River Authority and H. S. Arnold of Public Service Electric & Gas Corp. for their efforts on the early work of this Standard.)

BOARD ON PERFORMANCE TEST CODES

CONTENTS

Tables

Appendices

Figures

Table

SECTION 0 — INTRODUCTION

0.1 PURPOSE OF REPORT

This Report provides turbine-test procedures for the analysis and supervision of relative performance throughout the life of the turbine. These procedures will determine trends of operating efficiency, detect trouble, and furnish test data to evaluate efficiency changes in the turbine cycle. These procedures are designed to minimize test instrumentation and personnel. However, precision instrumentation at critical test locations is recommended for reliable results. A high degree of repeatability, rather than the acceptance test level of performance, is sought.

0.2 CONTENT

This Report contains test procedures for recommended instrumentation, planning, conduct, calculation, and evaluation of the test results. Separate procedures are recommended for specific turbine types.

0.3 PURPOSE OF TEST PROGRAM

A routine turbine-performance test program will:

(*a*) provide guidance in scheduling maintenance outages on the basis of performance;

(*b*) provide guidance in establishing the loading sequence of steam turbine-generator units according to their current relative performance;

(*c*) evaluate major modifications of the turbine or turbine cycle, and changes in operating procedure;

(*d*) detect performance changes in specific areas of the turbine or the turbine cycle;

(*e*) check the accuracy of station instruments by comparison with test instruments;

(*f*) train personnel in testing techniques.

0.4 REFERENCES

The ASME Test Code for Steam Turbines, PTC 6-1976, and Interim Test Code for an Alternative Procedure for Testing Steam Turbines, PTC 6.1-1984, are the basic references for this Report. The term "Code" used in this Report refers to these documents. The ASME Code on Definitions and Values, PTC 2-1980 and the applicable supplements of PTC 19 Series, on Instruments and Apparatus, provide supplementary information. A separate report by Performance Test Code Committee No. 6, Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines, should be used to evaluate the level of accuracy afforded by the instrumentation recommended for this Report's test procedures. Whenever PTC 19.1, Measurement Uncertainty, is referenced in this document, 95 percent confidence levels have been used in accordance with accepted practices. Appendix A to Test Code for Steam Turbines, PTC 6A-1982, and Sections 7 through 13 of this Report provide numerical examples of various turbine calculations which should prove useful in applying this Report's procedures.

SECTION 1 — OBJECT, SCOPE, AND INTENT

1.1 OBJECT

The test procedures of this Report are intended for periodic turbine tests and do not supplant the Code as the basic procedure for turbine acceptance tests. The Code is used for the accurate testing of steam turbines to obtain performance level with minimum uncertainty.

1.2 SCOPE

Sections 3 through 5 of this Report present general recommendations for instrumentation and test planning. These recommendations are based on current industry practice for the periodic determination of turbine-cycle performance. Section 6 discusses interpretation of test results and shows typical plots of test data for analysis of turbine performance. Sections 7 through 12 present test procedures for selected types of turbine cycles. Each of these procedures contains specific recommendations for instrumentation and method for testing a selected turbine type. Although all possible turbine types are not covered, some typical examples are presented. Combinations of the types presented may be used to cover other arrangements. For each recommended test procedure, the expected value of repeatability is estimated on the basis of current industry experience. This value of repeatability must be used to judge the significance of the indicated level of performance as compared to the chronological trend of past performance. (See para. 3.8.3 for discussion of repeatability.)

1.3 INTENT

This Report provides procedures to periodically monitor changes in overall turbine cycle performance. Supplementary instrumentation and data may be included in the test procedure to diagnose the causes of changed performance. This supplementary information may assist in evaluating the effect of component changes on the overall performance. Some users may prefer the simplicity of the recommended procedure and then run a second test only when a detailed analysis is required. These procedures define both primary and secondary data for a turbine-performance analysis.

For reference purposes only, Section 13 presents other test procedures for determining turbine-performance trends; however, these procedures may not provide complete data for analysis of all components in the turbine cycle. In special cases, they may provide adequate information and be advantageous due to their low cost and simplicity.

For on-line computer monitoring of steam-turbine performance, the simplified test procedures given in this Report can serve as a basis for choice of instrumentation and development of calculation procedures. The instrumentation is selected to achieve repeatable results consistent with the objective of monitoring the trend of turbine performance.

Diagnostic monitoring systems for vibration, oil cleanliness, rotor crack detection, and solid particle erosion and supervisory systems for differential expansion, bearing metal temperature, bearing wear, turbine load, and speed are not indicators of turbine performance. Analytical techniques using data from these systems can result in early identification of potential problems.

SECTION 2 — DEFINITIONS AND DESCRIPTION OF TERMS

2.1 SYMBOLS

The following symbols are to be used unless otherwise defined in the text.

Symbol	Description	Units U.S. Customary	Units SI
A	Area	in.^{2}	m^2
F	Force	lbf	N
g	Local value of acceleration due to gravity	ft/sec^2	m/s^2
g_0	Standard value of acceleration due to gravity = 32.17406 ft per sec per sec (9.80665 meters per sec per sec). This is an internationally agreed upon value which is close to the mean of 45 deg. N latitude at sea level.	ft/sec^2	m/s^2
h	Specific enthalpy	Btu/lbm	kJ/kg
J	Mechanical equivalent heat, 1 Btu = 778.1693 ft-lbf = 1/3412.142 kWh	Btu	J
M	Moisture, 100 − x	%	%
m	Mass	lbm	kg
N	Rotational speed	rpm	rps
P	Power	kW or hp	kW
p	Pressure	psia	Pa-a
s	Specific entropy	Btu/lbm°R	kJ/kgK
t	Temperature	°F	°C
T	Temperature	°R	K
V	Velocity	ft/sec	m/s
v	Specific volume	ft^3/lbm	m^3/kg
w	Rate of flow	lbm/h	kg/h
x	Quality of steam, percent of dryness	%	%

2.1 SYMBOLS (cont'd.)

Symbol	Description	Units U.S. Customary	Units SI
η	Efficiency	%	%
ϱ	Density	lbm/ft^3	kg/m^3
γ	Specific weight	lbf/ft^3	N/m^3

2.2 ABBREVIATIONS

Abbreviation	Term	Units U.S. Customary	Units SI
HR	Heat rate	Btu/kWhr Btu/hphr	kJ/kWh
SR	Steam rate	lbm/kWhr lbm/hphr	kg/kWh

2.3 SUBSCRIPTS

Subscript	Description
g	Generator
r	Rated condition
c	Corrected
s	Specified operating condition, if other than rated
t	Test operating condition
1	Condition at a point directly preceding the turbine-stop valves and steam strainers.
2	For turbines using superheated steam: condition at turbine-outlet connection leading to the first reheater. For turbines using predominantly wet steam: condition at turbine-outlet connection leading to external moisture separator.
3	For turbines using superheated steam: condition downstream of the first reheater, at a point directly preceding the reheat-stop valves, intercept valves or steam-dump valves, whichever are first, if furnished under turbine contract.[1] For turbines using predominantly wet steam: condition at external moisture separator outlet.

[1]It may be necessary to correct for pressure drop in piping between reheat or low-pressure stop valves, intercept valves, steam-dump valves and turbine shell if such piping is not furnished under the turbine contract.

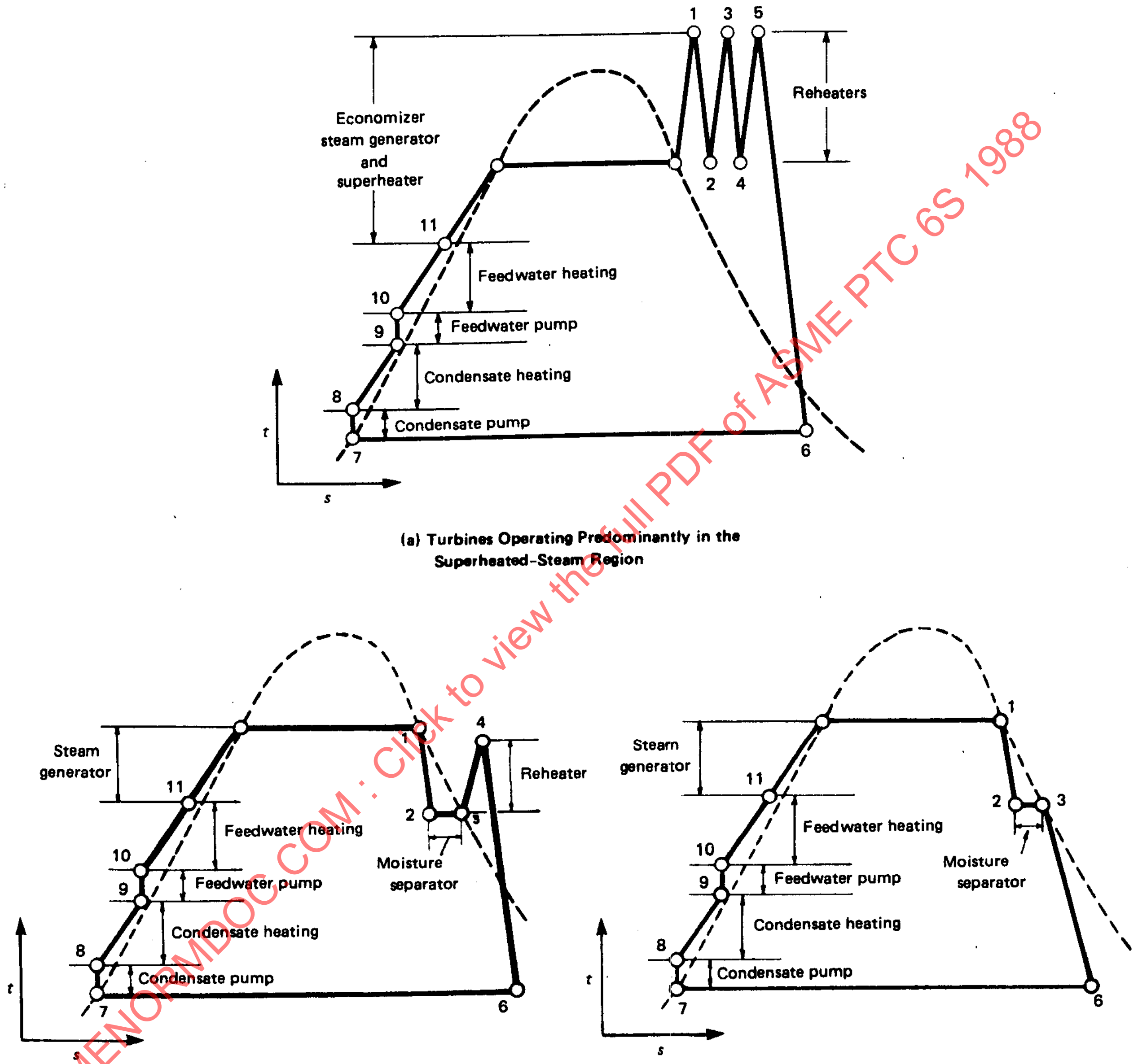
2.3 SUBSCRIPTS (cont'd.)

Subscript	Description
4	For turbines using superheated steam: condition at turbine-outlet connection leading to the second reheater. For reheat turbines using predominantly wet steam: condition downstream of the reheater, at a point directly preceding the reheat-stop valves, intercept valves or steam-dump valves, whichever are first, if furnished under contract.[1]
5	For turbines using superheated steam and two stages of reheat: condition downstream of the second reheater, at a point directly preceding the reheat-stop valves, intercept valves or steam-dump valves, whichever are first, if furnished under the turbine contract.[1]
6	Condition at turbine-exhaust connection.
7	Condition at condenser-condensate discharge.
8	Condition at condensate-pump discharge.
9	Condition at feedwater pump or feedwater-booster-pump inlet.
10	Condition at feedwater-pump discharge.
11	Condition at the discharge of the final feedwater heater.
*a*1	Superheater desuperheating water.
*a*2	First-reheater desuperheating water.
*a*3	Second-reheater desuperheating water.
cL	Condenser circulating water leakage.
E	Extraction steam.
e	Make-up water admitted to the condensate system.
pL	Packing leak-off (shaft or valve stems).
i,ii,..n	Sequence.

The temperature-entropy diagrams shown in Fig. 2.1 (a) through (c) are intended to serve as a key to the numerical subscripts employed in the foregoing.

2.4 DEFINITIONS

Term	Definition	Unit U.S. Customary	Unit SI
Steam rate	Steam consumption per hour per unit output in which the turbine is charged with the steam quantity supplied.	lbm/kWhr lbm/kWhr	kg/kWh kg/kWh



(a) Turbines Operating Predominantly in the Superheated-Steam Region

(b) Temperature-Entropy Diagram Without Reheat

(c) Temperature-Entropy Diagram With Reheat

(b) and (c) Turbines Operating Predominantly in the Wet-Steam Region

FIG. 2.1 TEMPERATURE-ENTROPY DIAGRAMS

2.4 DEFINITIONS (cont'd.)

Term	Definition	Unit U.S. Customary	Unit SI
Heat rate	Heat consumption per hour per unit output. The turbine is charged with the aggregate enthalpy[2] of the steam supplied plus any chargeable aggregate enthalpy added by the reheaters. It is credited with the aggregate enthalpy of feed-water returned from the cycle to the steam generator. Turbine-generator performance may be defined on the basis of the gross power output at the generator terminals less the power used by the minimum number of electrically-driven turbine auxiliaries and excitation equipment, supplied as part of the turbine-generator unit, required for reliable and continuous operation.	Btu/kWhr Btu/hphr	kJ/kWh kJ/kWh
Valve-loop curve	The continuous curve of actual heat rate for all values of output over the operating range of the unit.		
Valve points	Those valve positions which correspond to the low points of the valve-loop curve.		
Mean of the valve loops	A smooth curve which gives the same load-weighted average performance as the valve-loop curve.		
Enthalpy drop	The difference in steam enthalpy between turbine inlet conditions and turbine outlet conditions. This is applicable to individual turbine sections such as: high-pressure section or intermediate-pressure section.	Btu/lbm	kJ/kg
Power	The useful energy, per unit of time, delivered by the turbine or turbine-generator unit.	hp or kW	kW

2.5 CONVERSIONS TO SI (METRIC) UNITS

Quantity	Conversion	Multiplication Factor
Heat rate	Btu/kWhr to kJ/kWh	1.055056×10^{0}
Steam rate	lbm/kWhr to kg/kWh	4.535924×10^{-1}
Mass flow	lbm/hr to kg/h	4.535924×10^{-1}
Pressure	lbf/in.^{2} to bar	6.894757×10^{-2}
	in. Hg to mm Hg	$2.54^* \times 10^{+1}$
	in. Hg abs. at 0°C to kPa	3.386386×10^{0}

[2]*Aggregate enthalpy* — product of enthalpy, Btu/lbm (kJ/kg) and rate of flow, lbm/hr (kg/h); Btu/hr (kJ/h).

2.5 CONVERSIONS TO SI (METRIC) UNITS (cont'd.)

Quantity	Conversion	Multiplication Factor
Pressure (cont'd.)	lbf/in.^{2} to kPa	6.894757×10^0
(1 bar = 10^5 Pa = 760.06 mm Hg)		
Temperature measured	°F to °C	$t_C = (t_F - 32)/1.8$
Temp., thermodynamic	°F to K	$T_K = (t_F + 459.67)/1.8$
Density	lbm/ft^3 to kg/m^3	1.601846×10^{-1}
Specific enthalpy	Btu/lbm to kJ/kg	2.326×10^0
Specific entropy	Btu/lbm°R to kJ/kgK	$4.1868^* \times 10^0$
Specific heat	Btu/lbm°R to kJ/kgK	$4.1868^* \times 10^0$
Length	ft to m	$3.048^* \times 10^{-1}$
Area	ft^2 to m^2	$9.290304^* \times 10^{-2}$
Volume	ft^3 to m^3	$2.831685^* \times 10^{-2}$
Velocity	ft/sec to m/s	$3.048^* \times 10^{-1}$

*Exact relationships in terms of base units.

SECTION 3 — GUIDING PRINCIPLES

3.1 ADVANCE PLANNING FOR TEST

3.1.1 General. A plan for instrumentation of the turbine and turbine heat cycle should be developed prior to installation of the turbine. This plan should include adequate provision for physical location, installation, and number of test instruments needed to achieve test results with good repeatability. To minimize costs, this plan should be incorporated into the initial turbine-cycle design. Some items to be considered are:

(*a*) objectives of the test

(*b*) location and installation of a calibrated primary flow-metering section

(*c*) provision for the accurate measurement of output

(*d*) location and installation of test connections for primary pressure and temperature measurements

(*e*) location and installation of duplicate instrument connections required to insure correct measurements at critical points

(*f*) method of handling leak-off flows, orificed continuous-drain flows, bypass flows, continuous blowdowns, etc. to avoid complications in testing or the introduction of errors

(*g*) selection of test instrumentation capable of the repeatability required for consistent test results

(*h*) number and location of valves or selection of other means (see para. 3.2.7) to insure positive isolation of the cycle during tests

(*i*) location of test instruments in environmentally stable areas to reduce calibration drift

(*j*) the use of electronic data acquisition equipment

(*k*) location of test instruments in groups to facilitate calibration and use and to minimize the number of required observers

3.2 CYCLE ISOLATION

3.2.1 Cycle Isolation. Positive isolation of the turbine cycle during the test run is essential for heat rate and capability test repeatability. Isolation is generally not required for enthalpy-drop efficiency tests. However, cycle conditions affecting turbine-section pressure ratios should be consistent among tests. Positive cycle isolation includes both external and internal isolation.

3.2.2 External Isolation. External isolation concerns flows that enter or leave the turbine cycle. These flows should be isolated from the system, if possible, to eliminate measurement errors. If the isolation of these flows is open to question, provisions should be made to measure them.

3.2.3 Internal Isolation. Internal isolation concerns flows which do not enter or leave the turbine cycle but which may bypass their intended destination. Examples of such flows are steam line drain flows to the condenser or feedwater heater bypass flows. Internal isolation cannot be verified by the inventory summation method in para. 3.2.8.

3.2.4 Flows That Should Be Isolated. The following equipment and extraneous flows that should be isolated from the primary turbine-feedwater cycle are:

(*a*) large-volume storage tanks

(*b*) evaporators and allied equipment such as evaporator condenser and evaporator preheaters

(*c*) bypass systems and auxiliary steam lines for starting

(*d*) bypass lines for condensate primary-flow-measuring devices

(*e*) turbine sprays

(*f*) drain lines on stop, intercept and control valves

(*g*) interconnecting lines to other units

(*h*) steam to air preheaters (if isolation is not possible, flows shall be measured)

(*i*) demineralizing equipment (isolation of demineralizing equipment does not necessarily mean removing the equipment from the cycle. It does mean, however, that all interconnections with other units must be isolated and items such as recirculating lines that affect the primary-flow measurement must be isolated or their flows measured.)

(*j*) steam generator fill lines

(*k*) steam-generator vents

(*l*) steam-operated soot blowers

(*m*) condensate and feedwater-flow bypassing heaters
(*n*) heater-drain bypasses
(*o*) heater-shell drains
(*p*) heater water-box vents
(*q*) hogging jets
(*r*) condenser water-box priming jets
(*s*) steam or water lines for station heating
(*t*) steam or water lines installed for water washing the turbine

3.2.5 Flows That Should Be Isolated or Measured. Extraneous flows which enter or leave the cycle in such a manner that, if ignored, will cause an error in the flows through the turbine, should be isolated or measured. Typical of such flows are:

(*a*) boiler-fire-door-cooling flow and boiler-slag-tap cooling-coil flow
(*b*) sealing and gland-cooling flow on the following (both supply and return):
- (*1*) condensate pumps
- (*2*) feedwater pumps
- (*3*) boiler-water or reactor-circulating pumps
- (*4*) heater-drain pumps when not self-sealed
- (*5*) turbines for turbine-drive pumps
- (*6*) reactor control-rod seals

(*c*) desuperheating water
(*d*) feedwater-pump minimum-flow lines and balance-drum flow
(*e*) steam for fuel-oil atomization and heating
(*f*) steam-generator blowdowns
(*g*) turbine water-seal flow
(*h*) desuperheating water for turbine-cooling steam
(*i*) emergency blowdown valve or turbine-packing leakage and sealing system
(*j*) turbine water-seal overflows
(*k*) steam, other than packing-leak-off steam, to the steam-seal regulating valve
(*l*) make-up water, if necessary
(*m*) pegging or sparging steam (such as higher-stage extraction at light loads) for low-pressure operation of deaerator
(*n*) heater vents are to be closed if possible, and if not possible, shall be throttled to a minimum
(*o*) deaerator-overflow line
(*p*) deaerator vents
(*q*) water leakage into any water-sealed flanges, such as water-sealed vacuum breakers
(*r*) pump seal leakage leaving the cycle
(*s*) automatic-extraction steam for industrial uses
(*t*) continuous drains from wet-steam turbine casings and connection lines
(*u*) sub-cooled water used for moisture separator
(*v*) reactor-core spray
(*w*) heater-blanketing steam lines

3.2.6 Unmeasurable Flows. It will be necessary to use calculated values for internal pump leakages, shaft packing, valve-stem leakages, internal turbine leakages, and turbine-drain flows when it is impossible to measure these flows.

3.2.7 Methods of Isolating. The following methods are suggested for isolating or verifying isolation of miscellaneous equipment and extraneous flows from the primary turbine-feedwater cycle:

(*a*) double valves and telltales
(*b*) blank flanges
(*c*) blank between two flanges
(*d*) removal of spool piece for visual inspection
(*e*) visual inspection for steam blowing to atmosphere from sources such as safety valves and valve-stem packings
(*f*) closed valve which is known to be leak-proof and is not operated prior to or during test
(*g*) temperature indication
(*h*) tracer indication of presence of leakage

3.2.8 Cycle Water Balance. For an acceptable test accuracy, the difference between the sums of the measured storage changes and flows in-and-out of the system should not exceed ± 0.5 percent of the maximum turbine throttle flow. However, for good repeatability, a consistent level of unaccounted-for loss based on a chronological series of tests is satisfactory, even though the unaccounted-for loss exceeds the ± 0.5 percent of maximum throttle flow. Any change in the unaccounted-for loss exceeding 0.1 percent of the maximum throttle flow should be investigated, since it may directly affect the test result. If the cause of the change can be determined, the test result should be adjusted to compensate for this factor. If the cause cannot be determined, the change in losses directly affects the repeatability of this test in comparison with the trend of previous tests.

3.3 PRELIMINARY TESTS

Preliminary tests may be run for the following reasons:

(*a*) to establish the location of turbine valve points;
(*b*) to isolate the turbine cycle and make a water balance of the cycle. This establishes the magnitude of

unaccounted-for leakage which directly affects the accuracy of a test result. Any significant change in unaccounted-for leakage must be investigated before an acceptable test run can be made for comparison with previous tests.

(c) to check internal cycle conditions. If variations in equipment operation that affect turbine stage flows or stage pressure occur, corrective actions should be undertaken. The following is a partial list of items that should be considered:

(1) changes in feedwater heater performance
(2) bypassing of feedwater heaters
(3) change in positions of feedwater heater level control valves as an indication of heater leakage
(4) amount of desuperheating water used for control of steam temperature
(5) makeup evaporator in service
(6) cycle heat requirements for building heating or combustion air heating
(7) steam supplied to a turbine-driven feedwater pump or fan
(8) feedwater pump recirculation leakage

(d) to check the operation of test instrumentation and to familiarize test observers with the routine of proper testing procedures. The sequential timing of readings and the coordination of simultaneous readings can be checked out under test conditions.

(e) to compare test-instrument readings with station operating-instrument readings and to detect differences that should be investigated and corrected;

(f) to establish a preliminary evaluation of the condition of the turbine; and to determine whether diagnostic information is compatible with known laws of engineering principle.

3.4 CONDUCT OF A TEST RUN

3.4.1 Pretest Conditions. Calibration of test instruments should be established prior to the test run. All instruments should be put in service and checked for performance by comparison with previous test data or cross-checked by comparison with station instruments. Preferably, the turbine should be tested with governing valves wide open even if a reduction in throttle pressure is necessary. If valves wide open cannot be obtained, a previously selected valve position should be set using the same technique established on earlier test series. This setting may be made with the turbine on load-limit control to obtain steady-state operating conditions for the test run. During the period immediately following the setting of the turbine-valve position, the isolation procedure may be executed to establish final conditions for a test run. Test conditions which are as close as possible to specified conditions should be established and maintained to minimize the magnitude of test corrections to the turbine test heat rate.

3.4.2 Test Readings. Test readings should be taken according to the established test routine. These readings should be checked for consistency during the progress of the test. The duration of the test run should be adequate to establish accurate averaged data and accurate integrated primary measurements. The recommended reading frequencies for test readings for a two hour test should not be less than the following:

(a) each 2 minutes: primary flow, power output based on indicating meters, turbine speed for mechanical power output

(b) each 10 minutes: secondary flows affecting throttle-flow calculation, primary pressures, primary temperatures

(c) each 20 minutes: secondary flows not affecting throttle-flow calculation

(d) each 20 minutes: secondary electrical data not affecting generator output calculation, secondary pressures, secondary temperatures

(e) each 30 minutes: integrated flows, integrated electric-power measurements, storage level changes

3.5 DATA REDUCTION

Recorded test data should be examined for consistency and reliability. Inconsistent readings which are not substantiated by the trends of related data may be disregarded if at least 90 percent of the total readings are retained for averaging. PTC 19.1 - 1985 provides a procedure that may be used to examine data.

If the test is divided into specific time periods by synchronized readings of all integrating meters, the initial or final period may be discarded if unusual instability of a principal test variable occurred within such period. Automatic data collection and reduction are useful and, if provided, should be programmed to conform to conditions recommended in these procedures. The following steps, as applicable, are suggested for data reduction:

(a) average all test readings and obtain meter-integrated-flow differences and generator-output differences

(b) apply water leg and gage calibrations to averaged pressure readings and convert pressure readings to absolute pressure

(c) convert thermocouple outputs to temperature units and apply thermocouple calibrations

(d) apply meter calibrations and working-fluid density corrections to integrated quantities

(e) convert storage-level differences into flow rate

(f) correct generator output to specified power factor and hydrogen pressure

(g) calculate primary and secondary flows

(h) calculate steam and water enthalpies

3.6 COMPUTATION OF TEST RESULTS

Computation of test results should follow procedures given in the Code and in Appendix A supplementing the Code. The required computations for specific turbine types are also outlined in later Sections of this Report. Group II test corrections that cover all significant test variables should be developed by the user in cooperation with the turbine manufacturer.

3.7 REFERENCE TABLES

The 1967 ASME Thermodynamic and Transport Properties of Steam is the recommended reference for steam properties. Other formulations may differ slightly. Caution should be used when comparing test results, to assure that the same formulation of steam properties is used in each instance. Where machine computation is employed, the computer shall be programmed in accordance with the 1967 IFC Formulation for Industrial Use, which is included in the Appendix to the 1967 ASME Steam Tables.

3.8 REPEATABILITY OF TEST RESULTS

3.8.1 General. Test repeatability is the test instrumentation system's ability to produce test results of the same output/input relationship over extended time periods. Good long-term repeatability is essential in periodic testing to ensure that variations in performance indicated by test results are true indications of the deterioration of the equipment tested. Although absolute accuracy is not stressed in this Report, in general, accurate instrumentation is recognized to have better repeatability over the long term than will less accurate instrumentation. Therefore, the optimum combination of instruments selected must weigh the desired level of test repeatability against the initial cost of the instrumentation and cost of running tests. For guidance in selecting alternative instrumentation of Section 4, it is recommended that the uncertainty intervals given in the Report by PTC Committee No. 6 on Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines-1985, be consulted.

3.8.2 Instrumentation. The specific instrumentation recommended in Sections 7 through 13 was selected to provide consistent repeatability over a long time period. An analysis should be made of the chronological test results to establish the level of repeatability achieved by the specific test installation. PTC 19.1, Measurement Uncertainty, may be used as the procedure for this analysis. Where unexplained deviations or trends are detected, it may be necessary to recalibrate or to use more accurate instrumentation to raise the level of repeatability.

3.8.3 Estimate of Repeatability. Uncertainty intervals for various instruments, such as those specified in PTC-6 Guidance Report, are expressed typically as a single value meant to include both the random and bias uncertainty components. The relative contributions of random and bias errors are unknown for most instruments.

Caution should be used in applying statistical techniques such as reducing instrument errors by the use of multiple instruments or reducing sampling errors by increasing the number of sampling locations, without sufficient knowledge of the relative importance of the random and bias components.

Certain factors will affect uncertainty yet have no effect on repeatability. Bias error components, which are known and traceable and can be eliminated by calibration, will reduce the overall instrument uncertainty but not affect its repeatability. Long-term repeatability is the untraceable, long-term variation in the bias error.

Based on steam turbine industry experience, the Committee has estimated repeatability as one-half the overall instrument uncertainty. Accordingly, the values for test repeatability given in Section 7 through 13 and summarized in para. 3.8.4 were determined by taking one-half of the calculated test uncertainty.

3.8.4 Repeatability of Sample Calculations. The following tabulation shows the repeatability of the tests recommended in each Section:

Section	Turbine Type	Type Test	Repeatability
7	Condensing, non-extraction, superheated inlet steam	Steam rate	±0.5%
8	Condensing, regenerative cycle, superheated inlet steam	Maximum capability	±0.5%
		Heat rate	±0.5%
9	Condensing, reheat-regenerative cycle, superheated inlet steam	Enthalpy-drop	±0.3% to ±0.5%
		Maximum capability	±0.3%
		Heat rate	±0.6% to ±0.9%
10	Condensing, regenerative cycle, saturated inlet steam	Heat rate	±0.7%
11	Noncondensing, non-extraction, superheated exhaust	Enthalpy-drop	±0.5%
		Maximum capability	±0.3%
12	Noncondensing, extraction	Enthalpy-drop	±0.6% to ±0.7%
		Maximum capability	±0.5%
		Steam rate without extraction	±1.8%
		Steam rate with extraction	±2.4%
13	Special procedures	Heat rate	±0.7%

SECTION 4 — INSTRUMENTS AND METHODS OF MEASUREMENT

4.1 GENERAL RECOMMENDATIONS

This Section provides recommendations of test instruments and methods capable of the repeatability required for consistent test results. Specific recommendations for primary and secondary instrumentation are made in Sections 7 through 13 for particular types of turbine.

Primary instruments are those devices that detect the parameter to be measured. Secondary instruments are the devices between the primary instrument and the indicating or recording device.

For routine performance tests, computerized data acquisition systems provide accurate, repeatable, and high-speed recording provided the primary and secondary instrumentation is calibrated. Transducers convert the primary signal to an electronic signal which is used as input to an analog to digital converter for use in a computer data acquisition system. When these systems are installed, their use for routine performance tests is recommended.

However, all turbines are not equipped with such systems. Therefore, this Section also describes instruments that require manual recording of the instrument indication.

Pertinent sections of the Code, Appendix A to the Code, and publications of the ASME Performance Test Code Committee on Instruments and Apparatus should be used for reference. Alternative methods of measurement may be used, provided their effects upon accuracy and repeatability are considered in the interpretation of test results. Report PTC 6R-1985 on Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines, should be used to evaluate these effects. This Report provides detailed guidance on selection of instruments for each type of measurement and the accuracy uncertainty associated with the instruments.

4.2 AC GENERATOR OUTPUT MEASUREMENT

Recommended methods for measurement of AC generator kilowatt (active power) output, listed in order of preference to achieve the best degree of repeatability, are described in this Section.

Generator loss curves provided by the manufacturer may require either the kilovolt-ampere (kVA) output or the kilowatt output and power factor of the generator to determine the generator loss at specified hydrogen pressure.

4.2.1 Active Power Measurements. Active power (kilowatts) should be measured with calibrated high-accuracy watt or watthour transducers with comparable accuracy solid-state electronic digital readout meters, used with separate potential and current transformers in each phase. Transducers shall be either three single-phase transducers or one three-element transducer. Analog and digital outputs to computerized data recorders are recommended. Potential transformers and current transformers should be calibrated with the equivalent burden of the transducers and meters, with no additional burden in the metering circuit.

4.2.2 Reactive Power Measurements. Reactive power (kilovars) should be measured with calibrated high-accuracy var or varhour transducers with comparable accuracy solid-state electronic digital readout meters, used with separate potential and current transformers in each phase. Transducers shall be either three single-phase transducers or one three-element transducer. Analog and digital outputs to computerized data recorders are recommended. The potential and current transformers can be the same ones used for active power measurement provided the transformers are calibrated with the equivalent burden of all watt, watthour, var, or varhour transducers and meters.

4.2.3 Power Factor Measurements. Power factor (cosine of the phase angle difference between the voltage and current) should be measured with calibrated high-accuracy power factor transducers with comparable accuracy solid-state electronic digital readout meters. Separate potential and current transformers in each phase are recommended. Transducers shall be either three single-phase transducers or one three-element transducer. Analog and digital outputs to computerized data recorders are recommended. The potential and current transformers can be the same ones used for active power measurement provided the transformers are calibrated with the equivalent burden of all watt, watthour, or power factor transducers and meters. An alternative to measuring kilowatts and power factor is to measure the volts and amps of each phase and the power factor.

4.2.4 Calibrated Rotating Standards. Calibrated rotating standards, calibrated single-phase test watthour meters, or a calibrated three-element test watthour meter, used with separate potential and current transformers in each phase may be used. Potential transformers and current transformers should be calibrated with the equivalent meter burden with no additional burden in the metering circuit. Meter integration is by an auxiliary photoelectric counter, or its equivalent, to record disc revolution. Mechanical-drive counting registers should not be used.

4.2.5 Calibrated Permanently Installed Watthour Meters. Calibrated three-element watthour meters permanently installed with separate potential transformer, but with current transformer used also for other instruments and relays, may be used. The potential transformers should be calibrated with the equivalent meter burden. Meter integration should be by mechanical register with smallest register unit less than 0.5 percent of the hourly generation at rated capacity.

4.2.6 Same as para. 4.2.5, but with potential transformers used also for other electrical instruments and relays. Changes in potential transformer burden can cause significant metering errors.

4.2.7 Same as para. 4.2.6, but with either 2½ element or two-element watthour meter instead of three-element watthour meter.

4.2.8 Instrument Transformer Location. Instrument transformers should be located so that the total generator output is measured. If any external taps exist between the generator and the point of measurement, supplementary metering of equivalent accuracy must be provided to determine the total generator output.

Supplementary readings shall be taken to obtain the generator power factor and hydrogen pressure if these data are required to establish the corrected generator terminal output. When definitions of turbine heat rate require measurement of excitation power or other auxiliary service supplied to the turbine-generator unit, Code procedures should be followed to measure those quantities which are used to adjust the generator terminal output to conform to the heat rate definition.

4.3 MECHANICAL-DRIVE OUTPUT MEASUREMENT

The Code, paras. 4.04 through 4.08, gives procedures for measurement of mechanical output using absorption dynamometers (reaction torque system) or transmission dynamometers (shaft torque meters). Measurement of thermodynamic properties and calculation of feedwater pump power is described in the Code, para. 4.09. See PTC 19.7, 1980 on Measurement of Shaft Horsepower for measurement of mechanical output by other means. For other types of driven equipment, refer to applicable ASME Performance Test Codes. For routine testing of prime movers, which have a connected load during the test, the transmission dynamometer is recommended. Absorption dynamometers absorb the prime movers shaft output and are not suitable for measuring shaft output when the prime mover is driving a connected load.

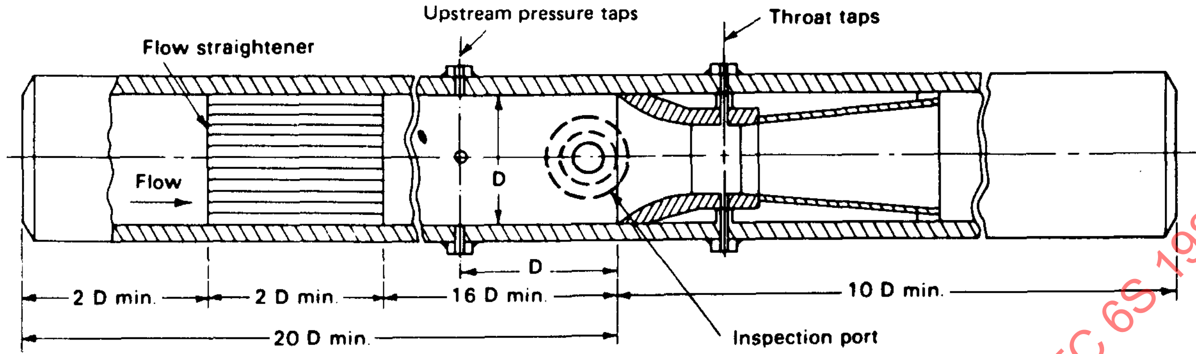
4.4 FLOW MEASUREMENT

Figures 4.1 (a), 4.1 (b), and 4.1 (c) illustrate the alternative designs recommended for flow measurement in Code tests.

4.4.1 Primary Flow Element. Measurement of water flow at either of the following locations is recommended as the basis for the accurate determination of the primary flow to the turbine.

(*a*) The primary flow element, with a flanged inspection port, shall be located in the feedwater system between the steam generator feedwater inlet and the highest pressure feedwater heater outlet. The high feedwater pressure may require that the primary flow element be welded into the feedwater piping system. The flanged inspection port provides access for inspecting the nozzle before and after the test and for cleaning the nozzle, if required, prior to the test.

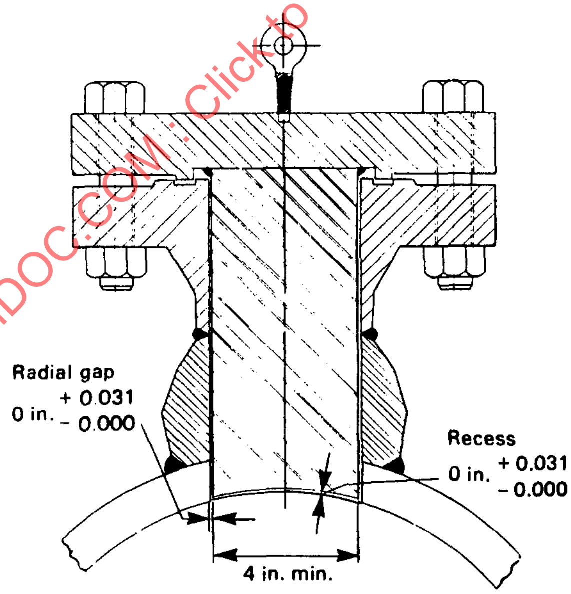
(*b*) The primary flow element located in the condensate system between the condensate pump



NOTE: No obstruction, such as thermocouple wells, backing rings, etc.

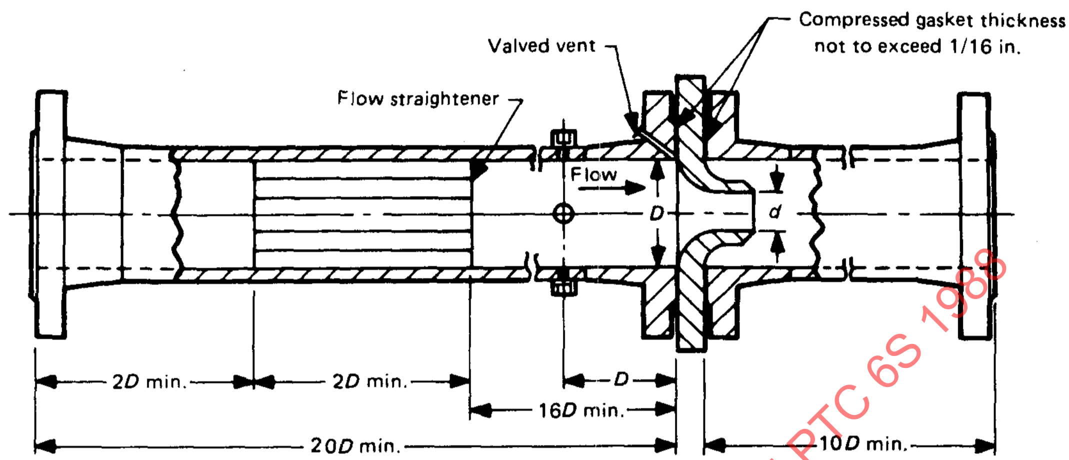
(This figure is diagrammatic and is not intended to represent details of actual construction.)

FIG. 4.1(a) WELDED PRIMARY FLOW MEASUREMENT SECTION



NOTE: The orientation of the nozzle on the pipe is determined by the designer.

FIG. 4.1(b) INSPECTION PORT FOR FEEDWATER FLOW NOZZLE



NOTE:
No obstruction, such as thermocouple wells, backing rings, etc.

FIG. 4.1(c) FLANGED PRIMARY FLOW MEASUREMENT SECTION

discharge and the feedwater pump suction. The condensate pressure is sufficiently low that a flanged flow element is typically used to facilitate removal for inspection or recalibration. Thus, the flanged flow metering section may be installed for the test only, and removed from service for normal continuous operation.

(c) The primary element for measuring water flow, listed in order of least uncertainty and best repeatability in the flow measurement, should be a throat-tap nozzle, pipe wall tap nozzle, orifice, or venturi tube meeting the specifications of Fluid Meters 1971, Part II, on Flow Measurement. The primary flow element should be installed in a specially designed flow metering pipe section including a flow straightener (Fluid Meters 1971, Part II) located upstream from the metering element. The complete flow metering pipe section should be calibrated as a unit before installation. For high capacity flows, the calibration should be extended to the maximum Reynolds number achievable in the calibration facility to minimize uncertainty due to extrapolation of the flow coefficient.

4.4.2 Flow Element Location in Cycle. The physical location of the primary water flow measuring element in the cycle is critical and will affect both the accuracy and repeatability of the flow measurement. No single location can be recommended because of the wide variations in cycle designs, piping arrangements, and costs. Alternate locations are shown on Fig. 4.2 and listed in Table 4.1 in order of preference from the standpoint of simplifying the test procedure and minimizing the readings required.

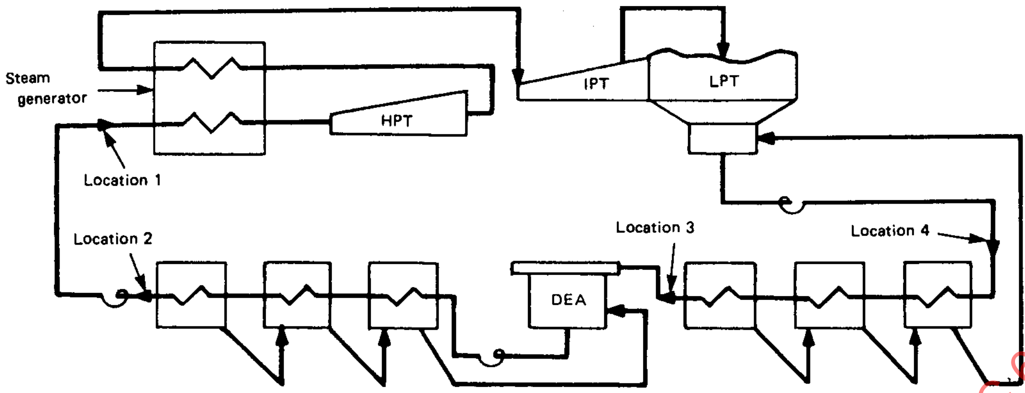
4.4.3 Flow Element Design Considerations for Least Uncertainty and Best Repeatability. The following design considerations for the primary flow element must be weighed to achieve a final design which will provide good repeatability of test results.

(a) The type of primary element should be selected so that a reasonably constant flow coefficient can be expected for the range of Reynolds numbers corresponding to the planned test load range.

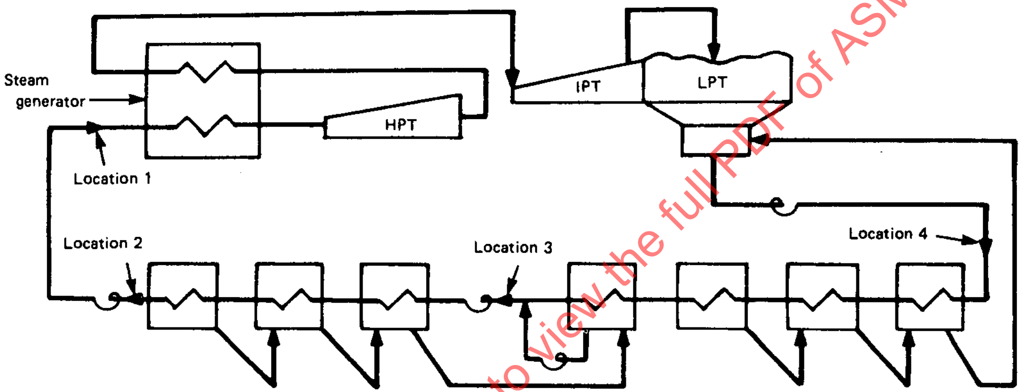
(b) To minimize uncertainty due to extrapolation of the coefficient curve, the Reynolds number range for calibration should be from the lowest value expected during the test to the highest value achievable by the calibration facility, or to the maximum value expected during the test if this value is less than the calibration facility's maximum capability. If the test Reynolds number is outside the range of the calibration facility, then the procedures given in the Code for extrapolation should be followed.

(c) The primary element should be sized to produce a pressure differential of at least 6 in. (153 mm) of mercury under water at the minimum test flow.

(d) The ratio of the flow nozzle throat diameter, or the orifice diameter, to the internal pipe diameter (Beta



Heat Cycle With Deaerator



Heat Cycle Without Deaerator

FIG. 4.2 ALTERNATE LOCATIONS FOR PRIMARY FLOW ELEMENT

ratio) should be within the limits of 0.25 (minimum) to 0.5 (maximum) for pipe wall tap flow nozzles and venturi tubes, and 0.3 to 0.6 for orifices. The Beta ratio for throat tap flow nozzles used in Code tests is bounded by 0.25 (minimum) to 0.5 (maximum).

(*e*) Limited experience with permanently installed feedwater flow nozzles, which have a flanged inspection port, show that nozzle deposits 6 to 10 mils thick can be removed through the inspection port to restore the nozzle performance to its original calibrated condition. The cleaning of the flow test section can be accomplished by the use of very high pressure water jet devices. Other cleaning methods may be used provided that they do not change the flow characteristics of the nozzle. Chemical cleaning should not be used unless the nozzle material is resistant to the chemicals.

4.4.4 Comparison of Secondary Flow Data to Primary Flow Measurement. When conducting routine tests on a flow section used in regular service, secondary data should be taken to compare the calculated primary flow with other flow measurements to detect deviations in trends as described below.

(*a*) From test data, compare primary flow with condensate and steam flows.

(*b*) From test data, compare calculated throttle flow with measured turbine first stage pressure. The correlation of first stage pressure and throttle flow established by previous test provides a useful indication of primary flow accuracy.

(*c*) From test data, a similar comparison using the measured pressure ahead of the reheat stop valve serves to check reheat flow.

TABLE 4.1
LOCATION OF PRIMARY WATER FLOW-MEASURING ELEMENT (Fig. 4.2)

	1	2	3		4
	At Steam Generator or Economizer Inlet	Between Final Feedwater Heater and Feedwater Pump Where Pump is Located Downstream of Feedwater Heater Outlet	With Deaerator At Deaerator Inlet	Without Deaerator At Feedwater Pump Suction Where Pump is Located Upstream of High-Pressure Heater	Between Condensate Pump and Lowest Pressure Feedwater Heater
ADVANTAGES					
Provides direct measurement of feedwater flow leaving cycle, thereby minimizing number of test measurements	X				
Eliminates potential flow errors due to recirculation of feedwater heater leakage, if present	X	X	X		
Eliminates potential flow errors due to recirculation of any leakage through feedwater pump recirculating valve to the deaerator	X		X		X
Allows use of low-pressure or intermediate-pressure flow metering instrumentation		X	X	X	X
Permits use of flanged flow metering section for easy removal for inspection or recalibration		X	X	X	X
Reynolds number may be within calibration range permitting direct use of calibration curve					X
Less sensitive to isolation problems such as feedwater heater drains emergency discharge to condenser	X				
DISADVANTAGES					
Requires use of high-pressure flow metering instrumentation	X				
Reynolds number range for test conditions may exceed range attained on calibration necessitating extrapolation of calibration curve	X	X	X	X	
Location probably requires welding the flow metering section in the line, making some repairs expensive and impractical for some devices	X				
Necessitates separate measurements of the gland sealing water supply and leak-off flows of the feedwater pump and any other pumps at the higher pressure levels of the cycle		X	X	X	X
Requires positive verification of tight closure of pump recirculating valve during test to assure zero recirculation to deaerator through flow metering section		X			

TABLE 4.1
LOCATION OF PRIMARY WATER FLOW-MEASURING ELEMENT (Fig. 4.2) (cont'd.)

	1	2	3		4
			With Deaerator	Without Deaerator	
	At Steam Generator or Economizer Inlet	Between Final Feedwater Heater and Feedwater Pump Where Pump is Located Downstream of Feedwater Heater Outlet	At Deaerator Inlet	At Feedwater Pump Suction Where Pump is Located Upstream of High-Pressure Heater	Between Condensate Pump and Lowest Pressure Feedwater Heater
Necessitates accurate pressure and temperature measurements around all feedwater heaters at the higher pressure levels for calculation of final feedwater flow by heat balance method			X	X	X
Requires positive verification of zero flow in recirculation line to condenser		X	X	X	X
Possibility of feedwater heater leakage in any or all heaters increases uncertainty of final feedwater flow rate				X	X
Proximity to pump discharge may subject flow metering instrumentation to unacceptable instrument oscillations from pump induced pressure pulsations					X

(*d*) From monthly performance data, compare computed steam generator efficiency, based on primary flow, with the level of turbine heat rate. A trend of increasing steam generator efficiency correlated with increasing (poorer) turbine heat rate indicates a drift of the primary flow measurement in the high direction.

(e) On cross-compound units where the steam leaving all turbine sections on one axis is superheated, it is possible to compare the turbine flow based on the primary flow measurement with a calculated flow using the output of the generator, related pressures and temperatures, and secondary flow measurements as described in the Code, para. 4.60. In general, this procedure can be used on cross-compound units with all low-pressure sections driving one generator but not for cross-compound units with low pressure sections driving two generators.

4.4.5 Steam Flow Measurement as an Alternate. An alternative method of determining the primary flow is to measure steam flow between the steam generator outlet and the turbine throttle. The absolute value of such a measurement is more uncertain than that of a water flow measurement because the primary element is usually uncalibrated and the discharge coefficient must be estimated from other sources. However, this approach may provide good repeatability for the specific case where superheated steam flow is measured with permanently installed flow nozzles.

Analysis of the test data available to the Committee yields a repeatability for this type of flow measurement over extended periods of time of ± 2 percent. It is recommended that water flow measurement be used for test purposes, and steam flow measurement be considered as a secondary check for comparison of chronological trends of repeated flow measurements.

4.4.6 Preferred Method of Flow Element Differential Pressure Measurement. The preferred method of measuring the differential pressure produced by the primary flow element is to use precision (0.1 percent of full accuracy) differential pressure transducers with comparable accuracy solid-state electronic digital readout meters. Analog outputs to computerized data recorders are recommended.

An alternative is to use a manometer to measure the differential pressure across the primary flow element. The manometer can provide equal accuracy to the

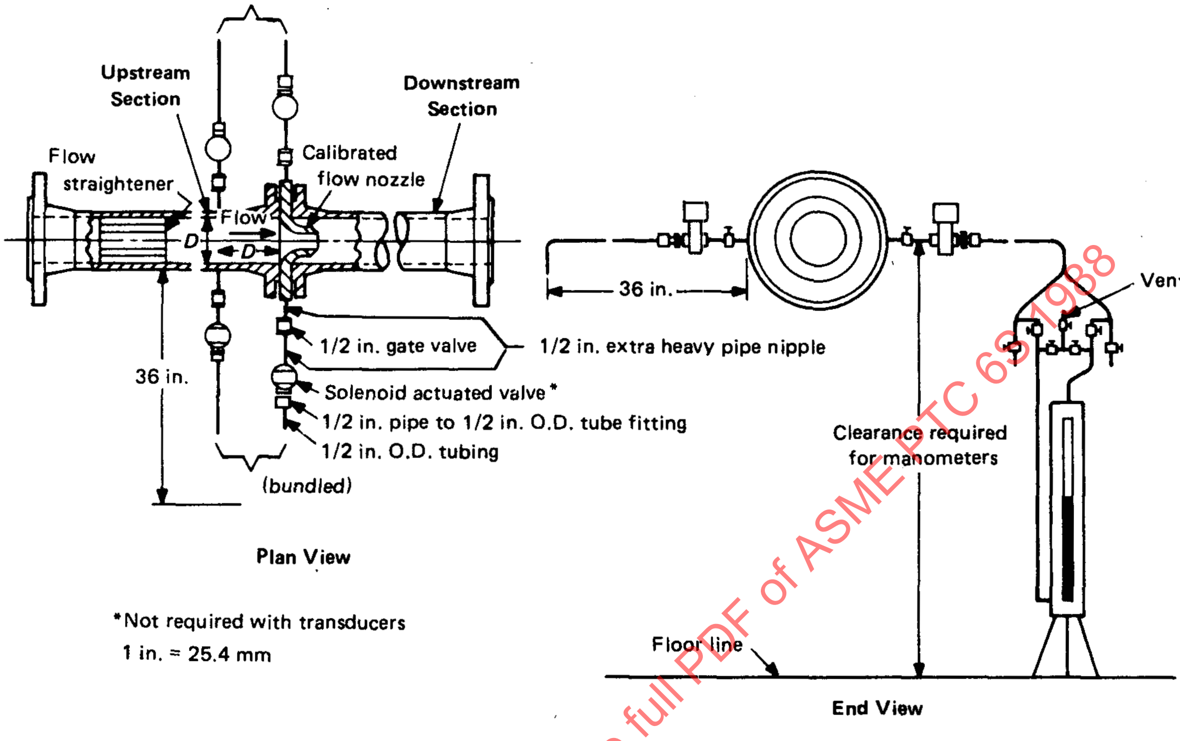


FIG. 4.3 CONNECTION BETWEEN CALIBRATED FLOW SECTION AND MANOMETERS

pressure transducers but may not provide computerized data logging capability. Regulations restricting or prohibiting the use of mercury may necessitate the use of other devices.

The differential pressure measuring devices should be calibrated before and after the test, at the conditions expected during the test.

4.4.7 Method of Connecting Flow Element to Measuring Device. Figure 4.3 illustrates the preferred method of connecting a precision transducer or manometer to the primary flow element. The test instrument should be located below the source and the connecting piping should slope downward toward the instrument. Connecting piping should be at least 3/8 in. (9 mm) ID. Both lines should be uninsulated and routed in close proximity to each other to minimize any temperature differential. Zero displacement solenoid operated valves may be installed in series with the source valves to eliminate column movement when reading the differential pressure. These valves should not be used with transducers for arresting readings but may be used for multiplexing instruments. Prior to the test, the instrument should be properly vented and checked at zero differential, and sufficient time must be allowed to permit the lines to reach equilibrium temperature after putting the manometer in service. The manometer should be readable to ± 0.05 inch (1.25 mm).

4.4.8 Secondary Flow Measurements. For secondary flow measurements required for heat balance purposes or detailed analysis of cycle performance, uncalibrated flow metering sections, meeting the specifications of Fluid Meters 1971, Part II, may be used.

For computerized data recording, differential pressure transducers and solid-state electronic digital readout devices are recommended.

Secondary flows may be read directly using calibrated indicating or recording flowmeters or calculated from differentials read from manometers. For low flows and low-pressure steam packing leakage flows, suitably designed forward-reverse pitot tubes may be used.

4.5 PRESSURE MEASUREMENT

4.5.1 Primary Pressure Measurements. Primary pressures which significantly affect the turbine test results should be measured with precision test instruments.

Precision test instruments are defined as commercially available instruments having the following measurement uncertainty:

Instrument	Measurement Uncertainty
(1) Pressure transducers	0.1% of full scale
(2) Deadweight gages	0.1% of measured pressure
(3) Manometers	Considered a primary standard
(4) Absolute pressure gages	0.25% of full scale reading
(5) Bourdon gages	0.25% of full scale reading
(6) Barometer	Considered a primary standard

The following instruments are recommended.

Instrument	Pressure Range
(1) Calibrated pressure transducer with suppressed range for accurate readout[2]	0 to 5000 psia (34470 kPa)
(2) Calibrated deadweight gage	Above 35 psia (240 kPa)
(3) Precision manometers[1]	0 to 35 psia (240 kPa)
(4) Precision absolute pressure gage	0 to 2 psia (14 kPa)
(5) Laboratory type bourdon test gage[2]	35 to 1000 psia (240 to 6890kPa)
(6) Precision barometer[3]	Atmospheric pressure

NOTES:
(1) See para. 4.02 in the Code for precaution on the use of mercury as the manometer fluid.
(2) Must be calibrated against a secondary standard immediately before and after the test, to ensure an accurate calibration. Transducers should be temperature-compensating or located in a temperature controlled environment.
(3) See PTC 19.2 - 1987 on Pressure Measurement.

The actual choice of instrument is optional where the specified pressure ranges overlap.

Precision pressure transducers with comparable accuracy, solid-state electronic digital readout devices, and an output to a computerized data recorder are recommended for routine performance test.

4.5.2 Secondary Pressure Measurements. Secondary pressures, which have a minor effect on test results, may be measured using standard plant instrumentation. These instruments should be maintained in good condition and recalibrated at least once every 6 months.

4.5.3 Pressure Instrument Piping. Piping connecting the pressure-measuring instruments with the pressure source should be at least ⅜ in. (9 mm) inside diameter or equivalent tubing. For details of installation of pressure-source taps and arrangement of connecting piping, refer to the Code, paras. 4.84 through 4.89. Pressure multiplexing valves may be used to select the pressure signal connected to the valve to allow selected pressure signal transmittal to a pressure transducer. The multiplexing valve should be located as close as possible to the pressure transducer. After each pressure signal source is selected, the pressure should be allowed to stabilize before reading or recording the pressure.

4.5.4 Test-Pressure Connection Precautions. Test-pressure connections not in continuous service should be thoroughly blown down before the start of the test to ensure clear lines and to permit time for the water legs to reach equilibrium temperature. Subatmospheric connections should be properly vented to clear the lines of accumulated mositure and should be provided with minimum-flow air bleeds to purge the lines except when readings are made. Similar purging is recommended for connections slightly above atmospheric pressure, especially when the instrument is located above the source connection.

4.5.5 Deadweight Gages. For periodic tests, deadweight gages of good quality do not require routine calibration. The gages should be examined during the test to ensure that the pistons rotate freely. If a gage error is suspected, the calibration may be checked by the following:

(*a*) checking against a deadweight tester;

(*b*) checking the gage against a column of mercury, and by comparing the weights against each other on a precision balance.

Pressure transducers and Bourdon gages for primary measurements should be calibrated in-place by a deadweight tester before and after each test. Transducers and gages for secondary pressure measurements should be calibrated with a deadweight tester every 6 months.

4.5.6 Manometers. Manometers may be of the U-tube type with a scale to read the level in each leg, or single-leg well-type manometer with a compensated scale indicating the correct differential height between the indicating fluid column and the closed reservoir level. If mercury is used, it should be a clean instrument grade mercury. Regulations restricting or prohibiting the use of mercury may necessitate the use of other indicating fluids. Before filling a manometer, it should be flushed and thoroughly cleaned of residual deposits. Care must be exercised in filling well-type manometers with the specified quantity of indicating fluid to establish properly the zero level and to avoid entrapped air in the closed reservoir. After filling, the manometer should be exercised by application of pressure, and the zero level checked for a consistent reading. The manometer should be readable to ±0.05 in. (1.25 mm).

4.5.7 Precautions When Using Mercury in Manometers. Mercury is the manometer fluid used for primary pressure measurement. For secondary pressure measurement, commercial fluids of known specific gravity may be substituted to expand the scale reading, and thus reduce the reading error. Precautions concerning the use of mercury in instrumentation are discussed in para. 4.02 of the Code.

4.5.8 Exhaust Pressure Measurement Methods. The exhaust static pressure of a condensing turbine is to be measured at, or on either side of and adjacent to, the exhaust joint. The Code recommends that no fewer than two pressure measuring devices should be used per exhaust annulus. If possible, pressure measuring points should be selected which will indicate the average pressure of the entire exhaust area. If acceptance tests were run using Code procedures, the sampling points at each exhaust which produced readings closest to the average should be retained for periodic testing. The initial location of the pressure-measurement device requires judgement. Basket-tip pressure sensors, installed at a 45 degree angle to the flow as shown in Fig. 4.4(a), are recommended. Their location should be at the plane of the exhaust joint, downstream of the plane of the discharge annulus, and approximately on the center line of the turbine shaft. Close proximity to extraction steam piping or internal structural supports should be avoided. The exact location should be adjusted so that the basket tip is in the center of an unobstructed steam flow path from the exhaust to the condenser. The basket tip should be rigidly supported by anchoring its connecting piping. This piping should slope continuously upward to an external valve at the turbine casing just above the turbine floor level.

Guide plates shown in Fig. 4.4(b) should be arranged so that the steam flow is perpendicular to the pressure tap. The pressure corresponding to the average exhaust hood temperature may be used as a secondary check to verify the repeatability of the exhaust pressure measurement. Since these instruments may be at different locations, a pressure differential may exist.

4.5.9 Exhaust Pressure Measurement Devices. Absolute pressure gages, or standard mercury manometers may be used for the exhaust pressure measurement. In either case, these should be equipped with scales and verniers that permit reading to ±0.01 in. (0.25 mm). Absolute pressure gages must be scrupulously clean before filling with instrument grade mercury. Before the test, the gages should be compared with each other when connected to a common vacuum source, or compared with a precision manometer and barometer.

4.5.10 Corrections to Pressure Measurements. Pressure, whether by deadweight gage, Bourdon gage, or mercury manometer, shall be the algebraic sum of the following:

(*a*) the instrument reading, using the proper conversion factors for the measuring fluid. Refer to Fluid Meters 1971, Fig. II-1-2.

(*b*) the negative correction for manometer temperature to 32°F (0°C)

(*c*) the instrument correction, including any scale correction required

(*d*) the gravity correction to reduce the reading to the value which would be obtained if gravity at the location of the instrument had the International Standard Value of 32.17406 ft/sec^2 (9.80665 m/s^2)

(*e*) the water column correction

(*f*) the measured barometric pressure including the correction to the elevation of the gage

4.6 TEMPERATURE MEASUREMENTS

4.6.1 Primary Temperature Measurement Instruments. The following instruments are recommended for primary temperature measurement (see Code paras. 4.100 through 4.107):

4.6.1.1 Standardized Resistance Thermometer. A standardized resistance thermometer with a precision resistance bridge, digital multimeter or data acquisition system, installed for test only.

4.6.1.2 Test Thermocouples With Water-Ice Bath Cold Junction. A calibrated thermocouple constructed

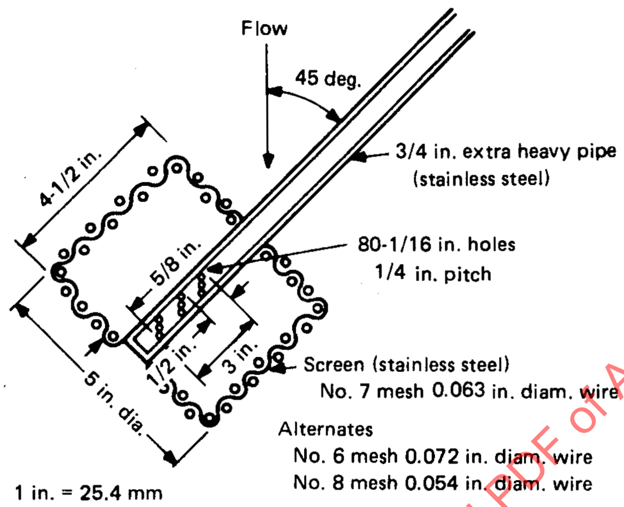


FIG. 4.4(a) BASKET TIP

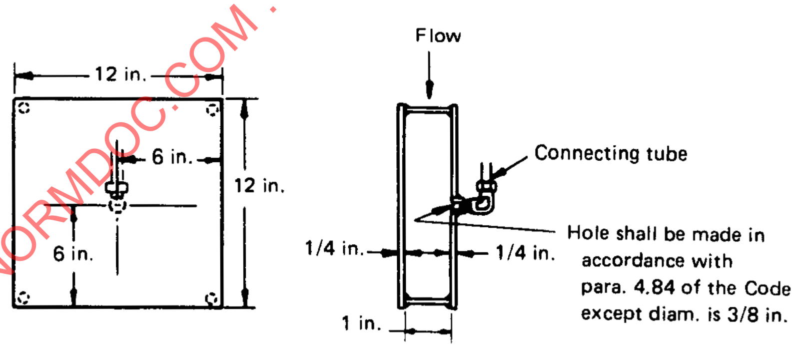


FIG. 4.4(b) GUIDE PLATE

with continuous leads having an integral cold junction and a balanced precision potentiometer, digital voltmeter, or data acquisition system, installed for test only, with a water-ice bath to maintain the solid-state cold junction at the ice point.

The water-ice mixture should be stirred frequently to eliminate temperature gradients and the temperature verified with a mercury-in-glass thermometer.

When using digital voltmeters, proper guarding procedures shall be followed to minimize stray electric signal errors.

4.6.1.3 Test Thermocouples With Electronic Reference Junction. A calibrated thermocouple constructed with continuous leads to an electronic reference junction and precision potentiometer, digital voltmeter, or data acquisition system. These thermocouples are installed for test only.

4.6.1.4 Test Thermocouples With Temperature Controlled Cold Junction. A calibrated thermocouple connected to permanently installed lead wire of the same materials as the thermocouple, and a temperature controlled cold junction located at a common terminal board for a group of test thermocouples. The cold junction temperature should be measured with a calibrated instrument, and access to the reading without opening the terminal board enclosure.

The millivolt output of each thermocouple should be read, using either a precision potentiometer or digital voltmeter properly compensated for the cold junction temperature, by sequentially connecting each thermocouple through a rotary test switch or by suitable test jacks.

These special test thermocouples should not be left in continuous service but should be removed from their well and reinstalled for each test.

4.6.2 Secondary Temperature Measurements. Secondary temperature measurements, having a minor effect on the test result, may use station thermocouples connected to recording or indicating instruments. Before testing, these instruments should be checked to ensure correct standardization. Use of mercury-in-glass thermometers is recommended only for temperatures of 200°F (100°C) or lower.

4.6.3 Measurements to Determine Steam Enthalpy. Special attention is required in the location of test points for pressure and temperature measurements where these readings are used to determine steam enthalpy. Pressure taps should be located as close as possible to the point of corresponding temperature measurement. Duplicate temperature measurements should be made at each point where the steam enthalpy is a primary factor in the calculation of the test result. If more than one steam line serves a common location, duplicate measurements should be provided in each line. For acceptable accuracy, the steam should be superheated 25°F (14°C) or more at the point of measurement. Care should be taken that the temperature reading is representative of the mixed flow in the pipe.

For steam exhausting from turbine sections, the temperature measurement should be downstream of an elbow or tee to provide mixing of the stratified steam leaving the turbine steam path.

For turbine extraction steam temperature measurement, the possibility of internal or external steam bypass flows which could affect the main flow should be checked, and precautions taken to locate the temperature measurement point accordingly.

In pipes less than 4 in. (100 mm) internal diameter, the thermocouple well should be positioned axially by insertion in an elbow or tee.

4.6.4 Thermowells. Unless limited by design consideration, the temperature sensitive element should be immersed in the fluid at least 3 in. (75 mm), but not less than one-quarter of the pipe diameter. Thermometer wells shall be installed in conformity with the ASME Boiler and Pressure Vessel Code and Power Piping Code. Tubes and wells shall be as thin as possible consistent with safe stress and other ASME Code requirements, and the inner diameters of the wells must be clean and dry and free from corrosion or oxide. The well material should resist corrosion. A clean well will permit easy removal of the thermocouple. The internal diameter of the well should be as small as possible so the thermocouple will fit closely in the well and be firmly held against the bottom. The portion of the thermocouple well external to the pipe should be carefully covered with heat insulation to minimize radiation losses.

4.6.5 Calibration of Temperature Measuring Devices. Primary test thermocouples or resistance thermometers should be recalibrated at periodic intervals. The frequency of the recalibration should be determined by experience. The Code, para. 4.106, recommends suitable calibration procedures.

Secondary temperature measurements should be checked by comparison with primary test thermocouples whenever the indicating temperature is in doubt. This can be done in the field under steady-state conditions by interchanging the permanent thermocouple and a test thermocouple in the same well and comparing the readings.

4.7 STEAM QUALITY MEASUREMENT

Steam samples should be taken in accordance with PTC 19.11, 1970 on Water and Steam in the Power Cycle (Purity and Quality, Leak Detection and Measurement). The following methods may be used, as applicable, to determine steam quality.

(*a*) tracer — tracer technique, radioactive or non-radioactive, (for throttle steam and extraction steam). For details refer to the Code, Par. 4.109.

(*b*) heater drain flow and heat balance — heater drain flow measurement (extraction steam only) and heat balance

(*c*) throttling calorimeter — throttling calorimeter for direct determination of quality (throttle steam only)

Selection of one of the above methods for determining steam quality is dependent upon many conditions and each method has limitations which govern its use. The radioactive tracer method is currently used for many nuclear fueled power plant turbines. The Code, para. 4.109 through 4.115, provides details regarding its use.

The principal disadvantage of the calorimeter is that its accuracy is directly affected by the extent to which the steam sample represents the average condition of the steam flowing in the pipe. Figure 4.5 illustrates a moisture sampling tube which is designed so that the average velocity through the eight sampling holes is approximately equal to the average steam velocity in the pipe. In large pipes, there is evidence that the moisture in the steam tends to collect along the pipe wall thus escaping isokinetic sampling. Techniques for steam sampling are described in PTC 19.11, 1970, I&A, Part II on Water and Steam in the Power Cycle (Purity and Quality, Leak Detection and Measurement).

4.8 TIME MEASUREMENT

The time of test periods and other observations may be determined by the following:

(*a*) for computerized data acquisition systems, an electronic clock built into the computer can provide automatic, accurate time measurements;

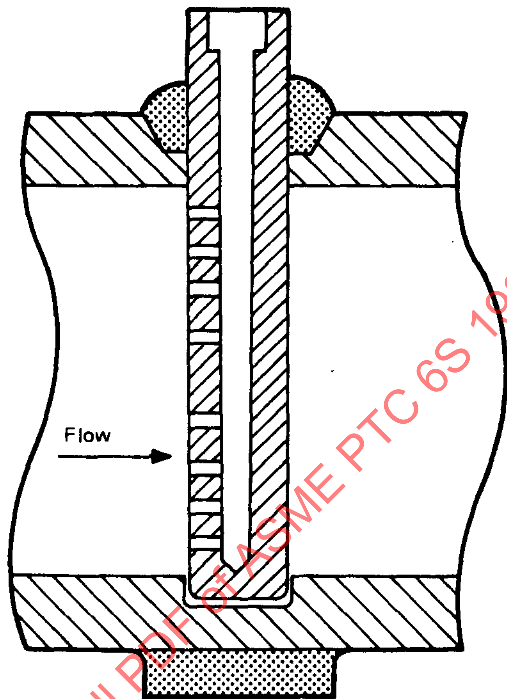


FIG. 4.5 MOISTURE-SAMPLING TUBE

(*b*) signals from a master clock or timekeeper, if the tests require the starting or stopping of certain readings simultaneously;

(*c*) observations of synchronized watches by the individual observers;

(*d*) radio communication of the master time signal.

When signals from a master clock are used, the signaling means shall be permit signal receipt within 0.5 sec of the desired time.

Reliable watches or clocks having an accuracy within one minute per day shall be employed. Watches and clocks shall by synchronized at the start of the test.

4.9 CYCLE LEAKAGES AND STORAGE CHANGES MEASUREMENTS

Water or steam leakages from the plant cycle between the point of primary flow measurement and the turbine throttle must be measured to a degree of accuracy consistent with the primary flow. Typical leakages are pump leakages, injection flows, auxiliary steam flow, steam generator blowdown, and valve stem leakages.

Low temperature leakages, such as pump shaft leakage, should be collected for a timed period and weighed to establish the leakage rate.

In a similar manner, any system storage in the area of the cycle should be calibrated for changes in level and the levels recorded during the test period. Typical items are condenser hotwell storage, deaerator storage, and steam generator drum storage.

4.10 STATION INSTRUMENTS FOR SECONDARY READINGS

Station instruments can be used for secondary readings which have a minor effect on the test result provided they are properly maintained and calibrated within 6 months before the test.

Detailed information on instruments is contained in PTC 19, Series on Instruments and Apparatus.

4.11 MANUFACTURER'S DATA FOR TEST CALCULATIONS

The equipment manufacturer may provide data and/or correction curves which are pertinent to the test calculation. Some of these data are as follows.

(*a*) electrical losses correction curves — electrical losses relative to generator power factor and hydrogen pressure to adjust the generator output to rated conditions. Electrical losses that are determined from manufacturer's curves may require measurement of the active power (kilowatts) and either reactive power (kilovars) or power factor to determine generator kVA.

(*b*) turbine heat rate or steam rate corrections — turbine heat rate or steam rate corrections to adjust the test rate to standard conditions defined by the heat/steam rate formula

(*c*) turbine load corrections — turbine load corrections to adjust the test output to standard conditions defined by the heat/steam rate formula

(*d*) unmeasurable secondary flows or steam leakage — unmeasureable minor secondary flows or steam leakages which affect the calculation of the throttle flow

(*e*) pump efficiency curves — pump efficiency curves which establish the pump enthalpy rise under test condition

(*f*) thermal inputs to cycle — minor thermal inputs to the cycle from heat exchangers such as air ejector condensers, generator hydrogen coolers, or oil coolers which cannot be conveniently measured as part of the test

4.12 ROTATING SPEED MEASUREMENT

Shaft speed shall be measured by use of an integrating counter conforming to PTC 19.13 - 1961 on Measurement of Rotary Speed, para. 4.118. A stroboscopic device may be substituted if a calibrated accuracy of 1% can be demonstrated. Additional details are contained in the Code, para. 4.117.

For mechanical drives, such as feedwater pump prime movers, where power is determined by measuring shaft speed and torque, it is essential to measure accurately the shaft speed. This is required because an error in the shaft speed measurements will produce an error in calculated shaft power which is directly proportional to the error in the shaft speed measurement. For mechanical drives, shaft speed shall be measured by utilizing a digital or analog tachometer system. This system consists of three basic components as follows:

(*a*) the gear or key on the shaft, considered the exciter. The exciter is utilized in conjunction with an electronic-integration counter as discussed in the Code, para. 4.118, and shall be constructed of ferrous material.

(*b*) the magnetic or eddy current speed transducer which converts the rotating exciter surface depth variations into an AC voltage signal with no mechanical coupling to the shaft

(*c*) the electronic counter or tachometer. It is the device that receives the AC signal from the pickup, completes the computation, and conditions the signal for readout purposes. For computerized data recording, an output to the computer is also required.

SECTION 5 — PRELIMINARY TEST

5.1 LOCATION OF TURBINE VALVE POINTS

5.1.1 General. The turbine valve points must be determined in order to establish turbine performance over the load range. If any test throttle flow is obtained with a valve only partially open, flow variations from test to test may result, leading to inconsistent test results. A savings in personnel and time can be realized if the valve points are determined prior to the start of the turbine test series.

A valve point may be established in terms of high-pressure turbine efficiency, certain measured turbine pressures, or valve stem positions. The turbine is tested at the established point.

5.1.2 High-Pressure Turbine Efficiency Method. For units with a high-pressure section operating entirely in the superheat region, a valve point may be located by finding a point of local maximum high-pressure turbine efficiency. To do this, the flow to the unit is changed in small increments throughout a range which includes the valve points. At each flow increment, pressure and temperature measurements are taken at both inlet and exhaust so that high-pressure turbine section efficiency can be derived. A local maximum efficiency will be evident, provided suitable instruments and test procedures have been used. During testing, a parameter which varies with flow [such as governing valve position(s), or pressure ratio across either the first stage or the complete high-pressure section] should also be recorded so the valve point can be readily set during the test series.

5.1.3 Measured Turbine Pressure Method. Valve points may be established by measuring steam pressures using a pressure tap provided for each steam admission zone, downstream of the governing valves. While the valve remains closed, the pressure in the zone below it will be nearly the same as the first stage exit pressure. As the valve opens, the zone pressure builds up and the difference between it and the first stage pressure changes. The valve point is defined at this point of change.

Figure 5.1 illustrates data from this type of test. On this illustration, (A) is the pressure downstream of the initial control valve(s), (B) is the admission zone steam pressure for a specific valve, and (C) is the first stage pressure. The valve point occurs when a change in the pressure difference between first stage pressure (C) and admission zone pressure (B) is detected as the valve starts to open. Normally, with the governing valves properly set, the next succeeding valve starts to open when the admission zone pressure rises (B) to within a few percent of the throttle pressure.

5.1.4 Valve Stem Position Method. Valve positions on turbines may also be determined by utilizing a precision depth gage to determine the individual valve stem lifts. In addition, a reference scale to indicate the travel of the servo driving the valve cam shaft should be installed. The physical valve cam and roller position is helpful in locating the approximate valve positions. During the preliminary test, sufficient readings are taken to establish the throttle flow at each test point, and turbine stage pressures are read. The test is run by raising the turbine load in suitable small increments throughout the entire load range. At each point, the load is held constant by setting the turbine load limit with the speed control out of range. At each load point, the following readings are taken:

(*a*) position of each turbine governing valve by depth gage measurement;
(*b*) position of cam shaft servo drive;
(*c*) turbine stage pressures;
(*d*) turbine load (for reference only);
(*e*) load limit control oil pressure.

Figure 5.2 shows data from such a test indicating the valve point loading. Turbine throttle flow is used as a parameter. In using this data for setting up periodic turbine tests, the servo position is used. Lines are scribed on the position scale indicating the valve opening points, and the test loads are set up at scale positions slightly below the reference marks. If maintenance during turbine inspections included work on the turbine valve control, the test just described must be repeated to reestablish the location of the turbine valve points. A

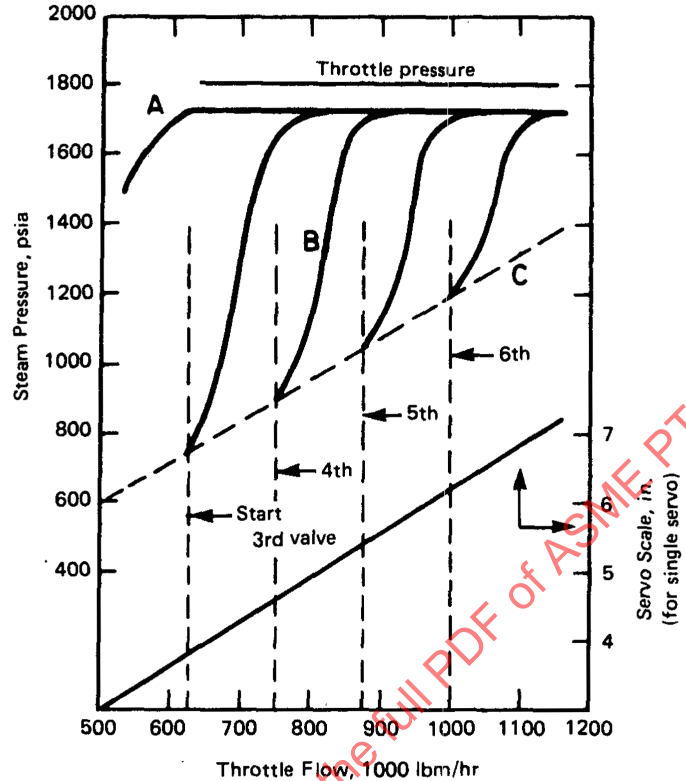


FIG. 5.1 TYPICAL TURBINE VALVE-POSITION TEST DATA BASED ON INDIVIDUAL STEAM PRESSURE MEASUREMENTS

check of an individual valve must include a check of the next sequential valve to ensure that it has not lifted.

5.2 TEST FOR CYCLE ISOLATION

In this test the cycle isolation procedure (para. 3.2 of this Report and paras. 3.11 through 3.17 of the Code) is followed and all flows, in and out of the cycle, are read for a period of at least 1 hr. Changes in cycle storage are carefully checked and the unaccounted leakage is determined. The unaccounted leakage should amount to less than 0.5 percent of the maximum throttle flow and every effort should be made to minimize it. In the calculation of heat rate, the unaccounted leakage is usually assumed to be from the steam generator and the amount is subtracted from the measured feedwater flow to calculate throttle flow and heat input. When the assumption is true, the impact on test heat rate is small and comparisons to previous tests should be valid. If however, the leakage occurs elsewhere in the cycle, and particularly if it is located upstream of the primary-flow measurement, then the effects of unaccounted leakage or significant changes in leakage should be investigated to validate the assumption of the source of the leaks. It is desirable that this test be made under steady load conditions simulating an actual test run.

5.3 TURBINE SECTION EFFICIENCY BY THE ENTHALPY-DROP METHOD

5.3.1 Measurements. Measurement of turbine section efficiencies by the enthalpy drop method is especially useful as a guide to the cleanliness of the turbine steam path. Immediately following an inspection of the turbine, when the steam path is of known cleanliness, a test should be made as soon as the turbine is loaded to valves wide open and normal operating conditions are established. However, if extended operation at reduced pressure is required, a special test should be made when 50 percent of normal throttle pressure is obtained. At each point where superheated steam enters or leaves the turbine casing, pressures and temperatures are recorded using accurate test instruments. In

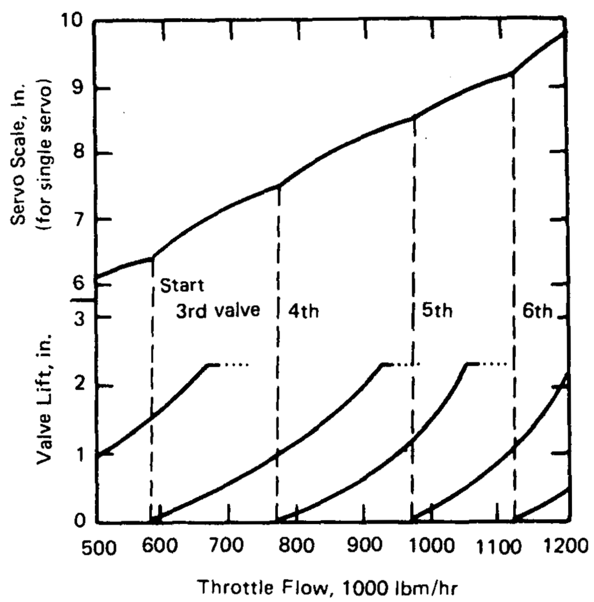


FIG. 5.2 TYPICAL TURBINE VALVE-POSITION TEST DATA BASED ON INDIVIDUAL VALVE-LIFT MEASUREMENTS

addition, readings are taken to establish turbine throttle flow and first-stage pressure.

For a reheat turbine, possible locations for measurements are as follows:

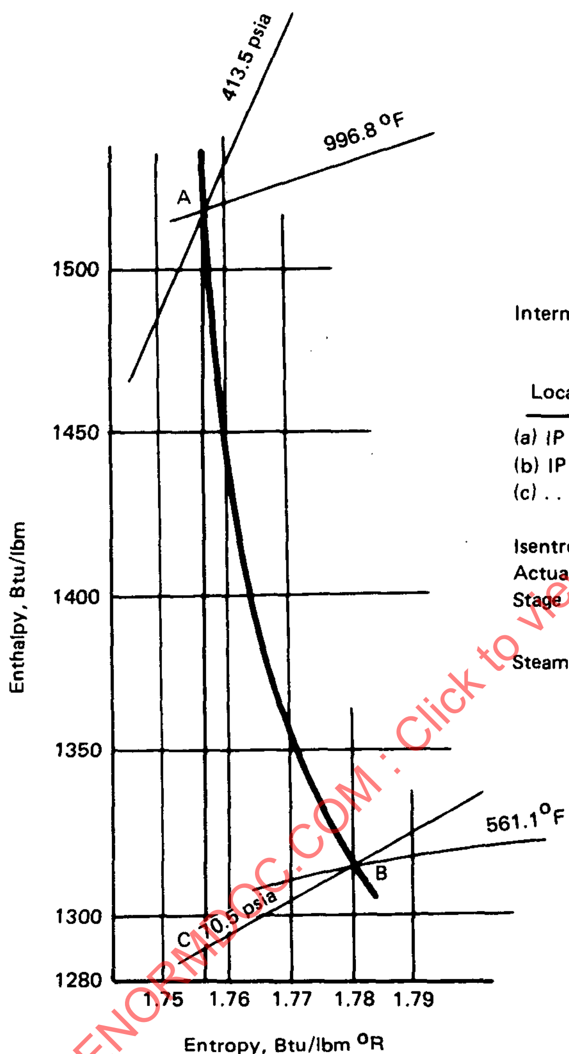
(*a*) turbine throttle valve inlet
(*b*) high-pressure turbine exhaust (cold reheat)
(*c*) intermediate-pressure turbine inlet (hot reheat)
(*d*) low-pressure turbine inlet (after crossover)

From the test data, turbine section efficiencies can be calculated, as illustrated in Fig. 5.3. A one hour test run under stabilized conditions is sufficient for this type of test.

Measurements and calculations of intermediate pressure turbine efficiency on opposed flow turbines should account for the effect of high-pressure turbine to intermediate pressure turbine shaft leakage.

5.3.2 Indicators of Turbine Cleanliness. Repeating the turbine section efficiency test at periodic intervals will indicate the turbine cleanliness. Lower section efficiencies, and decreasing throttle flow when adjusted to standard conditions, are indications of turbine steam path deposit accumulation. Also, a significant increase in low pressure feedwater heater pressures is indicative of steam path fouling of the low pressure turbine section. The condition of the low pressure section generally cannot be measured by this method due to insufficient superheat in the turbine exhaust steam.

5.3.3 Low Pressure Turbine Enthalpy-Drop Tests. It may be possible to conduct low pressure turbine enthalpy-drop tests at light loads on some steam turbines by increasing the exhaust pressure while holding a reheat temperature to obtain a dry exhaust. Since absolute pressures in the order of 4.5 to 5.0 in. of mercury are necessary for this method, both the turbine manufacturer and the condenser manufacturer should be consulted regarding operating limitations and the feasability of conducting this test before it is attempted. (See ASME paper 60-WA-139 for further details on this method.)



Intermediate-Pressure-Turbine Enthalpy Drop

Location	Press., psia	Temp., °F	Enthalpy Btu/lbm	Enthropy Btu/lbm °R
(a) IP inlet	413.5	996.8	1521.1	1.758
(b) IP EXH	70.5	561.1	1312.4	
(c)	70.5	. . . .	1290.5	1.758

Isentropic enthalpy drop = 1521.1 – 1290.5 = 230.6 Btu/lbm
Actual enthalpy drop = 1521.1 – 1312.4 = 208.7 Btu/lbm
Stage Group Efficiency = (208.7/230.6) 100 = 90.5% %

Steam flow at IP inlet = 939,000 lbm/hr

FIG. 5.3 STAGE GROUP EFFICIENCY BY ENTHALPY-DROP METHOD

SECTION 6 — PRESENTATION AND INTERPRETATION OF TEST RESULTS

6.1 GENERAL

One of the most important objectives involved in a turbine testing program is the proper interpretation of the test results. It permits sound judgment as to corrective action that need be taken when there is indication of mechanical damage which may require prompt removal of the unit from service, or alternatively scheduling a future outage for internal inspection. In either case, proper interpretation serves to determine the degree of outage urgency, and helps determine the spare parts needed to restore the unit to its normal efficiency level. Some test results, which indicate deficiency in load-carrying ability, increased packing leakage, or decreased section efficiency, are relatively easy to evaluate, but knowledge of the turbine characteristics is necessary to understand the reason for a performance change. It is important to understand the variations in section efficiencies, stage pressures, and stage temperatures which occur due to changes in load, throttle pressure and temperature, reheat temperature, condenser pressure, or due to the removal of a heater from service. In this Section, numerous thermodynamic and flow equations have been used to facilitate the discussion of turbine test performance. Although these equations can be found in engineering texts, selected equations with derivations are included in paras. 6.25(a) through 6.25(d).

6.2 PRESSURE, TEMPERATURE, AND FLOW RELATIONSHIPS

The general flow equation for all turbine stages can be expressed as follows:

$$w = (3600\ (C_q)\ (A_n)) \times \sqrt{(2g)\left(\frac{\gamma}{\gamma-1}\right)\left(\frac{p_1}{v_1}\right)\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma+1}{\gamma}}\right]} \quad (1)^*$$

*See para. 6.25(a) for derivation of Eq. (1).

Most stages, including all those between the first and last stage, operate at a nearly constant pressure ratio under changing governing valve setting, throttle flow, condenser pressure, and throttle steam conditions.

For these stages, by assuming a constant p_2/p_1, and by ignoring the very small changes in γ and A_n, Eq. (1) becomes:

$$w = C_q \times \text{constant} \quad \sqrt{p_1/v_1} \quad (2)$$

Although C_q varies slightly with Reynolds number, practically it can also be considered a constant. Thus:

$$\frac{w}{\sqrt{p_1/v_1}} = \text{a constant} \quad (3)$$

or

$$\frac{w}{p_1\sqrt{\dfrac{1}{R_1\,T_1}}} = \text{a constant} \quad (4)$$

where, in the above and in Fig. 6.1:

- w = rate of flow, lbm/hr
- C_q = flow coefficient
- A_n = nozzle area, ft^2 (stationary-blade flow area)
- γ = ratio of specific heats (c_p/c_v)
- p_1 = stage-inlet pressure, psia
- p_2 = pressure between stationary and rotating blade rows, psia
- p_3 = stage-outlet pressure, psia
- R_1 = universal gas constant at stage inlet
- g = acceleration due to gravity, ft/sec^2
- v_1 = specific volume at stage inlet, ft^3/lbm
- T_1 = absolute temperature at stage inlet, °R
- V = velocity, ft/sec

The relationship shown in Eqs. (3) or (4) may also apply to the first stage, depending on the operation of the governing valves.

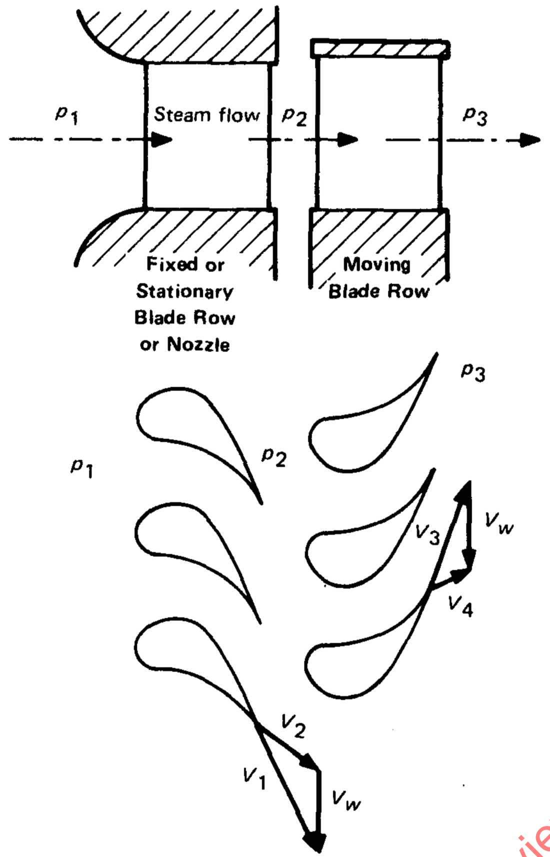


FIG. 6.1 TYPICAL BLADING DIAGRAM FOR SINGLE STAGE

6.3 FIRST-STAGE RELATIONSHIPS

In a turbine designed for partial-arc admission, the first-stage nozzle area is divided into separate areas, each served by one or more governing valves. When the turbine is operating with all valves wide open, the flow is a maximum. When a valve is closed, the flow area to the first stage is decreased and, therefore, the total flow will decrease. Flow through the second stage also decreases, and Eq. (4) shows that second-stage inlet pressure, usually referred to as first-stage pressure[1], decreases. The pressure ratio across the first stage decreases because the pressure ahead of the first stage remains approximately equal to throttle pressure. Because the pressure ratio changes with changing valve setting, Eqs. (3) and (4) are valid for the first stage only if the valve setting is constant. Typical expansion lines for the high-pressure section of a partial-arc admission condensing turbine are shown in Fig. 6.2.

6.3.1 Application of Eqs. (3) and (4) to the first stage of a full-arc admission turbine is valid under changing conditions. When the turbine is operated at a constant throttle pressure, the governing valves are closed in unison to reduce flow from the valves-wide-open flow. Pressure drop across the valves increases with a resulting increase in throttling loss. First-stage inlet pressure and first-stage flow decrease, and the pressure ratio across the first stage remains constant. Typical expansion lines for the high-pressure section of a full-arc admission condensing turbine are shown in Fig. 6.3.

6.4 LAST-STAGE RELATIONSHIPS

For the last stage of a condensing turbine or for any turbine with a fixed exhaust pressure, as the inlet pressure changes, exhaust pressure remains unchanged. Thus, the pressure ratio changes and Eq. (4) does not apply. Under these conditions, flow varies with pressure ratio according to Eq. (1). This is shown graphically in Fig. 6.4. However, the last stage of a condensing turbine is designed with nearly critical pressure ratio across its nozzle; therefore, small changes in pressure ratio result in no change in the pressure-flow relationship (where p_2/p_1 is slightly less than critical), or in very small changes (where p_2/p_1 is slightly greater than critical). For all practical purposes, the effect of changes in last-stage pressure ratio can therefore be ignored at design exhaust pressure.

At higher-than-design exhaust pressure, the last-stage flow characteristics become progressively more sensitive to changes in pressure ratio.

6.5 PRESSURE VS. FLOW OVER THE LOAD RANGE

The pressure-flow relationships[2] for constant throttle and reheat temperatures, and constant exhaust pressure, are plotted in Fig. 6.5. The first-stage pressure-flow relationship is not a straight line, because at reduced loads a larger proportion of the total work is accomplished in the first stage, and the temperature

[1] Note that first-stage pressure is second-stage inlet pressure and that the level of first-stage pressure depends upon second-stage performance.

[2] Because the pressure-flow relationship can change depending on turbine condition, first-stage pressure should not be used to calculate flow in lieu of measurement of flow. For any stage, it is important to remember that changes in stage pressure-flow relationships are reasons for concern.

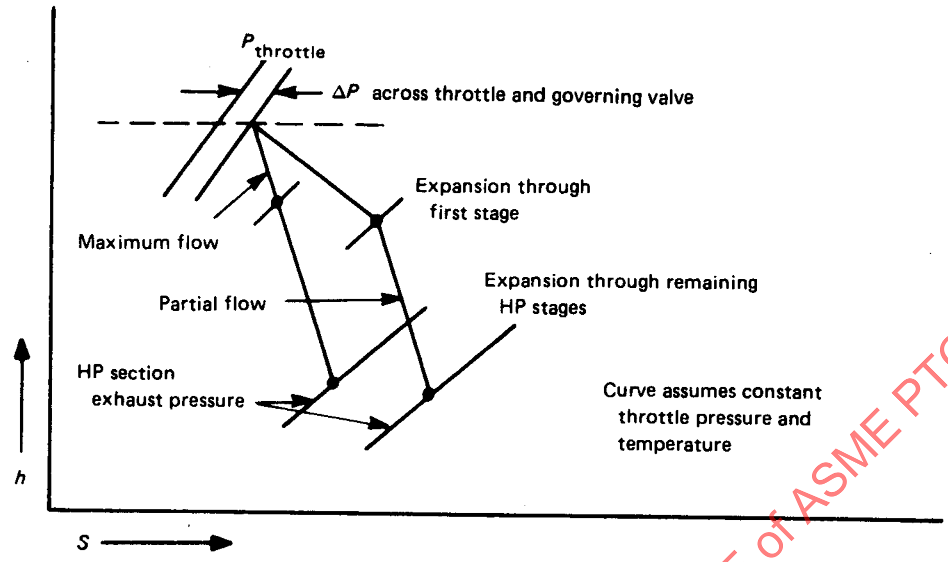


FIG. 6.2 TYPICAL EXPANSION LINES FOR A HIGH-PRESSURE SECTION PARTICAL-ARC ADMISSION, CONDENSING TURBINE

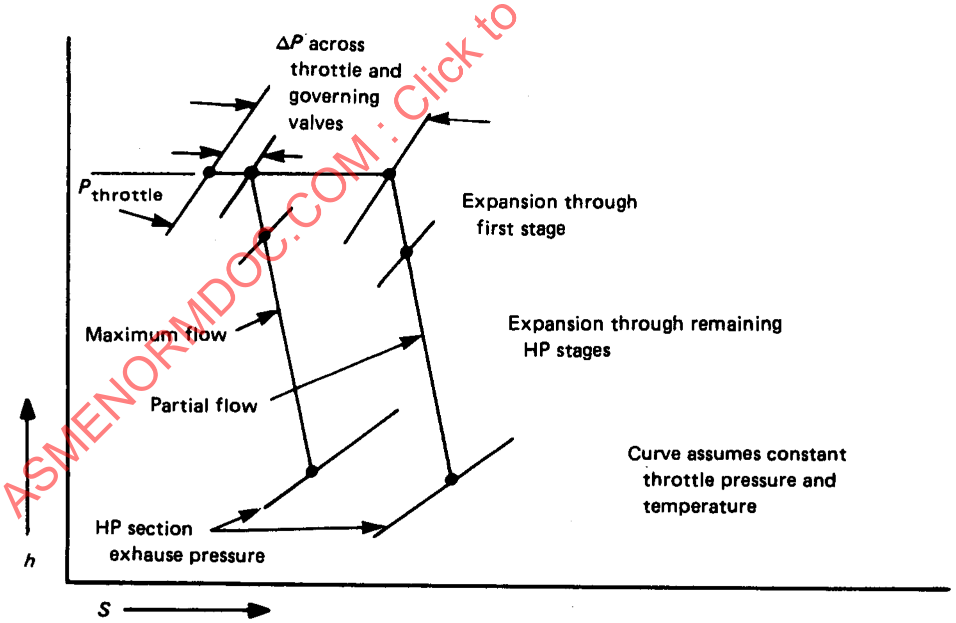
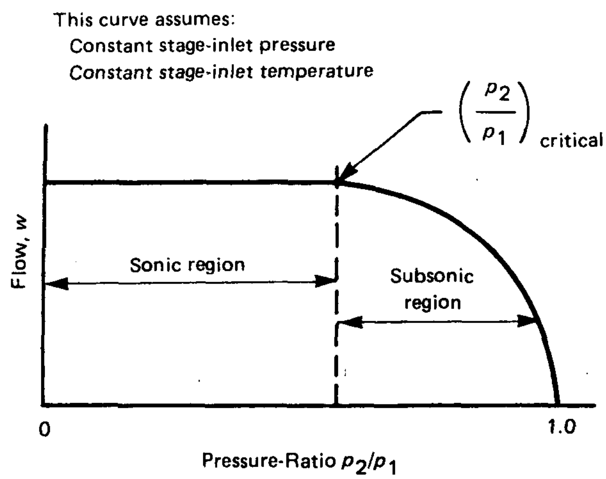


FIG. 6.3 TYPICAL EXPANSION LINES FOR A HIGH-PRESSURE SECTION FULL-ARC ADMISSION, CONDENSING TURBINE



$$\left(\frac{p_2}{p_1}\right)_{\text{critical}} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma - 1}} \qquad (5)^*$$

FIG. 6.4 CONDENSING TURBINE LAST-STAGE STEAM FLOW VS. STAGE PRESSURE RATIO

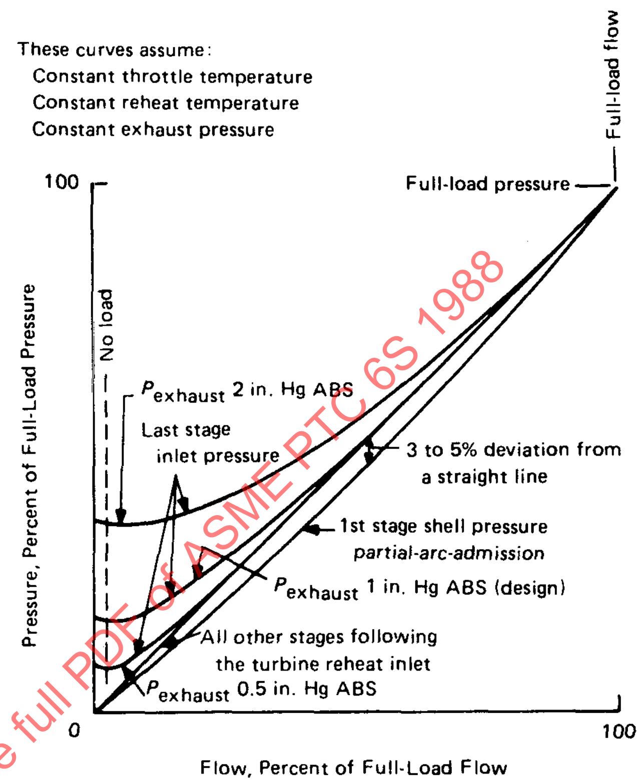


FIG. 6.5 PRESSURE-FLOW RELATIONSHIP

and pressure entering the second stage is lower than at full load. Because of the lower temperature, Eq. (4) shows that less pressure is required to pass this reduced flow than would have been required if the temperature remained constant. This same trend continues for all stages in the high-pressure turbine, with the 3- to 5-percent deviation from a straight line decreasing for succeeding stages in this turbine section. For a full-arc admission turbine, the pressure-flow relationship for the high-pressure section stages is closer to a straight line. For either type of turbine admission, all stages below the reheater discharge have straight-line pressure-flow relationships except the last stage inlet as explained in para. 6.6.

The last-stage inlet pressure at zero flow is equal to the condenser pressure. However, at no-load flow it is lower than condenser pressure due to the pumping action of the last stage. As the last-stage inlet pressure increases, the flow increases at a greater rate, since the last-stage pressure ratio is concurrently decreasing. As the pressure ratio approaches the critical value, the flow-pressure relationship becomes proportional to the stage-inlet pressure. As the exhaust pressure is increased, the deviation from a straight line of the last-stage pressure-flow relationship becomes more pronounced.

6.6 VARIATIONS IN OPERATING CONDITIONS

Equations (1) through (4) may be used to study changes in flow to a stage, stage temperature, and stage pressure when the turbine operating conditions change and governing valve position remains constant.

6.7 VARIATION IN THROTTLE PRESSURE

The general flow Eq. (1) can also be applied to the governing valves. At a constant valve setting, by assuming that pressure ratio across the valves remains constant, Eq. (4) can be used to show that an increase in throttle pressure results in a proportional increase in flow. The increased flow results in a proportional increase in pressure ahead of the first turbine stage. Therefore the pressure ratio across the governing valves remains constant, validating use of Eq. (4). This

*See para. 6.25(b) for the derivation of Eq. (5).

reasoning can be used to show that stage inlet pressure increases in proportion to throttle pressure for all stages except the last stage of a condensing or other constant exhaust pressure turbine. The pressure ahead of the last stage also increases approximately proportionally to increasing throttle pressure when last stage pressure ratio is less than or close to critical. Otherwise an increase in throttle pressure does not result in an equivalent increase in pressure ahead of the last stage.

6.8 VARIATION IN THROTTLE TEMPERATURE

If throttle steam pressure and governing valve position are held constant and the throttle temperature is increased, Eq. (4) shows that the throttle flow decreases. Flow to the following turbine stages likewise decreases by the same amount. There is essentially no change in pressure ratio across the governing valves or the first stage; consequently, there is no significant change in first-stage inlet or exit pressures for this increase in throttle temperature. Similarly, there is no change in first-stage inlet or exit pressures for this increase in throttle temperature. Similarly, there is no change in these pressures for a decrease in throttle temperature. With one minor exception, the same analysis holds true for the remaining stages in the high-pressure section of the turbine because the stage-inlet temperatures will also vary when throttle steam temperature changes.

If reheat steam temperature is held constant (assuming adequate reheater outlet temperature control), the decreased high-pressure turbine steam flow which occurs with an increase in throttle temperature will result in decreased pressure at the high-pressure turbine exit (cold reheat) and at the inlet to the reheat turbine section (hot reheat). Accordingly, the total pressure drop across the high-pressure turbine is increased, and the pressure ratios across the last few stages will vary slightly. The effects are relatively minor for the changes in throttle temperature usually encountered.

6.9 VARIATION IN REHEAT TEMPERATURE

With throttle steam conditions and governing valve position held constant, flow will remain constant with variations in reheat temperature. Applying Eq. (4) to the first reheat stage, an increase in reheat temperature results in an increase in reheat pressure approximately proportional to the square root of the ratio of the absolute temperatures before and after the increase. Actually, it is more accurate to use changes in the specific volume to calculate this change in pressure, Eq. (3). This increase in pressure is reflected in all stages downstream of the reheater discharge, and also causes a decrease in total pressure drop across the high-pressure turbine. Correspondingly, a decrease in reheat temperature has a reverse effect.

6.10 VARIATION IN AREA OF THE FIRST-STAGE NOZZLE

If the first-stage nozzle area increases due to solid-particle or other erosion, throttle flow will increase for the same steam inlet conditions and governing valve opening. In the case of low loads on a partial-arc admission turbine, the pressure drop across the first-stage nozzle is greater than critical and steam flow is directly proportional to the nozzle area. However, as flow increases, the pressure ahead of the second stage increases, causing the pressure drop to become less than critical. With the governing valves wide open, a 1.0 percent change in nozzle area allows only a small fraction of 1.0 percent change in flow, as illustrated in Fig. 6.6.

6.11 VARIATIONS IN STAGE-EXIT PRESSURE

Variations in stage exit pressure, such as those caused by changing reheat conditions or abnormal extraction flows, cause the stage pressure ratio to change, and Eqs. (3) and (4) do not apply.

Figure 6.7 shows a variation in stage-inlet pressures as related to changes in stage exit pressures for all stages between the first stage and the last stage. For these intermediate stages, the percent pressure drop across an individual stage is relatively low and permits the application of incompressible-flow theory. For incompressible flow γ is very large. Thus:

$$\left(\frac{\gamma+1}{\gamma}\right) \sim 1 \text{ and } \left(\frac{2}{\gamma}\right) \sim 0$$

Accordingly, the flow Eq. (1) simplifies as follows:

$$w = 3600\, C_q\, A_n \sqrt{\frac{2g(p_1 - p_2)}{v_1}} \qquad (6)^*$$

or, using the equation of state $p_1 v_1 = RT_1$, then:

$$w = 3600\, C_q\, A_n \sqrt{2gp_1 \frac{(p_1 - p_2)}{RT_1}} \qquad (7)^*$$

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*See para. 6.25(c) for derivation of Eqs. (6) and (7).

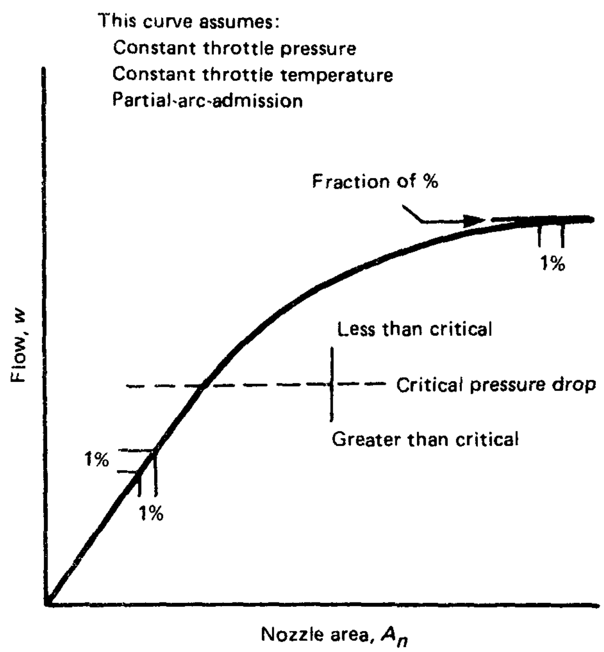


FIG. 6.6 STEAM FLOW VS. FIRST-STAGE NOZZLE AREA

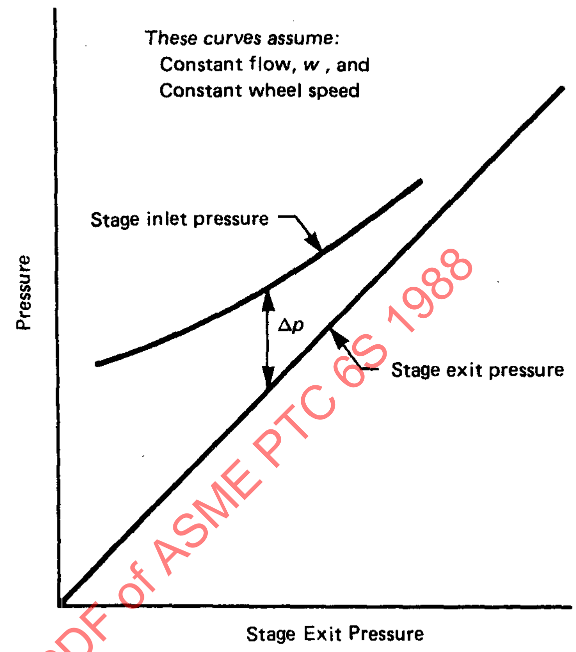


FIG 6.7 STAGE PRESSURE VS. STAGE EXIT PRESSURE FOR INTERMEDIATE STAGES

When stage flow remains constant as stage exit pressure increases, it is important to note that the inlet pressure also increases, but not as rapidly as does the exit pressure.

6.11.1 For a pure impulse stage, its downstream pressure is the same as the pressure between the nozzle and rotating blade row, or $p_2 = p_3$. The change in upstream pressure as downstream pressure changes can be calculated from Eq. (7) by assuming the flow remains constant as follows:

$$C_q A_n \sqrt{\frac{2gp_1\ (p_1 - p_3)}{RT_1}}$$

$$= C_q A_n \sqrt{\frac{2gp_1'\ (p_1' - p_3')}{RT_1'}} \tag{8}$$

where primed figures symbolize conditions after the downstream pressure changes, Eq. (8) simplifies to:

$$p_1\ (p_1 - p_3) = p_1'\ (p_1' - p_3') \tag{9}$$

For a 10 percent increase in downstream pressure:

$$p_3' = 1.1\ p_3 \tag{10}$$

and

$$p_1\ (p_1 - p_3) = p_1'\ (p_1' - 1.1\ p_3), \text{ or:} \tag{11}$$

$$1 - \frac{p_3}{p_1} = \frac{p_1'}{p_1}\left(\frac{p_1'}{p_1} - \frac{1.1\ p_3}{p_1}\right) \tag{12}$$

Assuming the stage pressure ratio, p_3/p_1, is 0.8, then

$$1 - 0.8 = \left(\frac{p_1'}{p_1}\right)^2 - \frac{1.1 \times 0.8\ p_1'}{p_1} \tag{13}$$

or:

$$\left(\frac{p_1'}{p_1}\right)^2 - 0.88\ \frac{p_1'}{p_1} - 0.2 = 0 \tag{14}$$

The positive root of the quadratic equation is:

$$\frac{p_1'}{p_1} = \frac{+0.88 \pm \sqrt{0.88^2 + 0.8}}{2} = 1.067 \tag{15}$$

Thus for a 10 percent change in p_3, p_1 changes only 6.7 percent.

This change in p_1, which is the exit pressure of the preceding stage, creates a similar change in the preceding stage inlet pressure. In this manner, the change in the exit pressure of one stage affects the pressure ratios of all upstream stages, but the effect becomes progressively smaller for stages further upstream.

As an example, for an eight-stage high-pressure section of an impulse turbine, the change in exit pressure of each stage for a 10 percent increase in high-pressure-section exhaust pressure is:

Stage No.	Increase in Exit Pressure, %
8	10.0
7	6.7
6	4.5
5	3.0
4	2.0
3	1.3
2	0.9
1	0.6

Thus, in this instance, the pressure after the first stage changes by only 0.6 percent, which results in a very small change in flow. This verifies the assumption of constant flow through a high-pressure turbine section as the exhaust pressure of this section increases.

A similar derivation for a typical reaction turbine shows approximately the same results.

6.12 USE OF TEMPERATURE, PRESSURE, AND FLOW RELATIONSHIPS

Plots of $w/\sqrt{p_1/v_1}$ Eq. (3), versus flow to the following stage for the first stage of the machine and for all extraction points, as well as for the inlet to the reheat section, are very useful in evaluating the consistency and accuracy of a test. These plots are also very helpful in determining errors in individual test points and any deviation from a horizontal straight line is indicative of such errors.

For units using wet steam, the expression $w/\sqrt{p/v}$ will decrease as the moisture content increases because of the drag effect of the moisture drops on the steam. Experience has shown that this phenomenon can be compensated for by dividing the expression by the square root of one minus the moisture fraction, as follows:

$$\frac{w/\sqrt{p/v}}{\sqrt{1-M}}$$

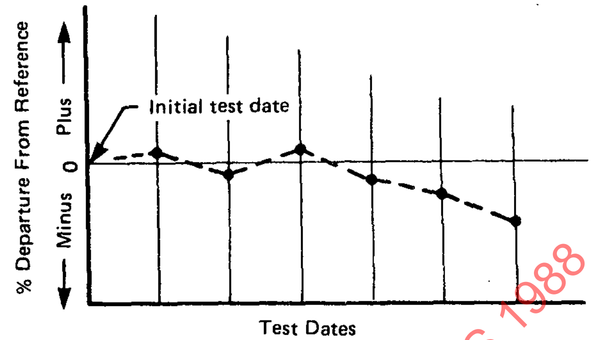


FIG. 6.8 PRESSURE OR CAPABILITY CURVE VS. CHRONOLOGICAL TEST DATES

Any change in the mechanical condition of a steam turbine which changes the flow from a given stage to subsequent stages, will change the pressure-flow relationship of the given stage. For interpretation of test deviations see paras. 6.24 through 6.24.9 which list typical reasons for deviations of steam turbine performance from reference, or design level.

6.13 CONSTRUCTION OF A CHRONOLOGICAL PRESSURE CURVE

Chronological plots of selected turbine stage or shell absolute pressures are also useful. These curves are based on the relationship that steam flow is proportional to the absolute pressure ahead of a stage, a group of stages, or a section of the turbine, except for small changes in specific volume due to changes in absolute temperature which may occur at the corresponding location. This can be seen by reference to Eqs. (2) and (4), using the assumption that v_1 and T_1 do not change significantly with changes in flow. It is important that all data to be plotted chronologically be taken at the same governing valve position, preferably with valves wide open, and with the same cycle arrangement. Cycle variations should be recognized as disturbing influences that could invalidate comparisons. However, small variations in feedwater heater performance have minimal effect on the flow-pressure relationship.

6.13.1 Figure 6.8 exemplifies a chronological pressure curve. The ordinate of the curve utilizes various selected steam pressures, psia, representing turbine stage or shell pressures. The stage or shell pressures observed during a test require correction, as described below, for departures from reference steam conditions.

6.13.2 The first-stage pressure observed during a test on the high-pressure section of a reheat turbine, or the stage or shell pressure for any stage of a non-reheat turbine, should be corrected to reference conditions by the following equation:

$$p_c = p_o \times \frac{p_d}{p_t} \qquad (16)$$

where

p_c = corrected pressure for plotting, psia
p_o = test stage or shell pressure, psia
p_t = test throttle pressure, psia
p_d = design, or reference, throttle pressure, psia

For a feedwater heating extraction from an intermediate stage in the high-pressure turbine section, the measured stage or shell pressure should also be corrected by use of this same equation. Although theoretically incorrect, Eq. (16) is a very close approximation.

6.13.3 For stage or shell test pressures at, or following, the inlet to the reheat section of a turbine, and for the exit from the last stage of the high-pressure section, additional corrections must be made for variations in throttle temperature and reheat temperature, and the correction equation becomes:

$$p_c = p_o \begin{pmatrix} \text{throttle pressure} \\ \text{and temperature} \\ \text{correction} \end{pmatrix} \times \begin{pmatrix} \text{reheat} \\ \text{temperature} \\ \text{correction} \end{pmatrix}$$

or

$$p_c = p_o \sqrt{\frac{p_d}{p_t} \times \frac{v_t}{v_d}} \times \sqrt{\frac{v_{dr}}{v_{tr}}} \qquad (17)^*$$

where

v_d = design or reference throttle specific volume, ft³/lbm
v_t = test throttle specific volume, ft³/lbm
v_{tr} = specific volume at test temperature and test pressure at inlet to reheat stop valves, ft³/lbm
v_{dr} = specific volume at design reheat temperature and test pressure at inlet to reheat stop valves, ft³/lbm

6.13.4 Typical reasons for deviations in stage pressures from reference or design level are given in Fig. 6.9 and paras. 6.24 through 6.24.9. The reasons may be any one of the possibilities or a combination of them in addition to a possible change in steam flow.

*See para. 6.25(d) for derivation of Eq. (17).

6.14 CONSTRUCTION OF A STAGE PRESSURE VERSUS FLOW CURVE

Figure 6.10 is an example of a stage pressure versus flow curve. The construction is based on the following:

Ordinate: Various selected steam pressures, psia, representing turbine stage or shell pressures, such as first-stage pressure, reheat turbine bowl (or reheat inlet) pressure, and crossover pressure. Stage or shell pressures observed during a test should be corrected as outlined in paras. 6.13.2 and 6.13.3.

Abscissa: Throttle steam flow to the turbine, corrected to reference, or design, conditions as follows:

$$w_c = w_t \sqrt{\frac{p_d}{p_t} \times \frac{v_t}{v_d}}$$

where

w_c = corrected throttle flow, lbm/h
w_t = test throttle flow, lbm/h

The other terms of this equation are defined in paras. 6.13.2 and 6.13.3.

6.15 TURBINE SECTION EFFICIENCIES

In this report, the term *section efficiency* refers to the overall efficiency of a turbine section from inlet to exhaust, and is calculated as shown in Fig. 5.3. The section efficiency thus accounts for entrance losses and stage efficiencies.

6.16 STAGE EFFICIENCY

Understanding the performance of a single stage is an excellent basis for analyzing the entire turbine. Figure 6.1 shows a single-stage steam path. The pressure upstream of the nozzle or stationary row is designated as p_1, the pressure between the nozzle and the rotating row as p_2, and the pressure downstream of the rotating row as p_3. As the flow passes through the nozzle, its velocity is increased and its direction is changed. The flow passing through the rotating row is changed in direction. The amount of change in relative velocity with respect to the rotating row is dependent upon the design of the stage. The relationship between p_2 and p_3 determines the amount of stage reaction; that is, if p_2 is equal to p_3, the stage is an impulse design. If p_2 is

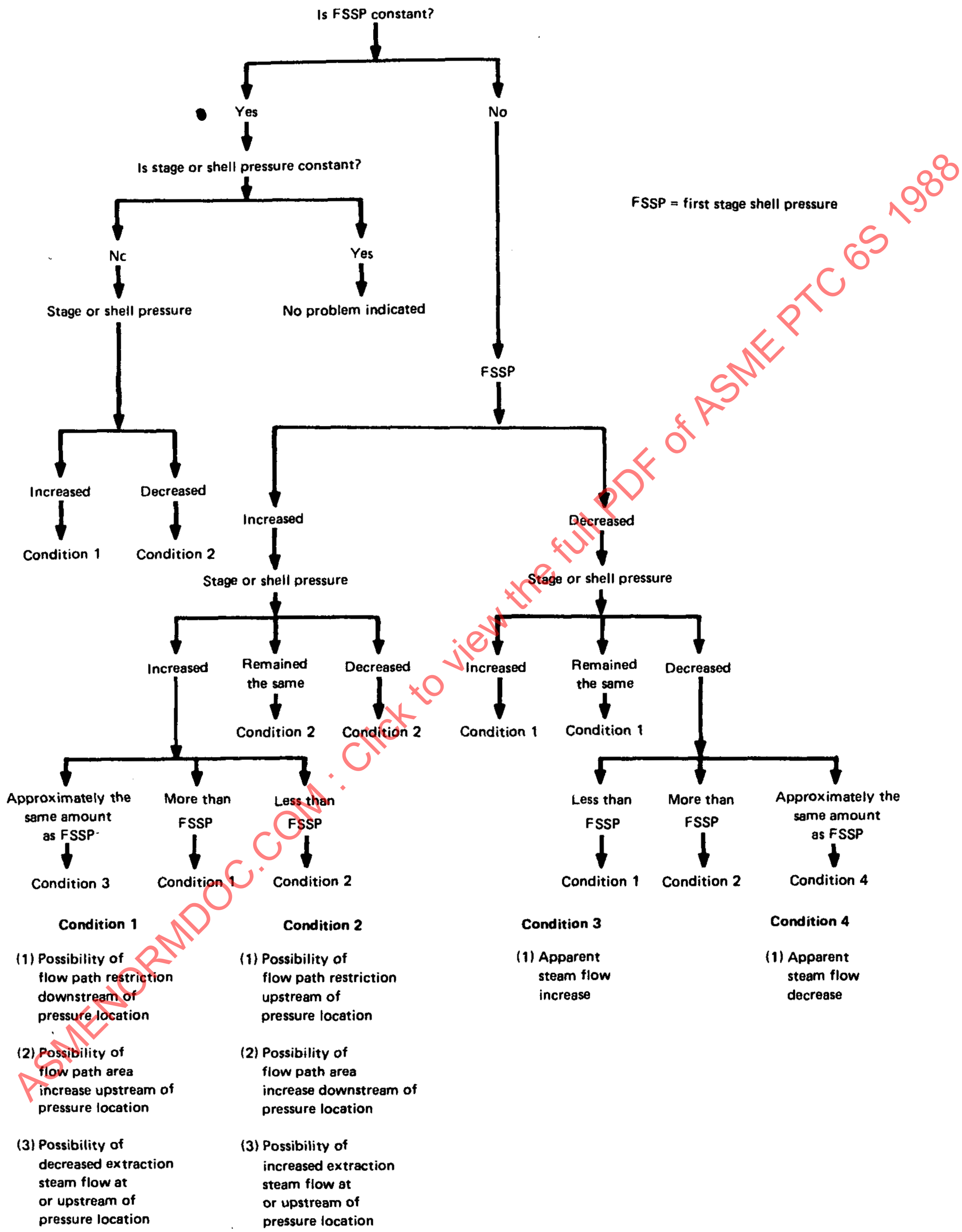


FIG. 6.9 CORRECTED PRESSURE DEVIATION INTERPRETATIONS AT CONSTANT CONTROL VALVE OPENING (REFER TO PARAS. 6.13.4 AND 6.24 THROUGH 6.24.9)

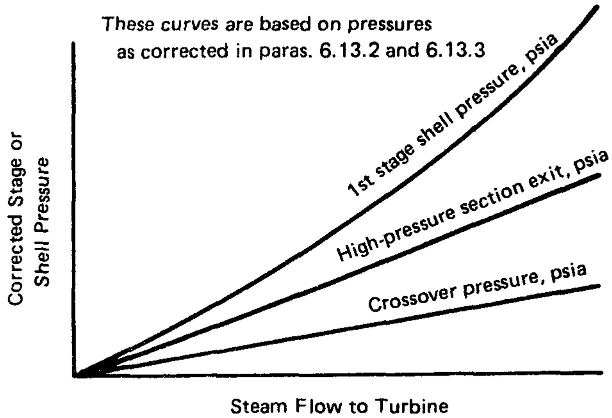


FIG. 6.10 STAGE PRESSURE VERSUS THROTTLE STEAM FLOW

higher than p_3, then the stage is a reaction design. The percent reaction depends on the amount of pressure drop taken across the rotating row.

6.16.1 In Fig. 6.1 the velocity of steam discharging from the nozzle is indicated as V_1. In order to determine the direction and magnitude of the entering steam velocity with respect to the rotating row, it is necessary to subtract vectorially from V_1 a vector V_w which represents the wheel speed of the rotating row. The steam velocity entering the rotating row is indicated as V_2. Figure 6.1 shows that the angle with respect to the axial direction is smaller for V_2 than for V_1. Since the kinetic energy varies as the square of the velocity, the kinetic energy of the steam entering the rotating row is smaller than that leaving the nozzle. For an impulse design, the entering rotating-row kinetic energy is approximately one-quarter of the nozzle-exit energy. For a reaction design, the entering rotating-row kinetic energy is a larger fraction of the nozzle-exit energy, which in turn is smaller than for an impulse design because the pressure drop across the stage divides evenly between the stationary and rotating rows. The discharge velocity from the rotating row is represented by V_3. To find the absolute velocity leaving the stage, again the wheel speed is subtracted vectorially to give V_4. The design of the stage is such that V_4 is kept as axial as possible, in order to keep the kinetic energy in the leaving steam to a minimum, thus maximizing the absorption of energy by the rotating row.

6.16.2 It is well to consider the characteristic of a single stage as the wheel speed varies. Although seemingly a peculiar approach to the operation of a constant-speed turbine, it lends understanding to the changes in turbine characteristics. Similar reasoning can be applied when other parameters are changed. If the wheel speed is decreased, V_2 will increase and will enter the rotating row at an angle different from the design angle, causing an increase in loss as the flow passes through the rotating row. Also, when wheel speed decreases, V_4 increases and results in an increase in leaving loss. If the wheel speed is increased, the angle at which the flow enters the rotating row will change and will also increase leaving loss. The efficiency of a single stage plotted against wheel speed is shown in Fig. 6.11. The curve is fairly flat near the design point; that is, for a small change in speed, there is a relatively small change in efficiency.

6.17 HIGH-PRESSURE SECTION EFFICIENCY

Efficiency of a partial-arc admission high-pressure turbine section varies as a function of flow as shown in Fig. 6.12.

The efficiency between valve points (valve loops) is poorer than the efficiency of the locus of valve points primarily because of the throttling loss through one or more governing valves that restrict the flow. The valve loops get progressively larger with more impact on efficiency as load is reduced because a larger percentage of the remaining flow is throttled. The first stage is the only stage where the nozzle area can be varied. When the turbine is operating with valves wide open, the pressure ratio across the first stage is about 0.8 and the flow is maximum, usually referred to as the maximum expected flow. When a valve is closed, the area through which flow can pass into the first stage is decreased and, therefore, the total flow will decrease. Since the flow is decreased, the pressure ratio across the first stage decreases, which increases the theoretical steam velocity (V_o) and, therefore, the ratio of wheel velocity to steam velocity decreases. This results in the flow entering the rotating row at an angle different from the design angle and causes an increase in leaving loss, with attendant decrease in efficiency.

6.17.1 The performance of the other stages in the high-pressure section of the turbine does not change very much as the governing valves are closed sequentially, because the inlet pressure to the second stage decreases with the decreased flow, as does also the inlet pressure to the third stage, resulting in nearly constant pressure-ratio across the second stage. This same reasoning applies to the other stages in the high-pressure section, and therefore, their efficiency remains substantially constant.

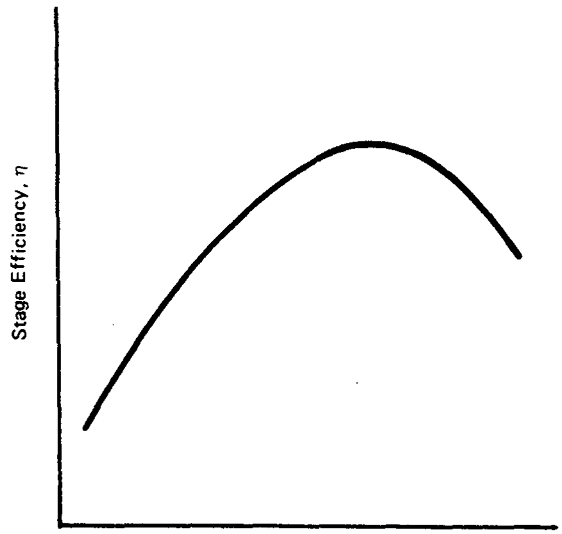


FIG. 6.11 SINGLE-STAGE EFFICIENCY VERSUS WHEEL SPEED

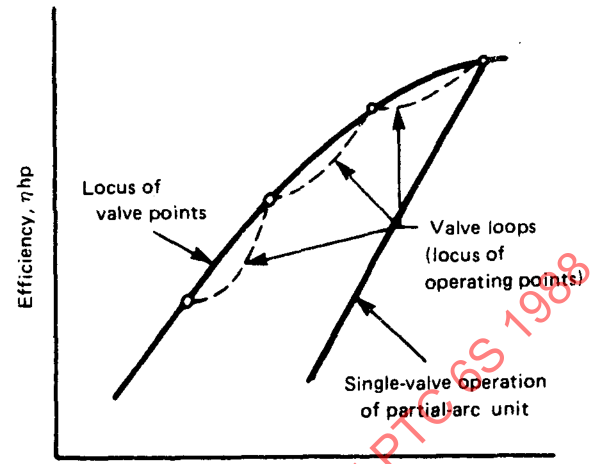


FIG. 6.12 PARTIAL-ARC ADMISSION UNIT (HIGH-PRESSURE TURBINE SECTION EFFICIENCY VS. FLOW OR PRESSURE PARAMETER)

6.17.2 At a constant throttle pressure, efficiency of a full-arc admission high-pressure turbine section varies as a function of flow as shown in Fig. 6.13. The pressure ratio across each stage remains constant as flow varies, and stage efficiencies remain nearly constant. As the governing valves close to reduce flow, throttling loss across the valves increases. This loss accounts for the nearly linear decrease in efficiency as flow is reduced. Full-arc admission turbines are sometimes operated over the load range with valves wide open, and flow control is achieved by varying steam generator pressure. In this variable pressure mode, the throttling loss is eliminated, and high-pressure turbine section efficiency remains nearly constant as does pressure ratio.

6.18 INTERMEDIATE-PRESSURE SECTION EFFICIENCY

The efficiency of stages in the intermediate-pressure (reheat) section does not change with changing flow since these stages also operate at a constant pressure ratio and thus at a constant ratio of wheel velocity to steam velocity. See paras. 5.6 and 6.24.7.

6.19 LOW-PRESSURE SECTION EFFICIENCY

Figure 6.14 shows the variation of low-pressure turbine efficiency versus exhaust steam volumetric flow or steam velocity. Variations in exhaust steam velocity and volumetric flow may be due to variations in either mass flow or condenser pressure, and this curve applies to both cases. Again, variations in leaving loss and losses associated with improper rotating-row entrance angle account for the variations in efficiency.

6.19.1 For low-pressure turbines operating in nuclear cycles where extra water removal from the steam path is significant, low -pressure efficiency is not an appropriate indicator of performance. More effective water removal leads to a lower indicated overall efficiency, which is a contradiction. Therefore, it is recommended that low pressure turbine performance for this case be described in terms of effectiveness[3], ϵ, where

$$\epsilon = \frac{\Delta h}{\Delta h + T_o \Delta S} \qquad (19)$$

Δh = sum of actual Btu work for all of the individual expansions in the low pressure steam path.
ΔS = sum of entropy changes for all of the Δh expansions used above.
T_o = absolute temperature, °R, of saturation at the low pressure turbine exhaust pressure.

[3]The concept of the effectiveness was reported by J. H. Keenan in "A Steam Chart for Second Law Analysis," *Mechanical Engineering*, Vol. 54, 1932, pp. 195–204, and is referred to in some textbooks on thermodynamics.

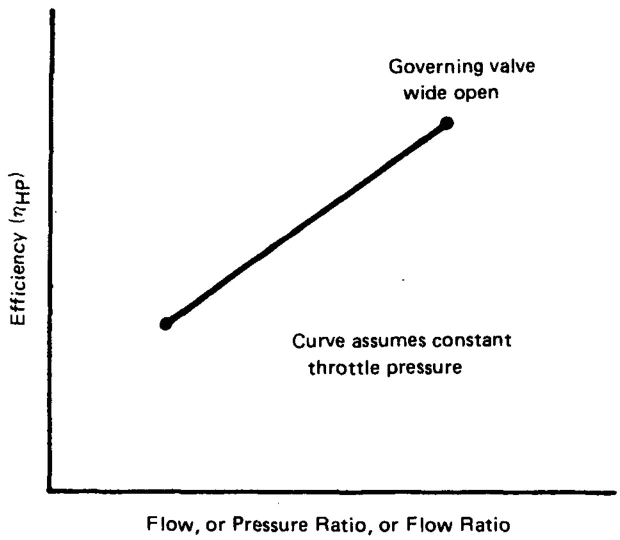


FIG. 6.13 FULL-ARC ADMISSION UNIT (HIGH-PRESSURE TURBINE SECTION EFFICIENCY VS. FLOW OR PRESSURE PARAMETER)

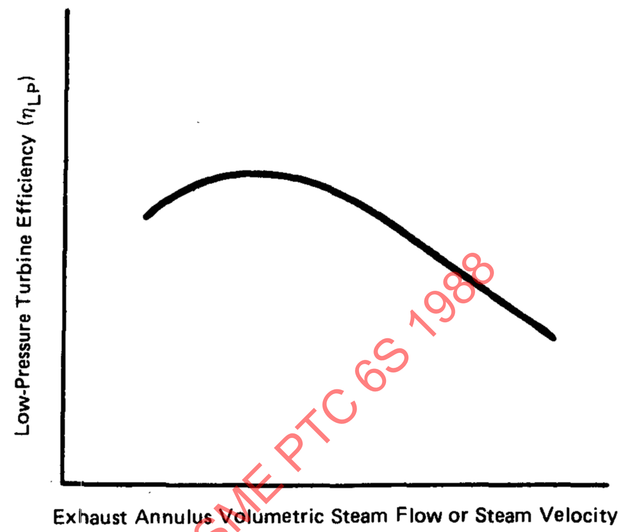


FIG. 6.14 LOW PRESSURE TURBINE SECTION EFFICIENCY VERSUS EXHAUST STEAM FLOW OR VELOCITY

The mechanics of the definition of effectiveness are illustrated in Fig. 6.15. Effectiveness should be plotted versus exhaust volumetric flow or exhaust annulus velocity for comparison with expected values or with other performance.

6.20 USE OF ENTHALPY-DROP EFFICIENCY CURVES

Turbine-section enthalpy-drop efficiencies based on periodic tests may be chronologically plotted. The following suggestions are recommended in the construction of such plots:

(*a*) For a high-pressure turbine section which includes governing valves operated to regulate flow, the best pressure ratio index of high-pressure section efficiency is first-stage pressure over throttle pressure. Alternately, section efficiency may be plotted against the ratio of section exhaust pressure to throttle pressure. Tests for chronological plots must always be conducted at the same governing valve opening for these turbine sections to avoid the variation shown in Figs. 6.12 and 6.13.

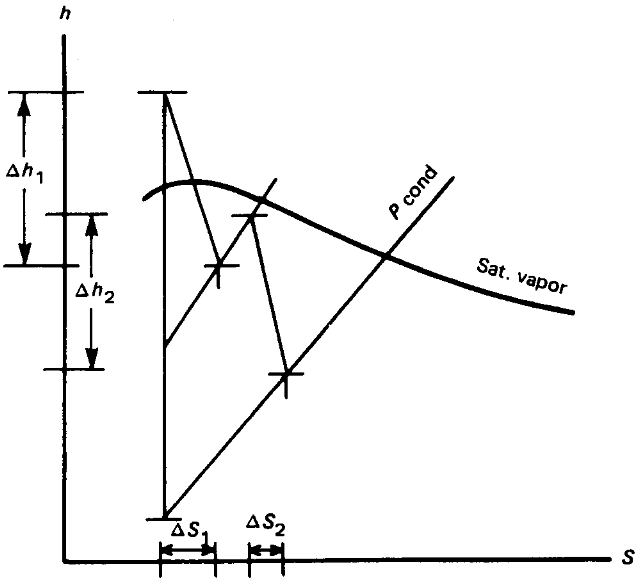
(*b*) For turbine sections which do not contain governing valves used to control flow, such as the intermediate-pressure section which directly follows the reheat inlet, or a high-pressure turbine section operated with variable throttle pressure, turbine section efficiency is relatively constant over the load range and the requirements in (a) above do not apply.

6.20.1 Enthalpy-drop efficiencies may be plotted throughout the load range of a turbine. Such a plot, based on a series of tests, will assist in the turbine performance analysis. These efficiencies may be plotted as a function of pressure ratio, certain stage pressures, or flow ratio, each of which is discussed in the construction of the abscissa of this curve. A typical plot of partial-arc admission high-pressure-turbine section efficiency is shown in Fig. 6.12 and a plot of full-arc admission high-pressure turbine section efficiency at a constant throttle pressure is shown in Fig. 6.13. Construction is based on the following:

Ordinate: Enthalpy-drop efficiency. For calculations of enthalpy-drop efficiencies, see paras. 5.3 and 6.15.

Abscissa: Absolute pressure is used for all comparisons.

(*a*) The pressure ratio, $\frac{p_{\text{first stage}}}{p_{\text{throttle}}}$ is a ratio of the high-pressure turbine first-stage shell pressure, to the throttle pressure. This ratio is an excellent basis for comparing tests.



$$\epsilon_{LP} = \frac{\sum_{i=1}^{2} \Delta h_i}{\sum_{i=1}^{2} (\Delta h_i + T_o \Delta S)}$$

FIG. 6.15 ILLUSTRATION OF LOW PRESSURE TURBINE EFFECTIVENESS

(*b*) The pressure ratio, $\frac{p_{\text{exit}}}{p_{\text{throttle}}}$ is a ratio of the high-pressure-turbine-section exhaust pressure to the throttle pressure. At full load, changes in main and reheat steam temperature do not significantly affect the high-pressure-turbine efficiency as it relates to high-pressure-ratio. At lower loads, temperature changes have a greater effect but not an appreciable one.

(*c*) High-pressure-turbine first-stage shell pressure, corrected to standard throttle conditions.

(*d*) Reheat-bowl pressure corrected to standard throttle and reheat conditions. It can serve as the abscissa for the curve showing efficiency of the intermediate-pressure section which follows the reheat inlet.

(*e*) A ratio of test flow, corrected to reference or design conditions, divided by the valves-wide-open steam flow, if available. It may also serve as the abscissa. The test flow must be corrected for departures from reference or design conditions by the following multiplying factor:

$$\text{Multiplying Factor} = \sqrt{\frac{p_d}{p_t} \times \frac{v_t}{v_d}} \qquad (20)$$

where

p_d = design throttle pressure, psia
p_t = test throttle pressure, psia
v_t = test throttle specific volume, ft³/lbm
v_d = design throttle specific volume, ft³/lbm

6.20.2 Turbine section efficiencies may deviate from normal reference data due to the following.

(*a*) A change in the slope of the test high-pressure efficiency curve from the reference, due to a greater decline in turbine section efficiency at lower loads than at full load, is indicative of a decline in the performance of the governing stage in the turbine.

(*b*) An overall efficiency decline in the turbine section tested may result from extensive packing damage, deposits on blading, and damage to the blading.

For further interpretation refer to paras. 6.24 through 6.24.9.

6.21 TURBINE CAPABILITY TESTS

Turbine capability test results plotted chronologically are of considerable assistance in detecting a decline in turbine performance. Often such curves are of important historical value, not only in turbine performance diagnostics, but also in correlating turbine performance with other plant operating experiences. Tests during initial operation of a turbine serve as a basis for future performance evaluation. Plots of tests immediately before and after turbine inspection and repair are valuable indices which reflect performance changes.

6.21.1 A typical chronological plot for a series of capability tests is shown by Fig. 6.8. Curve construction is based on the following:

Ordinate: Percentage change in turbine output as related to reference output. Measured output must be corrected for any deviation in initial, reheat, and exhaust steam conditions from the reference condition. Measured output must also be corrected for any other conditions which influence output as outlined in appropriate sections of this Report.

Abscissa: Usually a calendar scale using the test date as point of entry. The number of observations, as well as the time interval between tests, depends upon the judgment of personnel involved.

6.21.2 If, during an internal inspection, eroded first-stage nozzles are replaced or repaired such as to reduce the nozzle area, there may be a reduction in flow capability that may lower the output below that previously achieved. See para. 6.24.1(c). For further interpretation of the significance of decreasing turbine capability, refer to paras. 6.24 through 6.24.9.

6.22 SEALING AND LEAK-OFF FLOWS AND TEMPERATURES

A chronological plot of sealing and leak-off flows and temperatures is most helpful in determining the physical condition of associated seals, packings, balancing pistons, control-valve-stem bushings, and similar items. In an abbreviated test, seal and leak-off flows can readily be determined only if they are external to the turbine casing and a flow element can be installed. Usually, such flows constitute a small percentage of the energy input to the turbine and must increase considerably before any significant decline in turbine performance is experienced. Where seal and leak-off flows discharge into extraction steam lines, particularly those conveying superheated steam, the temperature of the steam downstream of such junction is a helpful index of leakage, and these leakage flows often can be determined by heat balance around the junctions. For the significance of changes in flow or temperature, refer to paras. 6.24 through 6.24.9.

6.23 PERIODIC TURBINE HEAT-RATE TESTS

Turbine heat rate tests at selected governing-valve positions may be chronologically plotted to trend turbine performance. The numerical value of the heat rate, in Btu per hphr or per kWhr, or its deviations from its reference value will serve equally well. Any heat rate values must be corrected for deviations from reference throttle steam conditions, reheat steam conditions, exhaust pressure and other cycle characteristics. Appropriate sections of this Report should therefore be consulted as required.

6.23.1 Turbine steam rate data, in lbm of steam per hph or per kWh, may be advantageously presented in curve form. Such data may be plotted chronologically for given governing and extraction valve positions. For varying turbine outputs, steam rate data may be plotted as a function of output, steam flow, or other selected variables. Steam rate data must be corrected to reference or design conditions, using appropriate curves furnished by the turbine manufacturer or established by acceptable tests. Construction and interpretations of steam rate curves should follow general recommendations presented for such curves in this Report.

6.24 TYPICAL REASONS FOR DEVIATIONS OF STEAM TURBINE PERFORMANCE FROM REFERENCE OR DESIGN LEVEL

Before a turbine is inspected for internal damage or deposits, all conditions external to the turbine should be carefully checked to assure that departures from reference performance are not due to external causes, such as cycle irregularities including leaky valves or other isolation problems, inaccurate instruments, observations, and/or calculations.

6.24.1 A change in mechanical conditions which changes the turbine steam-inlet areas may occur, such as:

(*a*) first-stage blading damage resulting in partial closing of the steam-flow area, thus reducing flow through the turbine;

(*b*) impairment or maladjustment of control valve(s), or governing system;

(*1*) a valve may fail to operate because of a broken component;

(*2*) valves may not operate in proper sequence, or with correct lift, due to improper reassembly or adjustment of control gear after inspection. Either could result in restricted steam admission or in excessive throttling throughout the load range.

(*c*) erosion of first-stage nozzles, thus increasing steam admission area, steam flow, and output.

6.24.2 Thermodynamic deformation of the cycle due to changes in extraction steam demands, or due to any change in steam or water flow to, or from, the turbine cycle causes a departure in performance level that appears to remain at a constant percentage difference from reference level.

6.24.3 Mechanical damage to internal seals, resulting in an increase in steam leakage, may occur suddenly during abnormal operation, or over a relatively long period of time. Stage steam pressures usually decrease due to this type of damage. Damage associated with an accident is reflected by a sudden departure of performance level from its reference value. Damage to leakage-control clearances is a more dominant factor in the high-pressure section of a turbine than in the low-pressure section and thus low-pressure spindle rubs have less effect on performance than rubs in other sections of the turbine.

6.24.4 Blading deposits which reduce steam passage areas may occur and often result in an increase in shell or stage steam pressures, accompanied by a gradual performance decline. Washing techniques should be discussed with the turbine manufacturer. Data carefully taken immediately before and after washing should reflect effectiveness of washing procedures.

The amount of deposit which can reduce turbine section efficiency from 1 to 3 percentage points ordinarily causes an increase in stage pressures of from 0.25 percent to 0.75 percent. Care must be exercised in making these pressure measurements.

6.24.5 Increase of leak-off steam flows often results in steam with high available energy being bypassed to lower sections of the turbine. Some high-pressure-seal and leak-off flows may bypass the reheater and consequently fail to gain the thermodynamic advantage of reheat. This usually shows up as a decrease in turbine output at any given governing valve setting.

6.24.6 Overload valve leakage on turbines equipped with these valves, permits steam with high available energy to bypass certain turbine stages resulting in an increase in both steam rates and heat rates.

6.24.7 Any change in performance of the intermediate-pressure turbine section due to deposits or internal damage, will decrease the efficiency of this section by the same amount at light loads as at full load, and will decrease the overall capability of the turbine unit. For the overall heat-rate effect of a decrease in intermediate-pressure turbine efficiency, see Section 9, para. 9.8(c). On some units, an increase in the internal leakage from the high-pressure section to the intermediate-pressure section can result in misleading improvements of intermediate-pressure turbine efficiencies; whereas the true efficiency of the intermediate-pressure turbine section may not have changed.

6.24.8 Damage to, or loss of, blading in the low-pressure section of the turbine usually cannot be detected by enthalpy-drop methods. Depending upon the extent of damage, this may appear as an offset in stage pressures, capability, heat rate, and overall plant performance.

6.24.9 Governing valve-stem clearances may decrease due to deposits on stems, thus reducing expected leak-off flows.

6.25 DERIVATION OF EQUATIONS

(a) Derivation of Eq. (1):

The general energy equation for steady flow in a nozzle is:

$$\frac{V_1^2}{2gJ} + h_1 = \frac{V_2^2}{2gJ} + h_2 \tag{1a}$$

In the case of $V_1 << V_2$ or V_1 is negligibly small, $\frac{V_1}{2gJ} \approx 0$:

$$V_2^2 = 2gJ\,(h_1 - h_2) \tag{1b}$$

For a frictionless nozzle,

$$h_1 - h_2 = -\int_1^2 vdp = C_p\,(T_1 - T_2) \tag{1c}$$

From the equation of state of an ideal gas,

$$pv = RT \tag{1d}$$

and for an isentropic process,

$$pv^{\gamma} = \text{constant} \tag{1e}$$

Combining Eqs. (1d) and (1e),

$$Tv^{\gamma-1} = \frac{T}{p^{\left(\frac{\gamma-1}{\gamma}\right)}} = \text{constant}$$

Since,

$$Cp = \frac{\gamma}{\gamma-1}R$$

equation (1b) becomes,

$$V_2^2 = 2gJ\,(h_1 - h_2) = \frac{2g\gamma}{\gamma-1}RT_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{1f}$$

or,

$$V_2 = \sqrt{2gRT_1 \frac{\gamma}{\gamma-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]} \qquad (1g)$$

From equation of continuity:

$$w = \frac{VA_n}{v_2} \times 3600 \qquad (1h)$$

and since,

$$v_2 = v_1 \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}} = \frac{RT_1}{p_1}\left(\frac{p_1}{p_1}\right)^{\frac{1}{\gamma}} \qquad (1i)$$

Mass flow rate is written,

$$w = 3600\, A_n \frac{p_1}{RT_1}\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} V_2$$

$$= 3600\, A_n P_1 \sqrt{\frac{2g\gamma}{\gamma-1}\frac{1}{RT_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma+1}{\gamma}}\right]}$$

$$= 3600\, A_n p_1 \sqrt{\frac{2g\gamma}{\gamma-1}\frac{1}{p_1 v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma+1}{\gamma}}\right]}$$

$$= 3600\, A_n \sqrt{\frac{2g\gamma}{\gamma-1}\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma+1}{\gamma}}\right]} \qquad (1j)$$

Equation (1j) represents the insentropic flow rate. For an actual flow rate, a nozzle coefficient C_q is applied. Therefore,

$$w = 3600\, C_q A_n \sqrt{\frac{2g\gamma}{\gamma-1}\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma+1}{\gamma}}\right]} \qquad (1)$$

(*b*) Derivation of Eq. (5):
Maximum flow rate of a nozzle is:

$$w_{max} = 3600\, C_q \times A_n \times$$

$$\sqrt{\frac{2g\gamma}{\gamma-1}\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma+1}{\gamma}}\right]_{max}}$$

The maximum value can be found by differentiating w with respect to $\frac{p_2}{p_1}$ and setting the derivative equal to zero; thus,

$$\frac{d\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma+1}{\gamma}}\right]}{d\left(\frac{p^2}{p^1}\right)}$$

$$= \frac{2}{\gamma}\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}-1} - \frac{\gamma+1}{\gamma}\left(\frac{p_2}{p_1}\right)^{\frac{\gamma+1}{\gamma}-1} = 0$$

$$\left(\frac{p_2}{p_r}\right)_{crit} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \qquad (5)$$

(*c*) Derivations of Eqs. (6) and (7):
For an isentropic process, pv^{γ} = constant
Differentiating it with respect to v,

$$\frac{d}{dv}(p\, v^{\gamma}) = 0$$

$$p\,\gamma\, v^{\gamma-1} + v^{\gamma}\frac{dp}{dv} = 0$$

$$\frac{dp}{dv} = -\gamma\,\frac{p}{v}$$

$$-\frac{dp}{\left(\frac{dv}{v}\right)} = \gamma\, p$$

where $-\frac{dp}{\frac{dv}{v}}$ is the bulk modulus of compression

For incompressible fluid flow,

$$\left(\frac{dv}{v}\right) \approx 0, \gamma \approx \infty$$

Therefore, $\frac{\gamma + 1}{\gamma} \approx 1$ and $\frac{2}{\gamma} \approx 0$

and $\frac{\gamma}{\gamma - 1} \approx 1$

Therefore, Eq. (1) becomes,

$$w = 3600 \, C_q A_n \sqrt{2g\,(1)\,\frac{p_1}{v_1}\left[1 - \frac{p_2}{p_1}\right]}$$

$$= 3600 \, C_q A_n \sqrt{\frac{2g\,(p_1 - p_2)}{v_1}} \qquad (6)$$

Since $p_1 v_1 = RT_1$,

$$w = 3600 \, C_q A_n \quad \frac{2g\, p_1 (p_1 - p_2)}{RT_1} \qquad (7)$$

(*d*) Derivation of Eq. (17):

w_c = corrected throttle flow
w_t = test throttle flow
T = absolute temperature
p_d = design throttle pressure
p_t = test throttle pressure
p_o = test reheat pressure
p_c = corrected reheat pressure for plotting
v_d = design throttle specific volume
v_t = test throttle specific volume
v_{dr} = design reheat specific volume at design reheat temperature and test reheat pressure
v_{tr} = test reheat specific volume
v_{cr} = corrected reheat specific volume

The equation for flow through a turbine stage whose pressure ratio is constant is:

$$w = K \sqrt{p/v} \qquad (17a)$$

Applying this equation to throttle conditions gives:

$$\frac{w_c}{w_t} = \sqrt{\frac{p_d v_t}{p_t v_d}} \qquad (17b)$$

which is also applicable to reheat flow. For a constant reheat temperature the general flow equation may be written:

$$w = K \sqrt{\frac{p}{v}} = K \sqrt{\frac{p^2}{pv}} \qquad (17c)$$

Since, $pv = RT$ = constant (for constant temperature)

$$w = K \sqrt{\frac{p^2}{pv}} = K_1 \, p \text{ where: } K_1 = \frac{K}{\sqrt{pv}} \qquad (17d)$$

Combining Eqs. (17b) and (17d), the correction to reheat pressure for deviations from design throttle conditions is:

$$p_c = p_o \sqrt{\frac{p_d v_t}{p_t v_d}} \qquad (17e)$$

To correct for deviations in reheat temperature, apply Eq. (17a) for constant reheat flow, i.e., $w_c = w_t$

$$\frac{w_c}{w_d} = \frac{K \sqrt{\frac{p_c}{v_{cr}}}}{K \sqrt{\frac{p_o}{v_{tr}}}} = 1$$

or,

$$\frac{p_c}{v_{cr}} = \frac{p_o}{v_{tr}} \qquad (17f)$$

However, since v_{cr} is unknown, Eq. (17f) requires a trial and error solution. To obviate this need, a new condition, v_{dr}, is introduced which is the specific volume at design reheat temperature and test reheat pressure. The equation of state for this condition is:

$$p_o v_{dr} = RT_d \qquad (17g)$$

Similarly for the corrected conditions:

$$p_c v_{cr} = RT_d \qquad (17h)$$

Dividing Eq. (17g) by Eq. (17h),

$$\frac{p_o v_{dr}}{p_c v_{cr}} = 1 \qquad (17i)$$

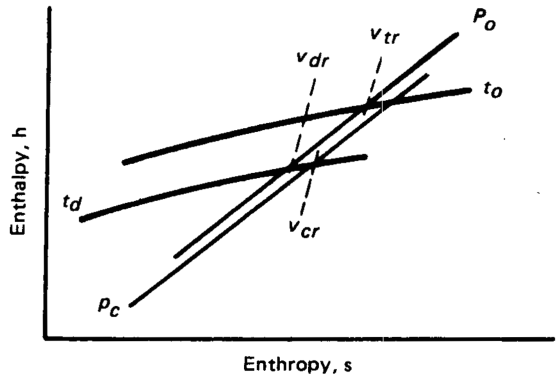


FIG. 6.16 RELATIONSHIPS OF v_{dr}, v_{cr}, AND v_{tr}

Combining Eqs. (17f) and (17i)

$$\frac{p_o \ v_{dr}}{p_o\left(\frac{v_{cr}}{v_{tr}}\right)v_{cr}} = 1 \tag{17j}$$

which reduces to

$$v_{cr} = \sqrt{v_{dr}\, v_{tr}} \tag{17k}$$

Substituting Eq. (17k) in Eq. (17f)

$$p_c = p_o \frac{\sqrt{v_{dr}\, v_{tr}}}{v_{tr}} = p_o \sqrt{\frac{v_{dr}}{v_{tr}}}$$

The total correction is obtained by combining Eqs. (17e) and (17f)

$$p_c = p_o \sqrt{\frac{p_d\, v_t}{p_t\, v_d}} \sqrt{\frac{v_{dr}}{v_{tr}}} \tag{17}$$

SECTION 7 — TEST FOR NONEXTRACTION CONDENSING TURBINE WITH SUPERHEATED INLET STEAM

7.1 INTRODUCTION

(a) A steam rate test is recommended for routine performance monitoring of nonextraction condensing turbines.

(b) Test procedures and instrumentation are listed below for the recommended test. Results obtained with the recommended procedures and instrumentation are repeatable within ± 0.5 percent when the driven element is an electric generator. (See para. 7.8 for derivation of this value and paras. 3.8 and 4.4.1 through 4.4.5 for additional discussion of test repeatability.) Repeatability of test results for mechanical drive turbines will depend on the type of driven element and the output measurements.

(c) The performance tests recommended in this Section will generally require three observers, including a test supervisor.

7.2 INSTRUMENTATION REQUIREMENTS

(a) Special test instrumentation is recommended for the following critical variables:

- Initial pressure
- Initial temperature
- First-stage pressure
- Exhaust pressure
- Power output, electrical or mechanical
- Condensate flow
- Condensate temperature

The recommended instruments will measure the critical variables with sufficient accuracy for test results that are repeatable to the percentage given in para. 7.1(b), if the Section 4 recommendations are observed.

(1) *Pressure.* See para. 4.5.

(2) *Temperature.* See para. 4.6.

(3) *Condensate Flow.* See paras. 4.4.1 through 4.4.3. Alternately, weighing tanks or volumetric measuring tanks may be used following the recommendations in PTC 6-1976, para. 4.20.

(4) *Power Output*

(a) *Electrical.* See para. 4.2.

(b) *Mechanical.* See para. 4.3.

(5) *Secondary Flows.* See para. 4.4.8.

(6) *Leakages.* See para. 4.9.

(7) *Storage Changes.* See para. 4.9. Levels measured to the closest one-eighth in. from a known reference point.

(b) For the use of manufacturer's data, see para. 4.11.

7.3 INSTRUMENT LOCATIONS

The location of instruments for measuring primary and secondary variables are given on Fig. 7.1. Items to be estimated from manufacturer's data are also indicated on this figure.

7.4 ISOLATION PROCEDURES

See paras. 3.2.1. through 3.2.7.

7.5 CONDUCT OF THE TEST

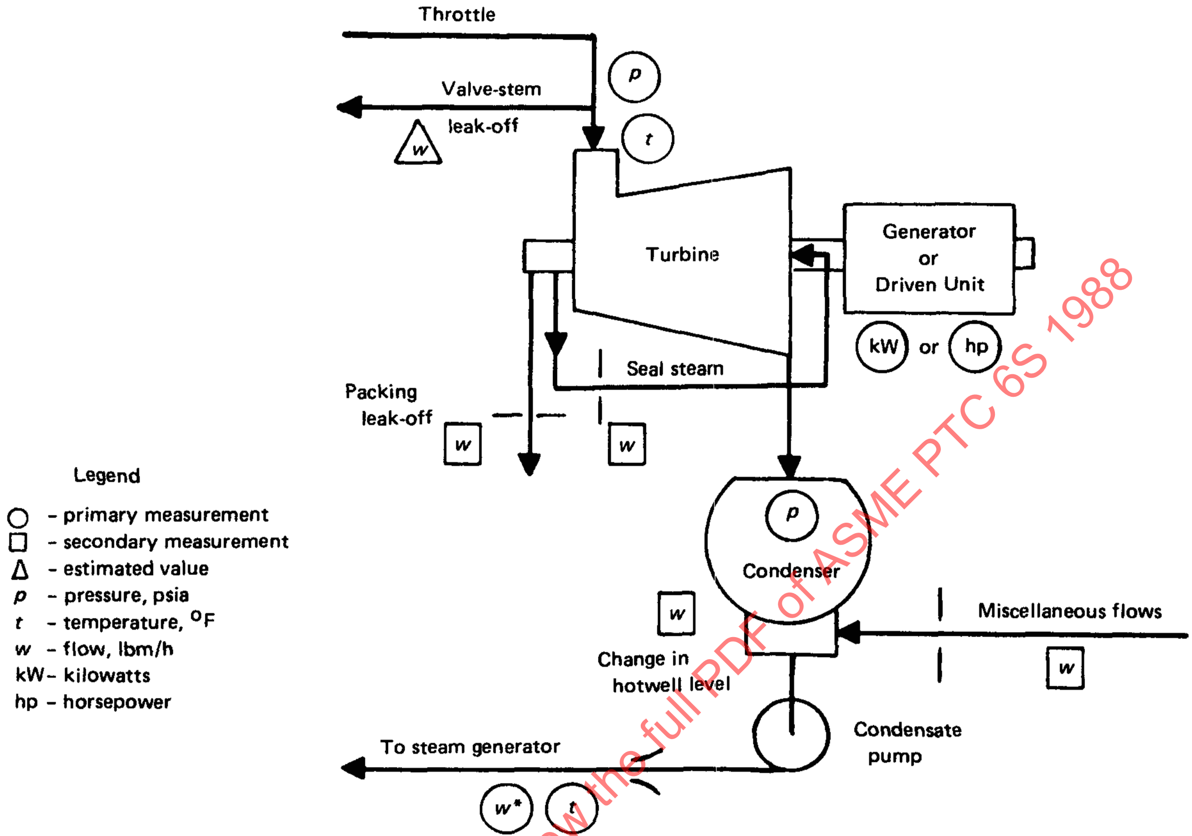
For turbines with multiple governing valves, the test should be made at a valve point. (See para. 5.1.) The general conduct of the test should be in accordance with para. 3.4. Turbine steam rate test should be of two hour duration.

7.6 CALCULATION OF TEST RESULTS

(a) *Data Preparation and Calculations.* Raw data should be examined for consistency and reliability. Paragraph 3.5 should be used as a guide for data reduction and calculation techniques.

(b) *Formula.* Steam rate, lbm per kWhr or lbm per hphr, is found from the following formula with nomenclature as given in Section 2.

$$\text{Steam rate} = w_1 / P_g$$



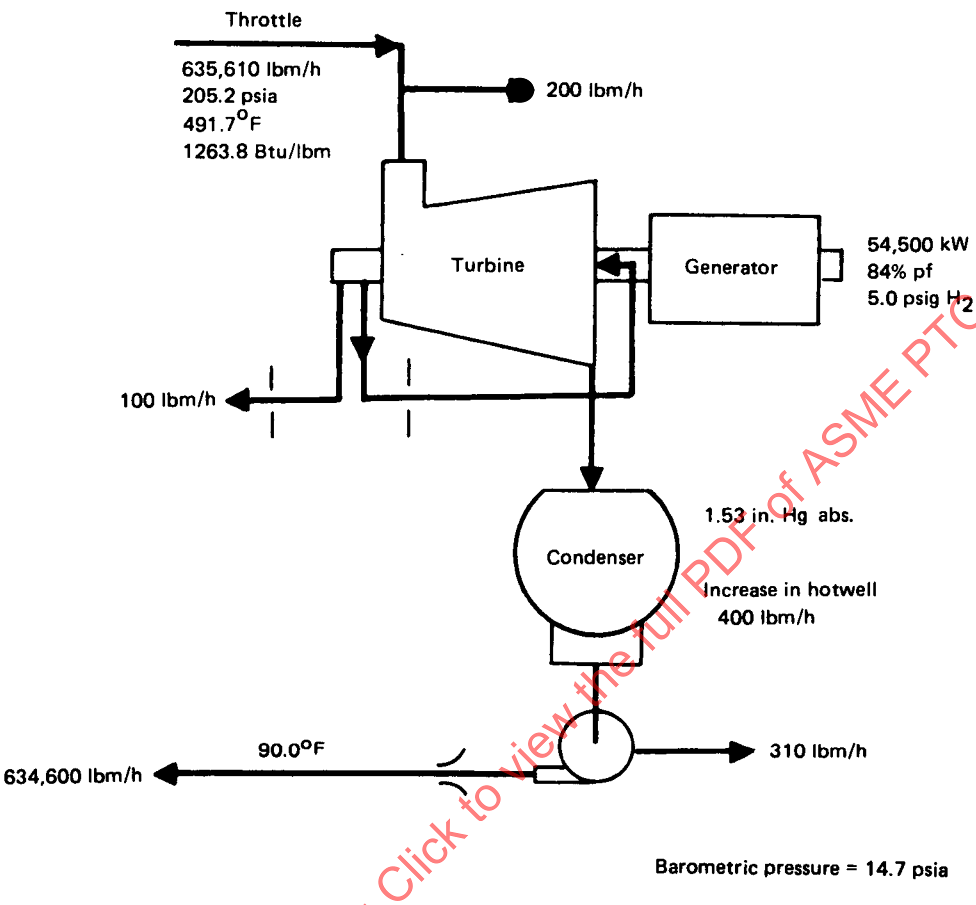
*NOTE: The flow at this point may be measured with either a nozzle (as indicated), an orifice, or weighing or volumetric measuring tanks (see para. 7.2).

FIG. 7.1 INSTRUMENT LOCATIONS

(c) *Test Correction Factors.* Correction factors in the form of divisors may be determined from the manufacturer's data or from previous tests to correct steam rate and output for deviations from specified values of throttle pressure, throttle temperature, and exhaust pressure.

(d) *Data Plots and Interpretation of Trends.* For suggested chronological plots designed to assist in the analysis of test data, refer to Section 6. Increasing corrected steam rates at the same valve point are indicative of deteriorating turbine performance. See para. 6.24 for possible causes.

7.7 SAMPLE CALCULATIONS



Specified conditions:

Initial pressure	185 psig
Initial temperature	500 °F
Exhaust pressure	1.5 in. Hg abs.
Hydrogen pressure	0.5 psig
Power factor	0.8

Reference steam rate:
11.63 lbm/kWhr based on value read from valve-point locus curve at generator corrected output

Throttle flow:

Measured condensate flow	634,600 lbm/hr
Increase in hotwell storage	400 lbm/hr
Hotwell pump gland leakage	310 lbm/hr
Valve stem leakage	200 lbm/hr
Gland leakage	100 lbm/hr
	635,610 lbm/hr

Generator output corrected to specified conditions:

Measured generator output	54,500 kW	
Losses for 0.84 pf and 0.5 psig H_2 pressure		626 kW*
Additional loss for 5.0 psig H_2 pressure		17 kW*
Total generator losses for test conditions		643 kW
Losses with specified conditions of 0.80 pf and 0.5 psig H_2 pressure		657 kW*

$$\text{Corrected generator output} = 54{,}500 + 643 - 657 = 54{,}486 \text{ kW}$$

Test steam rate:

$$\text{Steam rate} = \frac{\text{initial throttle flow}}{\text{corrected generator output}}$$

$$= \frac{635{,}610}{54{,}486} = 11.67 \text{ lbm/kWhr}$$

Corrections for deviations from specified conditions:

$$\text{Flow correction factor} = \sqrt{\frac{p_s}{p_t} \times \frac{v_t}{v_s}}$$

p = pressure, psia
v = specific volume, ft^3/lbm

Subscripts:
s = specified conditions
t = test conditions

$$= \sqrt{\frac{199.7}{205.2} \times \frac{2.624}{2.729}} = 0.9673$$

Corrected throttle flow = 0.9673 × 635,610 = 614,826 lbm/hr (278,880 kg/h)

Steam rate correction divisors:
(From manufacturer's curves)

	Percent Change	Correction Divisor
Initial pressure	−0.17	0.9983
Initial temperature	0.78	1.0078
Exhaust pressure	0.10	1.0010
Combined divisor		1.0071

$$\text{Corrected steam rate} = \frac{\text{test steam rate}}{\text{combined divisor}}$$

$$= \frac{11.67}{1.0071} = 11.59 \text{ lbm/kWhr (5.257 kg/kWh)}$$

*From Fig. 7.2

Generator output corrected to specified conditions:

$$\text{Corrected output} = \frac{\text{corrected throttle flow}}{\text{corrected steam rate}}$$

$$= \frac{614{,}826}{11.59} = 53{,}048 \text{ kW}$$

Corrected steam rate compared to reference:

$$\text{Percent change} = \frac{11.63 - 11.59}{11.63} \times 100 = 0.34\% \text{ better}$$

7.8 CALCULATION OF EXPECTED REPEATABILITY

(*a*) The repeatability value given in para. 7.1 was derived from the uncertainty values for the instrumentation recommended in para. 7.2 and the correction factor curves referred to in the sample calculations.

(*b*) The uncertainties in measuring each variable from PTC 6 Report-1985 are as follows:

Variable	Instrumentation	Uncertainty, ±
Condensate flow	Throat-tap nozzle, calibrated before installation and inspected before and after test ($U_B = 0.35$), 0.5 beta ratio ($U_\beta = 0.0$), 10-D straight pipe upstream ($U_{LS1} = 0.4$), 16 section straightener ($U_{LS2} = 0.36$), 4-D straight pipe downstream ($U_{DSL} = 0.51$).	0.82%
Throttle pressure	Transducer, medium accuracy laboratory calibrated	0.1%
Throttle temperature	Test thermocouple, separate test leads, calibrated against secondary standard with 0.05% potentiometer	3 °F
Exhaust pressure	Transducer with one probe per 64 square feet located per Code, paras. 4.92 and 4.93.	0.1 in. Hg
Generator output	Transformers with calibration curves, volt amperes and power factors of burdens available.	CT = 0.1% PT = 0.3%
	Three-phase electronic watt-hour meter with high accuracy digital readout, calibrated before test.	0.15%
Packing flows and leakages	Estimated	61 lbm/hr
Hotwell storage (3 ft diameter)	Scale ± 1/8 in.	4.6 lbm/hr

(*c*) *Uncertainty of Corrected Steam Rate*

(*1*) The uncertainty in the throttle flow is the square root of the sum of the squares of the uncertainties in condensate flow, the leakage flows, and hotwell storage.

$$\text{Uncertainty} = \sqrt{(5204)^2 + (61)^2 + (4.6)^2} =$$

$$\pm 5204 \text{ lbm/hr or } \pm 0.85\%$$

(*2*) The uncertainty in the output measurement using the specified instrumentation is the square root of the sum of the squares of the CT, PT, and watthour meter uncertainties.

$$\text{Uncertainty} = \sqrt{(0.10)^2 + (0.30)^2 + (0.15)^2} = \pm 0.35\%$$

(*3*) The uncertainty in the corrected output is the square root of the sum of the squares of the uncertainties in the measured output and the correction to specified generator losses. The power factor is assumed to have the same uncertainty as the power measurement and the hydrogen pressure is assumed to be measured with a calibrated 8 in. Bourdon-tube gage.

$$\text{Uncertainty} = \sqrt{(190.75)^2 + (2.19)^2 + (1.13)^2} =$$

$$\pm 190.8 \text{ kW or } \pm 0.35\%$$

(4) The uncertainty in the corrected steam rate is the square root of the sum of the squares of the uncertainties in throttle flow, corrected output, and the correction factors for throttle pressure, throttle temperature, and exhaust pressure, or

Uncertainty =

$$\sqrt{(0.85)^2 + (0.35)^2 + (0.057)^2 + (0.272)^2 + (0.4)^2} =$$

$\pm 1.040\%$

The repeatability is taken as one-half the uncertainty or $\pm 0.52\%$.

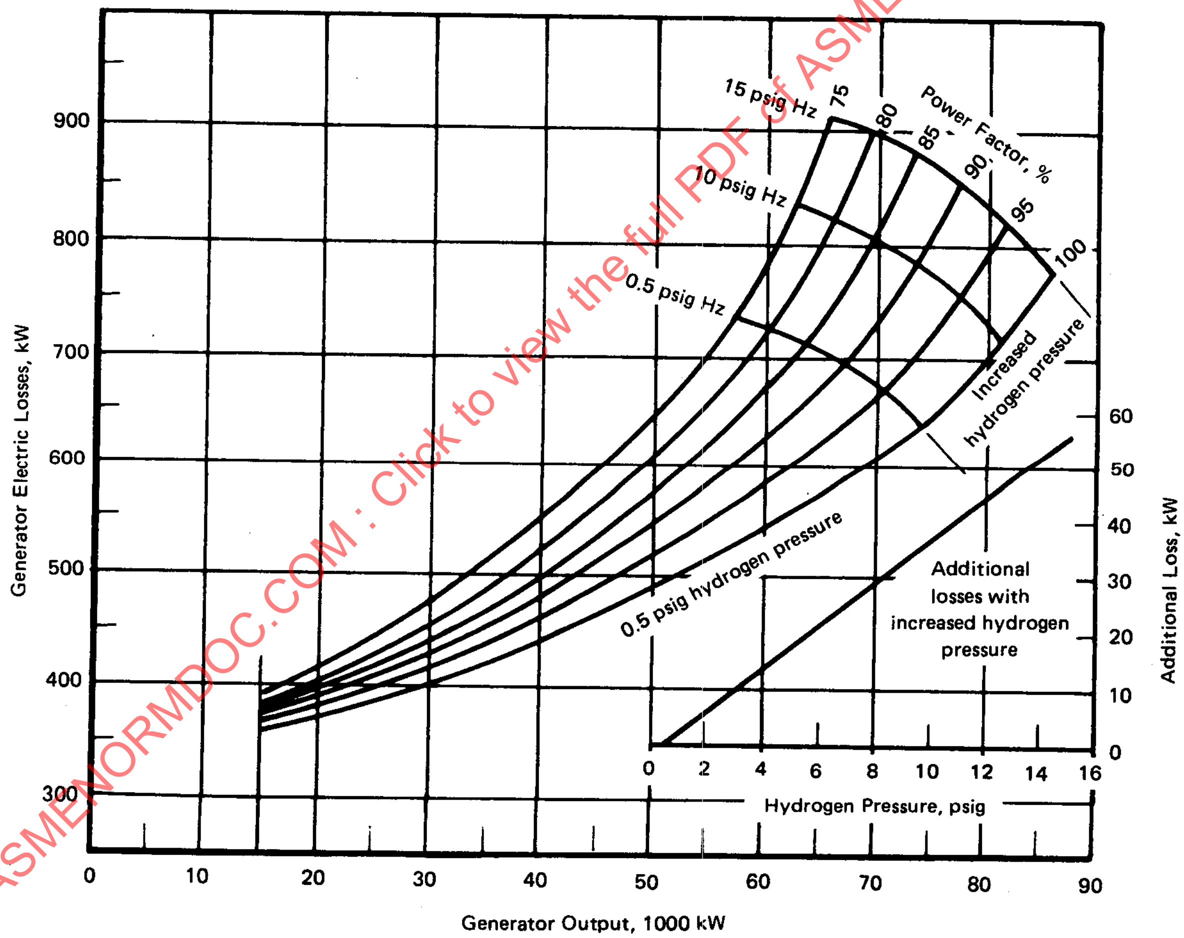


FIG. 7.2 GENERATOR ELECTRICAL LOSSES

SECTION 8 — TEST FOR CONDENSING TURBINE, REGENERATIVE CYCLE, WITH SUPERHEATED INLET STEAM

8.1 INTRODUCTION

(*a*) A maximum-capability test is the recommended frequent periodic routine test of the performance of this type of turbine. When such a test indicates a performance change, it may be necessary to conduct a simplified heat rate test to determine the cause.

(*b*) A simplified turbine heat rate test is recommended as a periodic routine check of the performance of this type of turbine. When change of performance is indicated, additional measurements should be made to locate the source of this change. These additional measurements include readings of all available stage pressures, gland leakages and heater terminal temperature differences.

(*c*) Recommended test procedures, instrumentation requirements, reading frequencies and test duration have been chosen to measure critical variables with sufficient accuracy to produce results of the maximum-capability test that are repeatable within ±0.5 percent* on a day-to-day basis. Similarly, simplified heat-rate test results should be repeatable within ±0.5 percent*. (See para. 3.8.3 and refer to paras. 4.4.1 through 4.4.5 for factors affecting expected repeatability over longer periods of time.)

(*d*) It is estimated that the recommended test will require a minimum number of four observers plus a test supervisor. If additional data beyond that required for heat rate determination is desired, additional observers may be needed.

8.2 INSTRUMENTATION REQUIREMENTS

(*a*) Special test instrumentation is recommended for the following readings:

	Maximum Capability	Heat Rate
First stage pressure	x	x
Generator output	x	x
Feedwater flow	...	x
Throttle steam pressure	x	x
Throttle steam temperature	x	x
Turbine exhaust pressure	x	x
Final feedwater temperature	...	x

The recommended instruments have been chosen to measure critical variables with sufficient accuracy to produce results that are repeatable as indicated in para. 8.1(c) if the recommendations of Section 4 for their installation and use are observed.

(*1*) *Generator Output*. See paras. 4.2.1 through 4.2.7.

(*2*) *Feedwater flow:*
Primary element: See paras. 4.4.1 through 4.4.4.
Location of primary element: Upstream of deaerator or upstream of steam generator

(*3*) *Pressure*. See para. 4.5.

(*4*) *Temperature*. See para. 4.6.

(*5*) *Low-Pressure-Turbine Exhaust Pressure*. See paras. 4.5.8 and 4.5.9.

(*b*) *Secondary Readings*. See para. 4.10.

8.3 INSTRUMENT LOCATIONS

The essential instrumentation for a simplified heat rate test is indicated on Fig. 8.1. Also shown are additional measurements which are of value in determining the causes of performance deterioration.

8.4 ISOLATION PROCEDURE

See paras. 3.2.1 through 3.2.7.

8.5 CONDUCT OF THE TEST

8.5.1 Test Conditions

(*a*) Isolate the turbine cycle in accordance with paras. 3.2.1 through 3.2.7.

(*b*) Establish test load so that the turbine operates at a known valve point, preferably at valves wide open

*See Table 8.1 for derivation of this value.

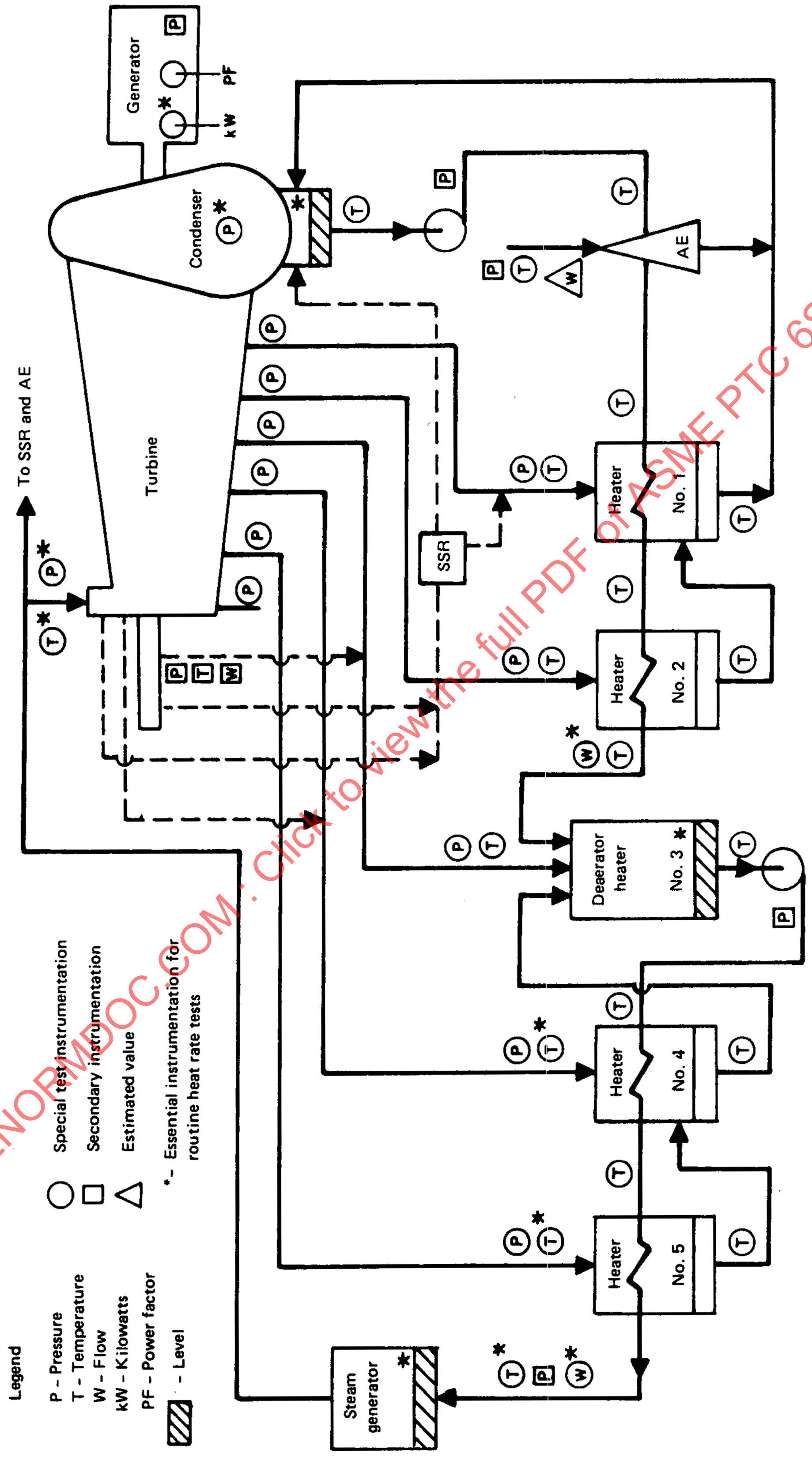


FIG. 8.1 INSTRUMENTATION FOR ROUTINE PERFORMANCE TESTS FOR CONDENSING TURBINE, REGENERATIVE CYCLE, SUPERHEATED INLET STEAM

(See para. 5.1) with operating conditions as close to specified conditions as possible and on load-limit control. In the case of single-valve or throttling turbines, reference measurements should be established to insure that the same percentage openings of the admission valves are established as for previous tests. The unit should be removed from automatic system-load-control devices and made as free of system disturbances as possible.

(c) Allow sufficient time to assure stable operating conditions. A one-half-hour minimum stabilization time should be allowed.

8.5.2 Test Duration. Turbine heat rate tests should be of two hour duration. Capability tests and special tests to determine relative effects upon heat rate of changes to the turbine cycle, which can be performed without changing throttle flow, may be of one hour duration.

8.5.3 Frequency and Coordination of Readings

(a) Readings should be coordinated by reliable time measurement in accordance with para. 4.8.

(b) Reading frequency should be established to produce a representative average. See para. 3.4.2 for frequency of test readings.

8.6 CALCULATION AND ANALYSIS OF TEST DATA

A general outline for a typical calculation of test results is provided in para. 8.8 and should be consulted as a guide.

8.6.1 Definition of Heat Rate. Heat rate in Btu/kWhr is defined by the following formula with nomenclature given in Section 2 and Fig. 2.1:

$$\text{Heat Rate} = \frac{w_1 h_1 - w_{11} h_{11}}{P_g}$$

8.6.2 Test Correction Factors

(a) Correction factors in the form of divisors shall be determined from manufacturer's data or prior tests to correct heat rate and output for deviations from specified values of throttle pressure and temperature and exhaust pressure.

(b) Correction factors to correct heat rate and output for deviation in top feedwater heater performance, condensate subcooling in the condenser and changes in pump-sealing and gland-cooling requirements may be determined from curves presented in PTC 6.1-1984 and Appendix A of this report. When sufficient information is available, correction factors may be developed from data from prior test analysis or cycle modeling.

8.6.3 Data Plots and Test Analysis. The presentation and interpretation of test results is discussed in Section 6.

Suggested parameters for performance monitoring of a condensing regenerative turbine supplied with superheated steam are:

(a) chronological plots of corrected heat rate and corrected generator output

(b) chronological plots of corrected stage pressures or $w/\sqrt{p/v}$

(c) when a sufficient number of test results at different valve points are available, corrected heat rate may be plotted against corrected output or corrected throttle flow. A comparison of these plots would show any changes in the timing (or overlap) of the governing valves.

8.6.4 Interpretation of Trend Changes. Increasing cycle heat rate at the same valve point, after correction to specified conditions, is indicative of reduction in performance of one or more components of the turbine or its cycle. The basic turbine heat rate test usually does not provide sufficient information to adequately analyze all components of the cycle, and it may be necessary to perform supplementary tests as shown in para. 8.7. Deterioration of component performance can be caused by several factors, such as:

(a) accumulation of turbine-blade deposits, or internal turbine damage. This is usually indicated by a change in stage pressures or stage pressure ratios. (See paras. 6.24 through 6.24.4).

(b) increases in leakage flows. (See paras. 6.24.3 and 6.24.5.)

(c) increased feedwater-heater terminal temperature differences resulting from reduced heat transfer coefficient, internal leakage, inadequate venting or failure to control condensate level (See para. 6.24.2.)

8.7 SUPPLEMENTARY TESTS

Supplementary tests, providing additional information required for analyzing the cause of performance deterioration, may be conducted in conjunction with the simplified heat rate test, or separately. When conducted separately, stable operating conditions and the same governing valve position that existed during the test should be used as the basis of comparison.

(a) Stage pressures can be measured periodically, preferably with valves wide open. The minimum recommended test duration is 30 minutes. See paras. 6.13 to 6.13.4 for corrections due to deviation of throttle pressure from specified value. A comparison of pressure ratios for turbine sections or for groups of stages, including the governing valves, is also of value.

(b) Terminal temperature differences (and drain-cooler temperature differences) of each feedwater heater in the heater cycle indicate changes in heater performance which affect heat rate results. Special temperature and pressure measurements should be taken at each point of steam and water entry or exit, at the specified valve openings. Test duration should be at least 30 minutes. Usually two observers are required for this test.

(c) Packing leakages can be measured with secondary flow-measuring devices at the specified valve opening under stable conditions. Increased packing-leakage flow is indicative of deterioration of internal packing clearances, resulting in poorer heat rates. Leakage tests should be a minimum of one hour in duration and usually require two to three observers.

8.8 SAMPLE CALCULATIONS

Specified Conditions

Throttle steam pressure, psia	1265*
Throttle steam temperature, °F	950*
Throttle steam enthalpy, Btu/lbm	1468.1*
Exhaust pressure, in. Hg abs.	2.00*
Power factor, %	80.0*
Hydrogen pressure, psig	30.0*

Reference heat rate: 9021 Btu/kWh, with maximum valve opening and with all operating conditions at the specified values.

Test Results

Throttle steam pressure, psia	1280*
Throttle steam temperature, °F	946*
Throttle steam enthalpy, Btu/lbm	1465.2*
Exhaust pressure, in.Hg abs.	2.09*
Feedwater leaving No. 5 heater:	
Temperature, °F	462.3
Enthalpy, Btu/lbm	444.6
Feedwater flow to steam gen.,lbm/hr	609,230
Generator output, kW	69,552*
Power factor, %	91.4*
Hydrogen pressure, psig	28.5*

Steam flows: Changes in stored water during 2 hr test period:

Condenser hotwell storage = −3000 (level fall) lbm
Steam generator storage = + 300 (level rise) lbm
Deaerator storage = 0 (no change)
Make-up = 0

System leakage $= \dfrac{3000-300}{2} = 1350$ lbm/hr (assumed to be from steam generator)

Steam to air ejector = 630 lbm/h (estimated on basis of previous data)

Throttle steam flow = feedwater flow − system leakage − increase in steam generator storage − steam to air ejector

$$= 609{,}230 - 1350 - \frac{300}{2} - 630$$

$$= 607{,}100 \text{ lbm/hr}$$

Generator Losses (From manufacturer's curves)

Output	= 69,552 kW*
80% power factor, losses	= 900 kW*
91.4% power factor, losses	= 790 kW*
Correction to 80% power factor	= −110 kW*

Additional Losses with increased Hydrogen Pressure

Hydrogen pressure was 28.5 psig, instead of the specified 30 psig, decreasing losses by 5 kW*

Generator output corrected to specified power factor and hydrogen pressure

$$= 69{,}552 - 110 - 5 = 69{,}437 \text{ kW*}$$

Heat Rate

$$\text{Test heat rate} = \frac{607{,}100 \times 1465.2 - 609{,}230 \times 444.6}{69{,}437}$$

$$= 8910 \text{ Btu/kWhr}$$

*Indicates typical values used for capability tests. The entire calculation shown here is required for heat rate tests.

Correction Factors (From manufacturer's curves)

	Heat Rate	Output
Throttle pressure	0.9994	1.0080*
Throttle temperature	1.0015	0.9980*
Exhaust pressure	1.0000	1.0000*
Product of correction factors	1.0009	1.0060*

Corrected data

$$\text{Corrected heat rate} = \frac{8910}{1.0009} = 8902 \text{ Btu/kWhr (9392 kJ/kWh)}$$

$$\text{Corrected output} = \frac{69{,}437}{1.0060} = 69{,}023 \text{ kW*}$$

$$\text{Heat rate below reference} = \frac{(8902 - 9021)}{9021} \times 100 = -1.32\%$$

$$\text{Corrected throttle flow} = \text{test flow} \times \sqrt{\frac{p_s}{p_t} \times \frac{v_t}{v_s}} = 607{,}100 \times \sqrt{\frac{1265}{1280} \times \frac{0.6090}{0.6191}}$$

$$= 598{,}600 \text{ lbm/h (271,500 kg/h)}$$

*Indicates typical values used for capability tests. The entire calculation shown here is required for heat rate tests.

TABLE 8.1
REPEATABILITY OF TEST RESULTS

		Effect of Uncertainty on		
Variable	**Instrumentation of Sample Test**	**Variable**	**Heat Rate**	**Capability**
(A) Generator output:				
(1) Potential transformers	Meeting Code requirement	±0.1%	±0.1%	±0.1%
(2) Current transformers	Other burdens	±0.2%*	±0.2%	±0.2%
(3) Watthour meters	Permanent; 3-phase calibration; mechanical register	±0.5%	±0.5%	±0.5%
Total for generator output	$\sqrt{(1)^2 + (2)^2 + (3)^2}$	±0.55%	±0.55%	±0.55%
(B) Primary flow:				
(1) Flow to deaerator	Pipe-wall tap nozzle, calibrated before installation, inspected before and after test with no change in flow element, 10-D upstream straight pipe, 50-section straightener β-Ratio = 0.5, 5.25-D downstream straight pipe, U_B = 0.60, U_β = 0, U_{LS1} = 0.40, U_{LS2} = 0, U_{DSL} = 0.3	±0.78%	±0.78%	...
(2) Feedwater flow to steam generator (alternate location)	Pipe-wall tap nozzle, no calibration, inspected before installation, 10-D upstream straight pipe, 20-section straightener, β-Ratio = 0.65, 4-D downstream straight pipe, U_B = 3.2, U_β = 0.5, U_{LS1} = 1.0 U_{LS2} = 0.8, U_{DSL} = 0.67	±3.55	±3.55%	...
(C) Throttle pressure	2000 psi transducer, deadweight tester calibrated, 0.50% of full scale	±0.8%	±0.03%	±0.8%
(D) Throttle temperature	Meeting Code requirement	±1.0°F	±0.02%	±0.05%
(E) Final feedwater temperature	Meeting Code requirement	±1.0°F	±0.10%	...
(F) Exhaust pressure	1 probe per 64 sq ft	±0.1 in. Hg	±0.2%	±0.2%
Exhaust pressure	Test manometer	±0.05 in. Hg	±0.1%	±0.1%
Combined uncertainty:				
With recommended flow measurement: $\sqrt{A^2 + (B_1)^2 + C^2 + D^2 + E^2 + F^2}$			±0.99%	±1.00%
With alternate flow measurement: $\sqrt{A^2 + (B_2)^2 + C^2 + D^2 + E^2 + F^2}$			±3.60%	±0.99%
Repeatabilities = 0.5 × (combined uncertainty)				
With recommended flow measurement			±0.49%	±0.50%
With alternate flow measurement			±1.80%	±0.50%

*Estimated.

SECTION 9 — TEST FOR CONDENSING TURBINE, REHEAT-REGENERATIVE CYCLE, WITH SUPERHEATED INLET STEAM

9.1 INTRODUCTION

(*a*) The enthalpy-drop efficiency test of turbine sections operating in the superheated steam region, combined with monitoring of generating capabilities at a known governing valve point, are recommended as the best routine performance tests for reheat turbines. When these tests detect performance deterioration, it may be necessary to conduct efficiency tests of other cycle components or simplified heat-rate tests to pinpoint the cause of the deterioration.

(*b*) The enthalpy-drop efficiency test consists of measuring the initial and final steam temperatures and pressures of turbine sections using superheated steam and calculating an engine efficiency from the resulting enthalpies. Although the test is simple, measurements must be precise. Instrumentation requirements and specific test procedures are presented in this section.

(*c*) The capability test consists of measuring electrical output at a particular governing valve point, preferably valves wide open, and steam pressures and temperatures which are necessary to apply load correction factors. Measured electrical output must be corrected for variation from specified values of power factor and hydrogen pressure prior to application of the load correction factors. This involves calculating the test coupling kilowatts and then subtracting the electrical and mechanical losses that would occur at specified power factor and hydrogen pressure. Instrumentation and procedures for this test are also included.

(*d*) The simplified heat-rate test consists of measuring the heat supplied to the turbine cycle and the electrical output at given valve points. In addition to the instrumentation of the recommended tests, this test requires flow measurement, cycle isolation and measurement of certain feedwater temperatures and pressures. Specific test procedures are presented for the simplified heat-rate test.

(*e*) Test procedures, instrumentation, duration and reading frequency are listed and were chosen to measure critical variables with sufficient accuracy to produce results estimated to be repeatable with the following percentages. (See para. 3.8.3):

Test	Repeatability [Note (1)]
Enthalpy-drop efficiency	±0.3% to ±0.5% depending upon range of available energy
Corrected generating capability	±0.3%
Simplified heat rate [Note (2)]	±0.55% to ±0.90% depending on location of primary water flow measuring element. See para. 4.4.2.

NOTES:
(1) See para. 9.9 for derivation of these values.
(2) Because of the importance of the flow measurement in this test, these values are based on short time spans (day-to-day) only. For expected repeatability over longer periods of time, indicators given in paras. 4.4.1 through 4.4.5 should be considered.

(*f*) The recommended enthalpy-drop and generating capability tests require an estimated minimum number of three observers. Simplified heat-rate tests require a minimum number of six observers plus a test supervisor. A fewer number of observers may be necessary when automatic data acquisition is used.

9.2 INSTRUMENTATION REQUIREMENTS

(*a*) Precision instrumentation is recommended for primary readings for each test as follows:

Reading	Enthalpy Drop	Generating Capability	Heat Rate
Throttle pressure and temperature	x	x	x
First-stage shell pressure	x	x	x
Cold-reheat pressure and temperature	x	x	x

Reading	Enthalpy Drop	Generating Capability	Heat Rate
Hot-reheat pressure and temperature	x	x	x
Crossover pressure and temperature	x	...	...
Final feedwater pressure and temperature	...	...	x
LP turbine exhaust pressure	...	x	x
Feedwater flow [Note (1)]	...	...	x
Generator output	...	x	x

NOTE:
(1) If the flow measurement is located upstream of the feedwater exit from the the turbine cycle, precision instrumentation is recommended around all feedwater heaters between the point of flow measurement and the water discharge from the cycle.

The appropriate paragraphs of Section 4 should be consulted for each category of instrumentation.

(1) *Pressure*. See para. 4.5.1. Calibrated pressure transducer with suppressed range for accurate readout, 0 to 5000 psia.

(2) *Temperature*. See paras. 4.6.1 through 4.6.4.

(3) *Feedwater Flow*. See paras. 4.4.1 through 4.4.5. The flow nozzle should be designed in accordance with para. 4.4.3 and flow section located in the feedwater system between the steam generator inlet and the highest pressure feedwater heater outlet. The primary flow element may also be located in the condensate system between the condensate pump discharge and feedwater pump suction.

(4) *Reheater Flow*. By heat balance, preferably, or by permanently installed steam nozzle with an indirect calibration less than 6 months old.

(5) *Generator Output*. See para. 4.2.

(6) *Secondary Flows*. See para. 4.4.8.

(7) *Leakages*. Suitably collected and measured. See para. 4.9.

(8) *Storage Changes*. Levels measured to the closest ⅛ in. from a known reference point.

(*b*) For the recommended use of station instruments, see para. 4.10.

(*c*) For the use of manufacturer's data, see para. 4.11.

9.3 INSTRUMENT LOCATIONS

(*a*) Figure 9.1 (a) shows schematic arrangements of pressure and temperature instrument locations for enthalpy-drop efficiency tests and corrected maximum generating capability.

(*b*) Figure 9.1 (b) also shows the instrumentation required for heat-rate determination. Certain required data obtained from manufacturer's curves are shown on this figure.

9.4 OUTLINE OF ISOLATION PROCEDURE

(*a*) For enthalpy-drop efficiency tests, isolation is generally not required. However, cycle conditions affecting turbine-section pressure ratios should be consistent among tests.

(*b*) For generating-capability and heat-rate tests, isolate the turbine cycle so that the unaccounted-for water loss is less than 0.5 percent of maximum throttle flow. (See para. 3.2.)

(*c*) Prior to each heat-rate and generating capacity test, isolation valves should be placed in the desired position and tagged. Performance test results shall be based only on information obtained when the turbine cycle is isolated as indicated above.

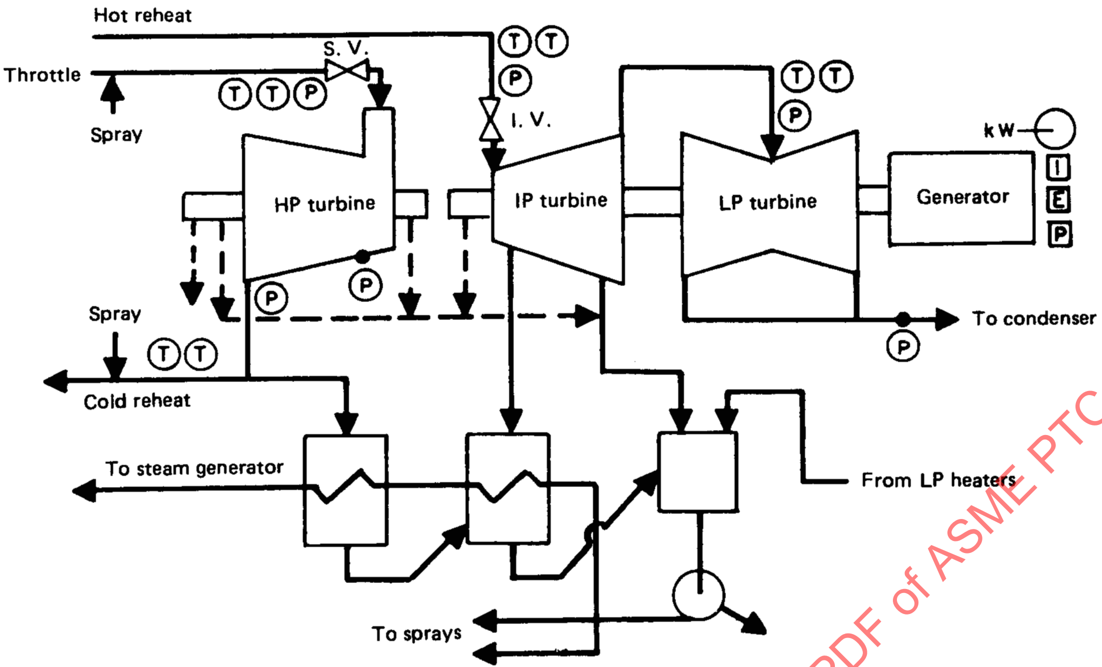
9.5 CONDUCT OF THE TEST

Enthalpy-drop efficiency tests and heat-rate tests should be conducted at valve points to avoid throttling losses through partially open valves and to permit the test point to be repeated over a period of years. Capability tests can also be run at valve points but it is preferable to conduct this test at the valves-wide-open position. The turbine should be on load-limit control and the generator should be free of system disturbances. A minimum of one-half hour stabilization period should be provided prior to the test run. Consult para. 3.4.2 for the recommended reading frequency and for the pertinent information required to conduct the test.

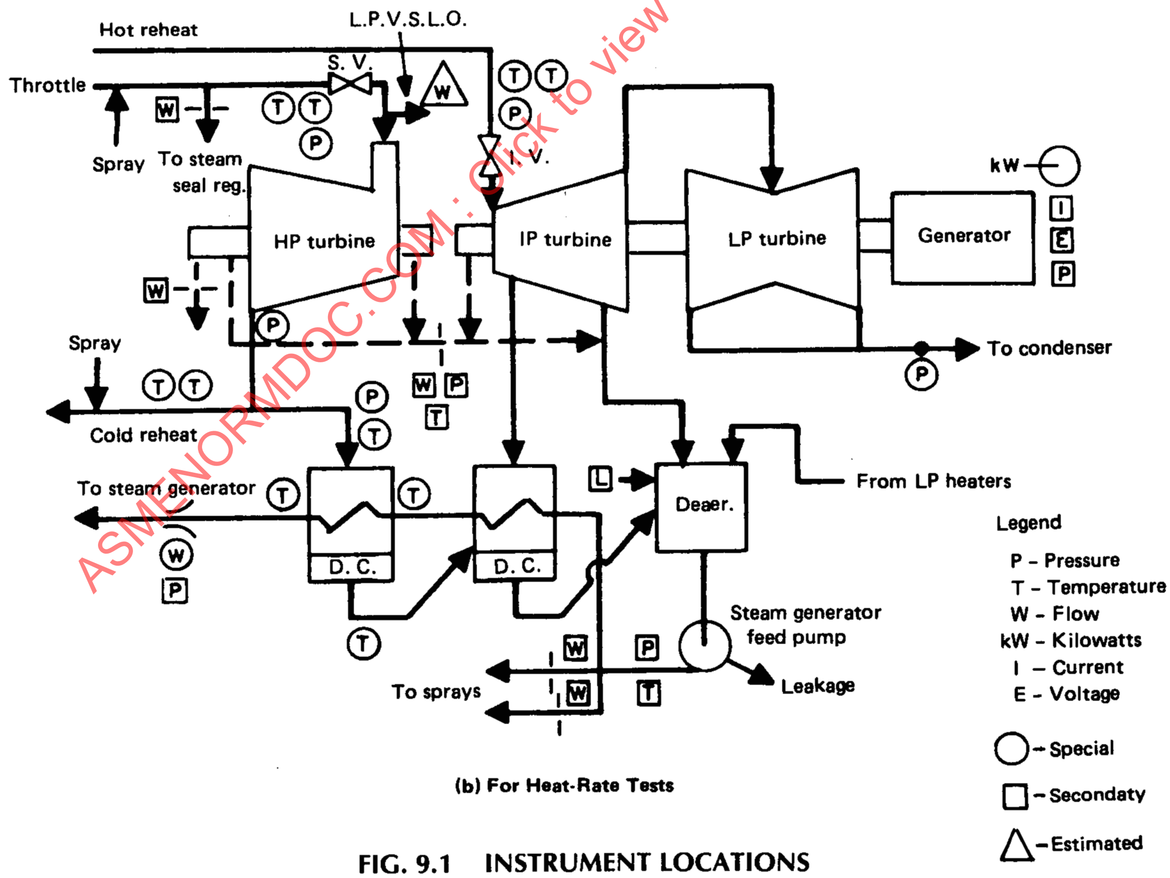
Recommended test duration is one hour for enthalpy-drop efficiency tests and generating-capability tests and two hours for heat-rate tests. Shorter durations have a direct effect upon accuracy and repeatability of test results.

9.6 CALCULATION OF TEST RESULTS

9.6.1 Data Preparation and Calculations. Raw data should be examined for consistency and reliability. Paragraph 3.5 should be consulted as a guide for data reduction and calculation techniques.



(a) For Enthalpy-Drop Efficiency Tests and Corrected Maximum Capability



(b) For Heat-Rate Tests

FIG. 9.1 INSTRUMENT LOCATIONS

9.6.2 Formulas and Sample Calculation

(*a*) Enthalpy-drop efficiency formula and sample calculations are given in Fig. 5.3 and in para. 9.8.

(*b*) Corrected capability sample calculations at valves wide open are shown in para. 9.8.

(*c*) Heat rate, Btu per kWh, is found from the general definition given in Section 2. Sample calculations of heat rate are shown in para. 9.8.

9.6.3 Heat Rate and Output Correction Factors

(*a*) Correction factors can be determined from manufacturer's data or prior tests to correct heat rate and output for deviations from specified values of throttle pressure, throttle temperature, reheat temperature, reheater pressure drop, and exhaust pressure. Correction factor curves normally supplied by the manufacturer are used as divisors and can be verified by test, if necessary.

(*b*) When sufficient test information is available, correction factors can be determined from data developed from prior tests or from the general curves of Appendix A of this report to correct heat rate and output for deviation of feedwater-heater performance, changes in desuperheating spray-water requirements, condensate sub-cooling in condenser and changes in pump-sealing and gland-cooling requirements.

9.6.4 Data Plots and Test Analysis. Enthalpy-drop tests and corrected heat-rate tests conducted at several different valve points can be compared in tabular form or curve form. Turbine section efficiencies and corrected heat rate at the same valve point and maximum generating capability at valves wide open can be compared chronologically either by test result or by deviation from some standard value. For heat-rate tests, when the test result is plotted, the estimated range of uncertainty should be clearly indicated. Suggested parameters for curve presentation and guidance for curve construction and interpretation are given in paras. 6.20 through 6.21.1, 6.23 and 6.23.1.

A change in HP turbine efficiency results in a change in output produced by that turbine section and a change in reheater duty. A change in IP turbine efficiency results in a change in output of both the IP and LP turbine sections. The change in efficiency of each of these turbine sections can be translated to a change in heat rate. (See ASME paper 60-WA-139, "Methods for Measuring Steam Turbine-Generator Performance" by K. C. Cotton and J. C. Westcott.)

(1) Worth of $\Delta\eta_{hp}$ in heat rate

$$\Delta HR\% = \frac{\Delta\eta_{hp}\%\,(UE_{hp})\,(Q_{hp})}{3412.142\,(kW_{tot})} - \frac{\Delta\eta_{hp}\%\,(UE_{hp})\,(Q_{rhtr})}{HR\,(kW_{tot})}$$

(2) Worth of $\Delta\eta_{ip}$ in heat rate

$$\Delta HR\% = \Delta\eta_{ip}\%\left[\frac{UE_{ip}}{UE_{rh}}\,(L.F.)\right]\left[1 - \frac{UE_{hp}\,(Q_{hp})}{3412.142\,(kW_{tot})}\right]$$

where UE = used energy, Btu/lbm
η = turbine section efficiency, %
$\Delta\eta = \eta_1 - \eta_2$
$\Delta\eta\% = \Delta\eta\,(100)/\eta_1$
Q = flow, lbm/hr
kW = generator output, kilowatts
HR = turbine heat rate, Btu/kWhr
$\Delta HR\% = (HR_1 - HR_2)\,100/HR_1$
$L.F.$ = loss factor (See Fig. 9.2)

and subscripts
hp = high-pressure turbine section
ip = intermediate-pressure turbine section
rh = entire reheat turbine
$rhtr$ = reheater
tot = total
1, 2, etc. = as defined in Section 2

9.7 SUPPLEMENTARY TESTS

In order to provide information to aid in the analysis of the cause of deterioration in turbine performance, certain supplementary tests are presented which, because of their simplicity, can be conducted with a minimum number of personnel and instrumentation. These tests may be made either in conjunction with simplified turbine heat-rate tests with some increase in both instrumentation and personnel, or as separate tests. Care should be exercised to provide stable conditions and turbine load settings comparable to those to which the results are to be compared. Paragraphs 9.2 through 9.4 shall be consulted as required.

(a) Stage pressures, and the pressure ratio across groups of stages, conducted at a valve point, preferably with all valves open, under stabilized conditions, can be used for periodic checking of turbine performance (see paras. 6.13 through 6.14). Secondary pressures can be read at each extraction stage and the turbine first stage for a test which should last a minimum duration of 30 minutes.

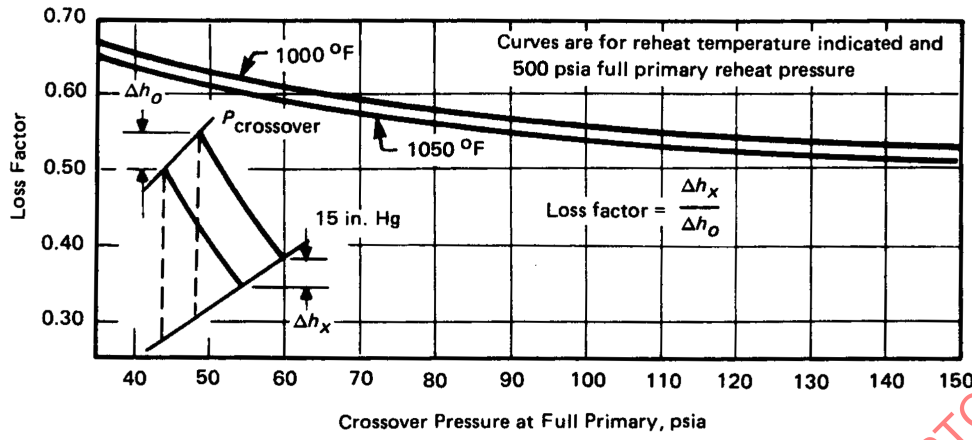


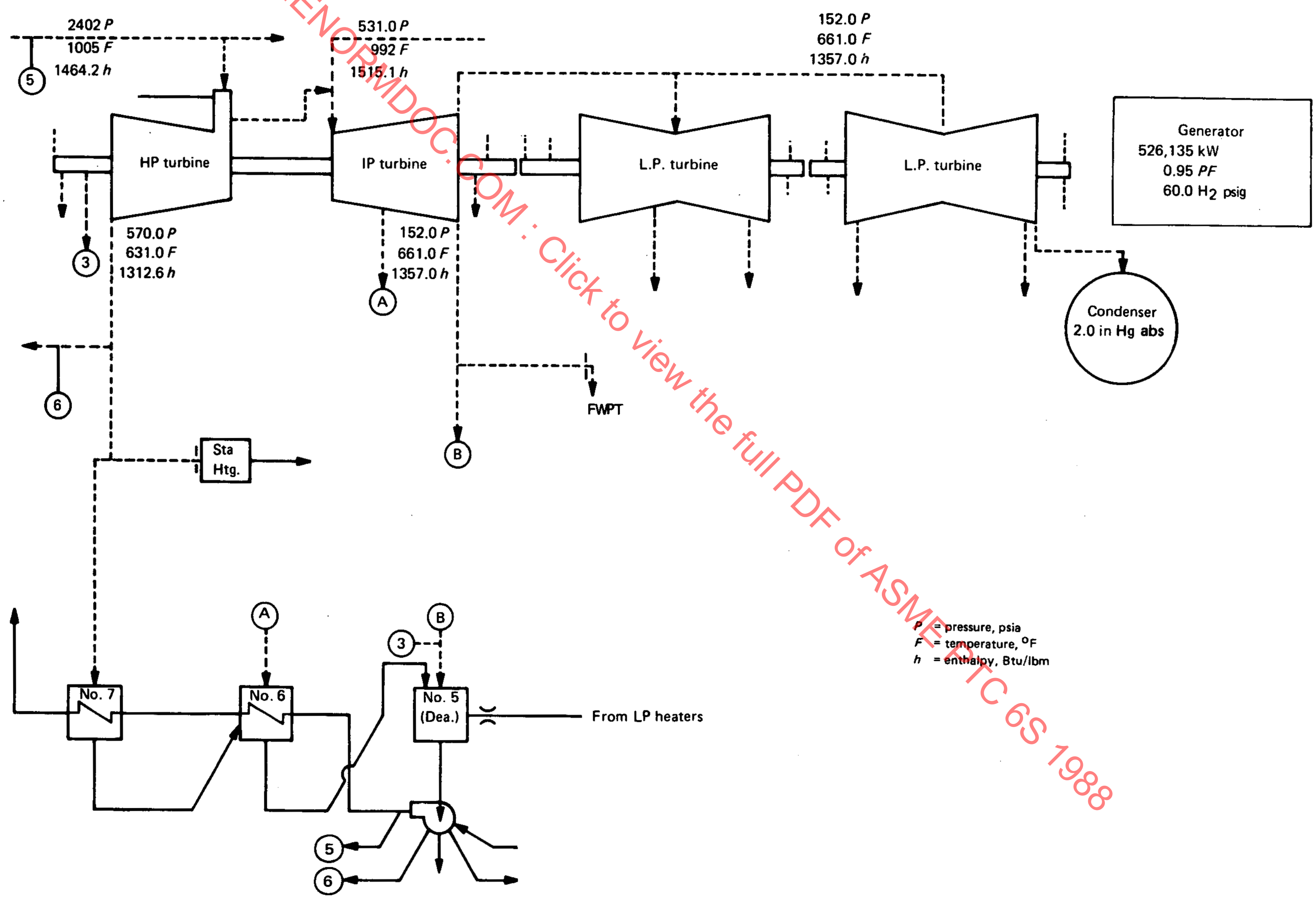
FIG. 9.2 LOSS FACTOR VERSUS CROSSOVER PRESSURE

(*b*) Terminal temperature difference (and drain-cooler approach difference) of each feedwater heater in the heat cycle can be compared periodically to indicate changes in heater performance which affect heat rate. Special temperature and pressure measurements should be read at each point of steam and water entry or exit, at valves wide open. The test should last at least 30 minutes. Two observers are usually required for this test.

(*c*) Packing leakages can be measured with secondary flow-measuring devices in accordance with para. 4.4.8 at valves-wide-open under stable conditions (see para. 6.22). Leakage tests should last a minimum of one hour. They usually require two or three observers.

9.8 SAMPLE CALCULATIONS

9.8.1 Enthalpy-Drop Efficiency



	Throttle	HP Turbine Exhaust	IP Turbine Inlet	IP Turbine Exhaust
Pressure, psia	2402	570.0	531.0	152.0
Temperature, °F	1005	631.0	992.0	661.0
Enthalpy,* Btu/lbm (actual)	1464.2	1312.6	1515.1	1357.0
Enthalpy,* Btu/lbm (isentropic)	...	1288.1	...	1344.0
Used energy, Btu/lbm	...	151.6	...	158.1
Available energy, Btu/lbm	...	176.1	...	171.1

HP turbine efficiency:

$$\eta_{hp} = \frac{h_i - h_o}{h_i - h_s} = \frac{1464.2 - 1312.6}{1464.2 - 1288.1} \times 100 = 86.1\%$$

IP turbine efficiency:

$$\eta_{ip} = \frac{h_i - h_o}{h_i - h_s} = \frac{1515.1 - 1357.0}{1515.1 - 1344.0} \times 100 = 92.4\%$$

Valve stem leakoff flows and gland leakage into turbine sections downstream in the flow path can affect the overall section efficiency. The intermediate- pressure turbine receives a mixture of steam to its first stage. See para. VI-14 of PTC 6A-1982 and para. 6.24.7 of this Report.

9.8.2 Maximum Generating Capability

Uncorrected load at 0.95 pf and 60.0 psig H_2 pressure	=	526,135 kW
Electrical losses at 0.95 pf and 60.0 psig H_2 pressure	=	6,045 kW**
Hydrogen pressure correction to 60.0 psig	=	0 kW**
Uncorrected unit output	=	526,135 kW

Group 2 LOAD CORRECTION (From manufacturer's curves)

The factors listed in the following table permit correcting the test load to specified conditions. Correction factors are defined as 1 + (% change)/100. These factors will be used as divisors when correcting from test to specified conditions.

	Conditions Specified	Conditions Test	Change	Percent Change in Load	Correction Divisor
Throttle pressure, psia, Fig 9.3	2415	2402	−0.54%	−0.54	0.9946
Throttle temperature, °F, Fig 9.4	1000	1005	+5°F	−0.05	0.9995
Reheat temperature, °F, Fig 9.5	1000	992	−8°F	−0.32	0.9968
Reheater pressure drop, %, Fig 9.6	10	6.84	−3.16%	+0.90	1.0090
Exhaust pressure, in. Hg abs, Fig 9.7	1.5	2.0	+0.5	−0.25	0.9975
Product of correction factors	...	...	...	...	0.9973

$$\text{Corrected maximum capability} = \frac{526{,}135}{0.9973} = 527{,}559 \text{ kW}$$

*Based upon the 1967 ASME Steam Tables.
**From manufacturer's curves.

9.8.3 Worth of $\Delta\eta_{hp}$% and $\Delta\eta_{ip}$% in Heat Rate

(Using an acceptance heat-rate test for turbine and generator performance data and the above enthalpy-drop test efficiencies, an estimated heat rate may be calculated using formulas from para. 9.6.4.)

Worth of $\Delta\eta_{hp}$% in heat rate

$$\Delta HR\% = \frac{\Delta\eta_{hp}\%\ (UE_{hp})\ Q_{hp}}{3412.142\ (kW_{tot})} - \frac{\Delta\eta_{hp}\%\ (UE_{hp})\ Q_{rhtr}}{HR\ (kW_{tot})}$$

From valves-wide-open acceptance test:

$\eta_{hp} = 86.4\%$
$UE_{hp} = 152.4$ Btu/lbm
$Q_{hp} = 3{,}263{,}986$ lbm/hr
$kW_{tot} = 489{,}288$
$Q_{rhtr} = 2{,}884{,}617$ lbm/hr
$HR = 7897$ Btu/kWhr

$$\Delta HR\% = \left[\frac{\dfrac{86.4 - 86.1}{86.4}\ (152.4)\ (3{,}263{,}986)}{(3412.142)\ (489{,}288)}\right] 100$$

$$- \left[\frac{\dfrac{86.4 - 86.1}{86.4}\ (152.4)\ (2{,}884{,}617)}{(7897)\ (489{,}288)}\right] 100$$

$$= (0.1035) - (0.0395)$$

$$\Delta HR\% = 0.0640$$

That is, there is a 0.06% increase in heat rate due to the loss in efficiency of the high-pressure turbine.

Worth of $\Delta\eta_{ip}$% in heat rate

$$\Delta HR\% = \Delta\eta_{ip}\% \frac{(UE_{ip})}{(UE_{rh})} (\text{loss factor}) \left[1 - \frac{(UE_{hp})\ (Q_{hp})}{3412.142\ (kW_{tot})}\right]$$

From valves-wide-open acceptance test:

$\eta_{ip} = 85.1\%$
$UE_{ip} = 142.1$ Btu/lbm

$$\frac{(100)\ (85.1 - 92.4)\ (142.1)}{(85.1)\ (512.9)} (0.53) \left[1 - \frac{(152.4)\ (3{,}263{,}986)}{(3412.142)\ (489{,}288)}\right]$$

(loss factor obtained from curve, Loss Factor vs. Crossover Pressure, in Fig. 9.2)

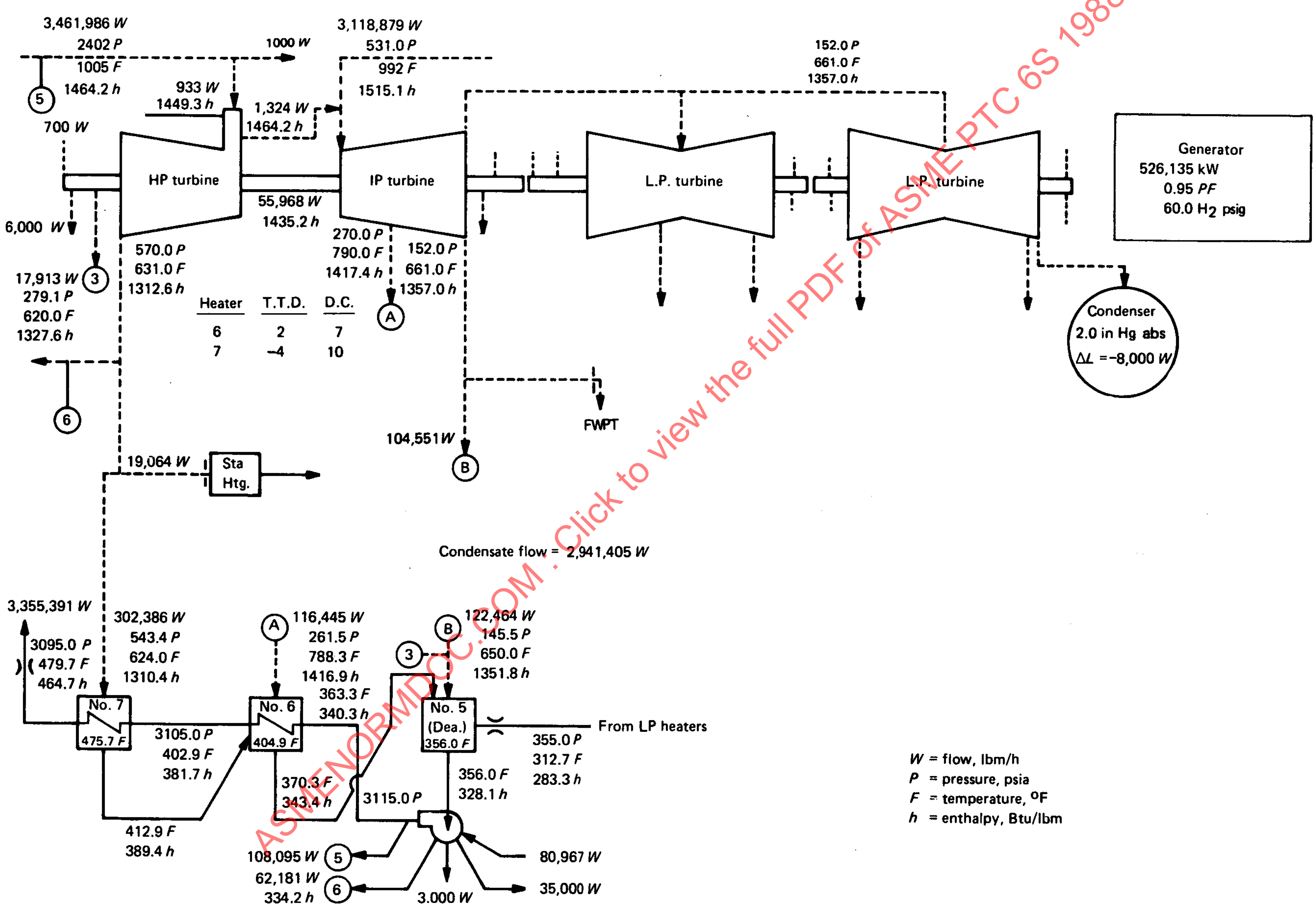
$$= -\ 1.26\ (1 - 0.2979)$$

$$\Delta HR\% = -\ 0.88$$

Similarly, there is a $-$ 0.88% change in heat rate due to the increase in efficiency of the intermediate-pressure turbine.

$$\Delta HR_{tot}\% = \Sigma \text{ changes due to } \Delta\eta_{hp} \text{ and } \Delta\eta_{ip}$$
$$= 0.06 + (-0.88)$$
$$= -0.82$$

9.8.4 Heat Rate



The heat input to the cycle is defined as:

$$w_t\,(h_t - h_{fw}) + W_r\,(h_{hrh} - h_{crh})$$

where:

w_t = throttle flow, (lbm/hr)
w_r = reheat flow, (lbm/hr)
h_t = throttle enthalpy, (Btu/lbm)
h_{fw} = final feedwater enthalpy, (Btu/lbm)
h_{hrh} = enthalpy leaving reheater, (Btu/lbm)
h_{crh} = enthalpy entering reheater, (Btu/lbm)

The net heat rate as specified for the test turbine cycle in this example is:

$$HR = \frac{(W_t - W_{shs})\,(h_t - h_{fo7}) + W_{shs}\,(h_t - h_{fi6}) + (W_r - W_{rhs})\,(h_{hrh} - h_{crh}) + W_{rhs}\,(h_{hrh} - h_{rhs})}{\text{generator output}}$$

where:

W_{shs} = superheat attemperation
W_{rhs} = reheat attemperation
h_{fo7} = enthalpy leaving No. 7 heater
h_{fi6} = enthalpy entering No. 6 heater
h_{rhs} = enthalpy of reheat attemperation

$$HR = \frac{\begin{array}{c}(3{,}460{,}986 - 108{,}095)\,(1464.2 - 464.7) + (108{,}095)\,(1464.2 - 340.3)\\ + \,(3{,}118{,}879 - 62{,}181)\,(1515.1 - 1312.6) + (62{,}181)\,(1515.1 - 334.2)\end{array}}{526{,}135}$$

$$= 7916 \text{ Btu/kWhr } (8352 \text{ kJ/kWh})$$

Group 2 HEAT RATE CORRECTIONS (From manufacturer's curves)

The factors listed in the following table permit correcting the test heat rate to specified conditions. Correction factors are defined as 1 + (% change)/100. These factors will be used as divisors when correcting from test to specified conditions.

	Conditions			Percent	
	Specified	Test	Change	Change in Heat Rate	Correction Divisor
Throttle pressure, psia, Fig 9.3	2415	2402	−0.54%	+0.02	1.0002
Throttle temperature, °F, Fig 9.4	1000	1005	+5°F	−0.08	0.9992
Reheat temperature, °F, Fig 9.5	1000	992	−8°F	+0.15	1.0015
Reheater pressure drop, %, Fig 9.6	10	6.84	−3.16%	−0.30	0.9970
Exhaust pressure, in Hg abs., Fig 9.7	1.5	2.0	+0.5	+0.25	1.0025
Product of correction factors	...	...	...	...	1.0004

$$\text{Corrected heat rate} = \frac{7916}{1.0004} = 7913 \text{ Btu/kWhr } (8349 \text{ kJ/kWh})$$

Throttle steam flow was corrected to specified conditions for the effect of deviation in initial pressure and temperature as specified in para. 5.27 of the Code.

$$w_s = w_t \sqrt{\frac{p_s \times v_t}{p_t \times v_s}}$$

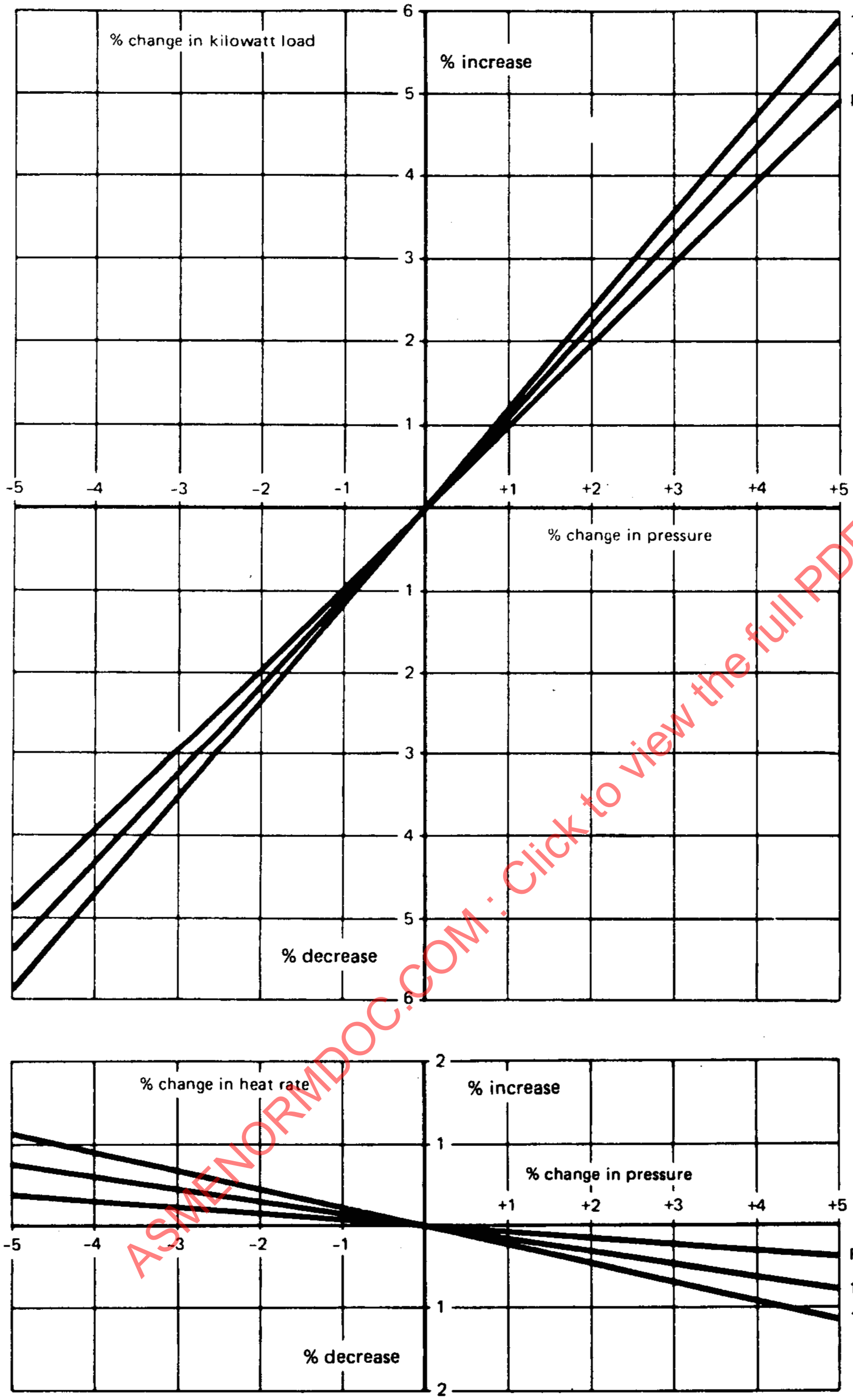
$$= 3{,}460{,}986 \sqrt{\frac{2415 \times 0.3229}{2402 \times 0.3193}}$$

$$= 3{,}489{,}712 \text{ lb}_m\text{/hr } (1{,}582{,}907 \text{ kg/h})$$

The test result can be compared to the specified curve for heat rate verses load (Fig. 9.8) or the locus of test points developed by acceptance tests.

At the corrected load of 527,559 kW (para. 9.8.2) the specified heat rate is 7915 Btu/kWhr.

The corrected test performance is 2 Btu/kWhr or 0.03% better than specified.



Method of Using Curves

These correction factors assume constant control valve opening and are to be applied to heat rates and kilowatt loads at specified steam conditions.

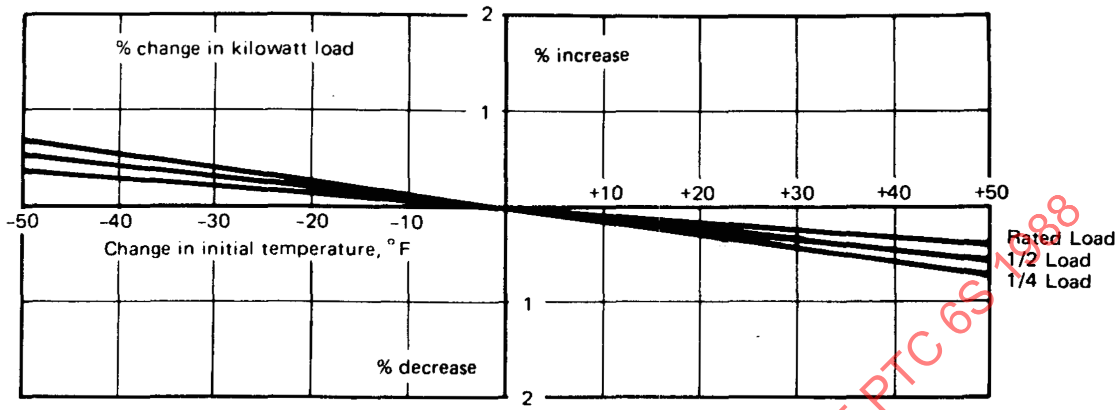
(1) The heat rate at the specified condition can be found by dividing the heat rate at test condition by the following:

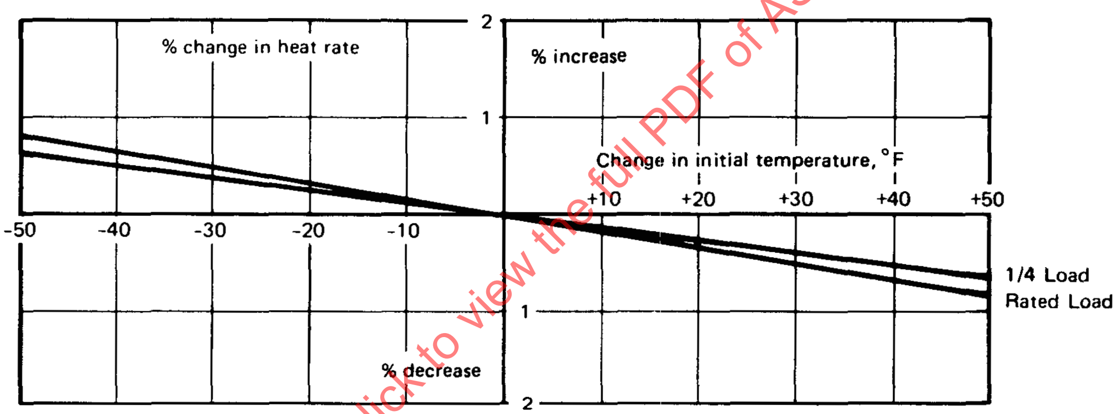
$$1 + \frac{\text{\% change in gross heat rate}}{100}$$

(2) The kilowatt load at the specified condition can be found by dividing the kilowatt load at test conditions by the following:

$$1 + \frac{\text{\% change in kW load}}{100}$$

FIG. 9.3 THROTTLE PRESSURE CORRECTION FACTORS





Method of Using Curves

These correction factors assume constant control valve opening and are to be applied to heat rates and kilowatt loads at specified steam conditions.

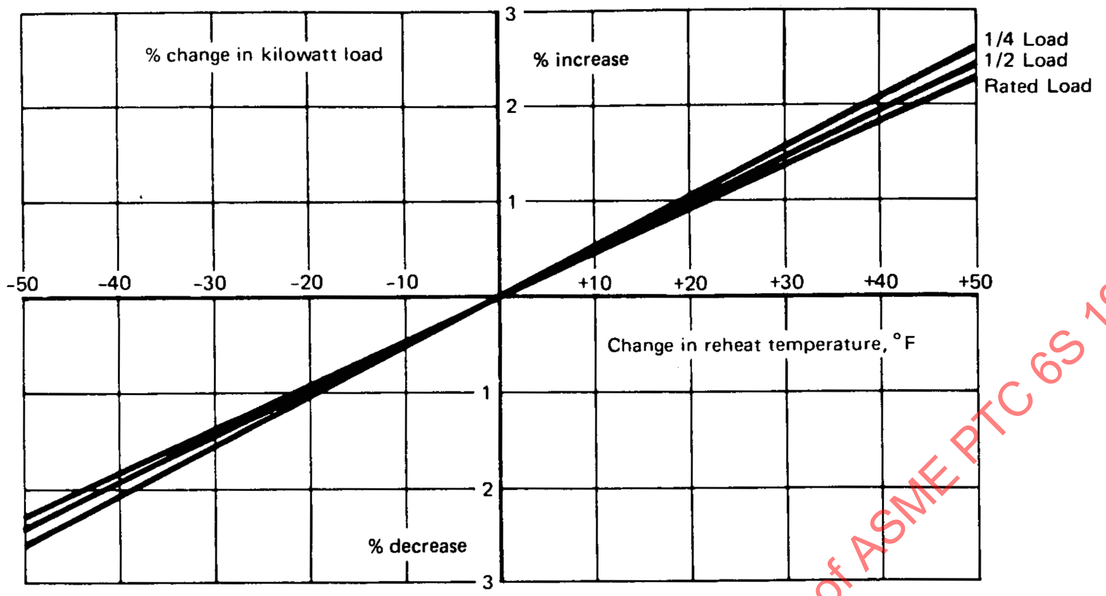
(1) The heat rate at the specified condition can be found by dividing the heat rate at test conditions by the following:

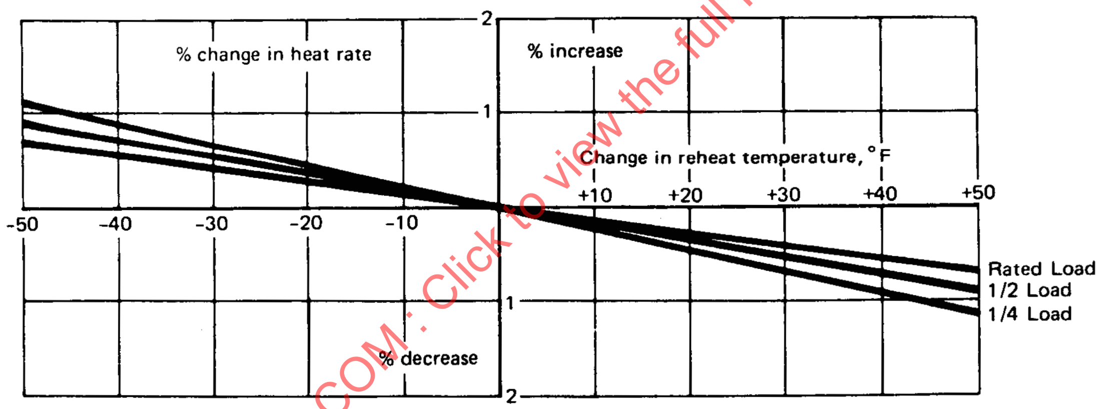
$$1 + \frac{\text{\% change in gross heat rate}}{100}$$

(2) The kilowatt load at the specified condition can be found by dividing the kilowatt load at test conditions by the following:

$$1 + \frac{\text{\% change in kW load}}{100}$$

FIG. 9.4 THROTTLE TEMPERATURE CORRECTION FACTORS





Method of Using Curves

These correction factors assume constant control valve opening and are to be applied to heat rates and kilowatt loads at specified steam conditions.

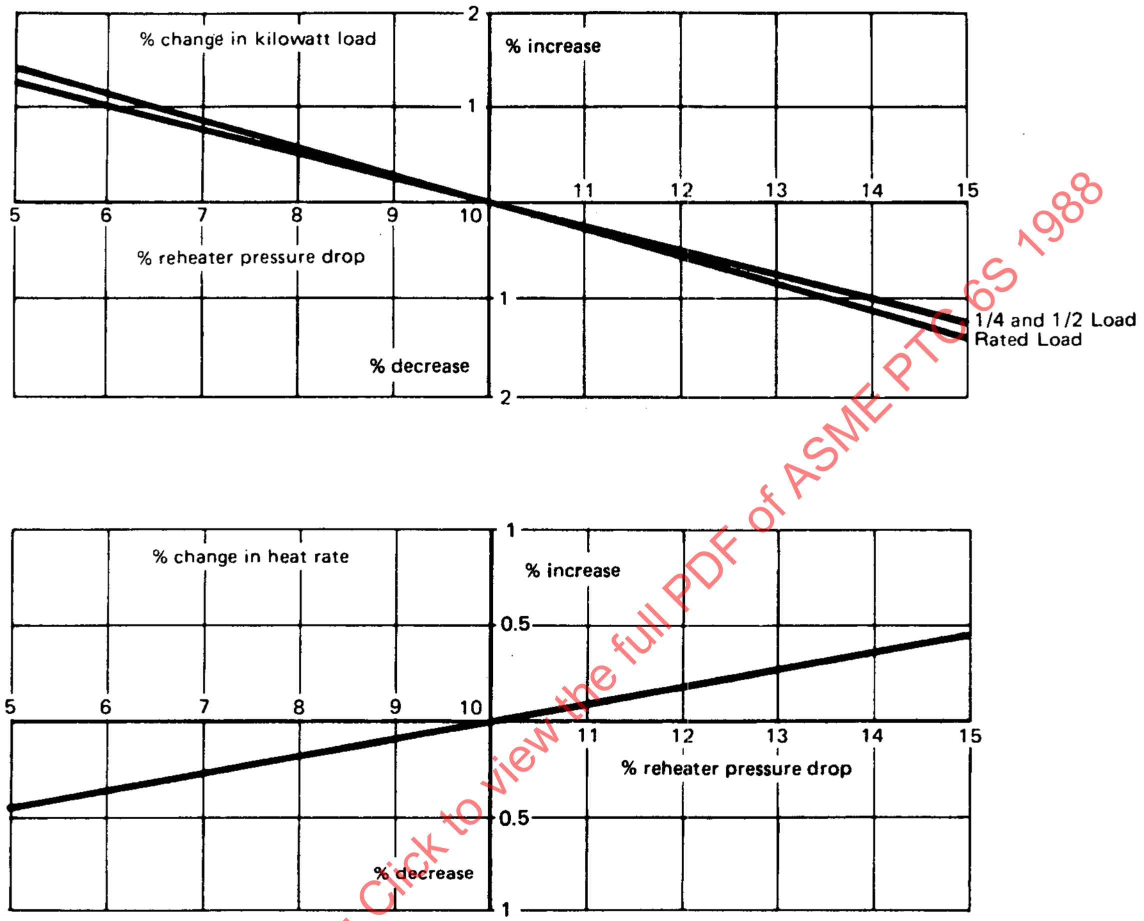
(1) The heat rate at the specified condition can be found by dividing the heat rate at test conditions by the following:

$$1 + \frac{\text{\% change in gross heat rate}}{100}$$

(2) The kilowatt load at the specified condition can be found by dividing the kilowatt load at test conditions by the following:

$$1 + \frac{\text{\% change in kW load}}{100}$$

FIG. 9.5 REHEAT TEMPERATURE CORRECTION FACTORS



Method of Using Curves

These correction factors assume constant control valve opening and are to be applied to heat rates and kilowatt loads at specified steam conditions.

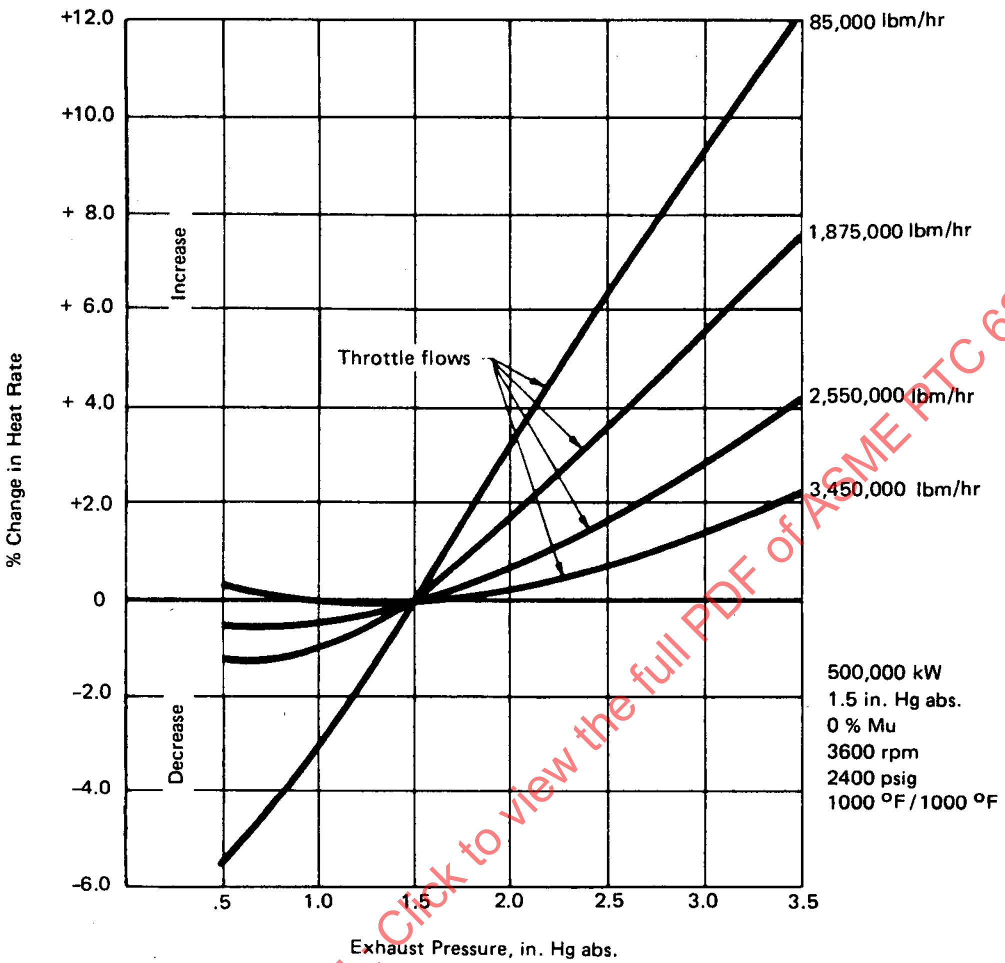
(1) The heat rate at the specified condition can be found by dividing the heat rate at test conditions by the following:

$$1 + \frac{\text{\% change in gross heat rate}}{100}$$

(2) The kilowatt load at the specified condition can be found by dividing the kilowatt load at test conditions by the following:

$$1 + \frac{\text{\% change in kW load}}{100}$$

FIG. 9.6 REHEATER PRESSURE DROP CORRECTION FACTORS



Method of Using Curves

Flows near curves are throttle flows at 2400 PSIG 1000°F. These correction factors assume constant control valve opening. Apply corrections to heat rates and kW loads at 1.5 in. Hg abs., 0% MU.

(1) The heat rate at the specified condition can be found by dividing the heat rate at test conditions by the following:

$$1 + \frac{\text{\% change in gross heat rate}}{100}$$

(2) The kilowatt load at the specified condition can be found by dividing the load at test conditions by the following:

$$1 - \frac{\text{\% change in gross heat rate}}{100}$$

FIG. 9.7 EXHAUST PRESSURE CORRECTION FACTOR

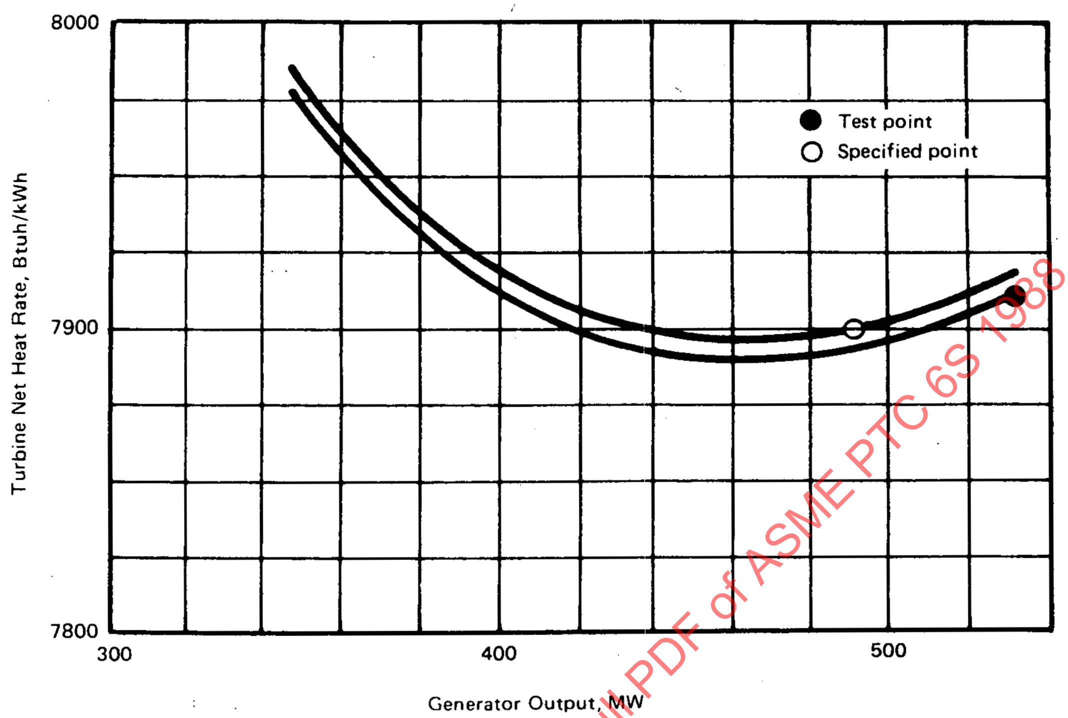


FIG. 9.8 HEAT RATE VERSUS LOAD

9.9 CALCULATION OF EXPECTED REPEATABILITY (See para. 3.8.3)

(*a*) The repeatability values given in para. 9.1 (e) were derived from uncertainty values for the selected instrumentation of para. 9.2. The derivations of these values are presented here to acquaint the user with the factors that were considered.

(*b*) Instrument uncertainties are given in Table 9.1 based upon the Report on "Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines," PTC 6 Report-1985.

9.9.1 Capability Tests

(*a*) Power measurement by one 2½-element polyphase meter and current and potential transformers of 0.3% metering accuracy class is assumed.

Overall power uncertainty is the square root of the sum of the squares of the individual uncertainties obtained from tables in PTC 6 Report-1985.

(*1*) Metering method uncertainty - from Table 4.1(c) = ±0.5 percent.

(*2*) Disc revolution uncertainty - assuming 50 disc revolutions were counted and timed. A miscount is possible, but this should be apparent by comparison with successive timings of the same run. Therefore, 0 percent certainty.

(*3*) Meter constant uncertainty - from Table 4.3 (c), ±0.25 percent.

(*4*) Potential transformer uncertainty - from Table 4.4 (b) = ±0.30 percent. The number of potential transformers required, 2, is from Table 4.1 (c). Therefore, the uncertainty is $\pm 0.30/\sqrt{2}$.

(*5*) Current transformer uncertainty from Table 4.5 (b) = ±0.10 percent. The number of current transformers required from Table 4.1 (c) is 3. Therefore, the uncertainty is $\pm 0.10/\sqrt{3}$.

(*6*) Timing uncertainty - assuming a time interval of 8 min for 50 revolutions and an increment on the clock of 1 second, the uncertainty is $\frac{1}{8 \times 60} \times 100 =$ ±0.21 percent.

Overall uncertainty =

$$\sqrt{0.50^2 + 0.25^2 + \left(\frac{0.30^2}{2}\right) + \left(\frac{0.10^2}{3}\right) + 0.21^2}$$

$= \pm 0.64\%$

TABLE 9.1
INSTRUMENT UNCERTAINTIES
(From PTC 6 Report–1985)

Instrument	Description	Uncertainty [Note (1)]
Transducer	Deadweight tester calibrated	±0.25%
Manometer (Table 4.13)	Test, precision-bored, compensated scale, without reading aid	±0.05 in.
Exhaust probes (Table 4.17)	Located at points of demonstrated accuracy, two only	±0.10 in.
Potentiometers (para. 4.29)	Precision, portable type	±0.03%
Thermocouples (Table 4.18)	Test, continuous leads, calibrated before and after test in accordance with para. 4.106 of the Code and used with ±0.03% potentiometer	±1°F
Potential transformers (Table 4.4)	Type calibration available, VA and pf of burden available, used near unity pf	±0.2%
Current transformers (Table 4.5)	Type calibration available, VA and pf of burden available	±0.1%
Watthour meters (Table 4.3)	Three-phase portable meter, temperature-controlled enclosure, without mechanical register, calibrated before test, three-phase calibration	±0.25%
Nozzle (alternate location)	Pipe-wall taps, calibrated before permanent installation, no inspection Base uncertainty, U_B = 1.25 β-ratio = 0.6, U_β = 0.20 10-D upstream length, U_{LS1} = 0.80 30 section flow straightener, U_{LS2} = 0.38 4-D downstream length, U_{DSL} = 0.67	±1.68%
Nozzle (recommended location)	Pipe-wall taps, calibrated before installation, inspected before and after test with no change in flow element Base uncertainty, U_B = 0.60 β-ratio = 0.50, U_β = 0.0 10-D upstream length, U_{LS1} = 0.40 50 section flow straightener, U_{LS2} = 0.0 4-D downstream length, U_{DSL} = 0.52	±0.89%

NOTE:
(1) See Report on "Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines," PTC 6 Report-1985.

(*b*) The effect of instrument uncertainty upon the load correction factors varies from turbine to turbine. Table 9.2 gives typical values for the reheat-regenerative turbine of the sample calculations.

(*c*) Uncertainty in corrected power is the square root of the sum of the squares of the correction-factor uncertainties and the measured-power uncertainty, or

$$\sqrt{(0.64)^2 + (0.034)^2 + (0.007)^2 + (0.033)^2 + (0.22)^2 + (0.12)^2}$$

$$= \pm 0.69\%$$

(*d*) The repeatability by para. 3.8.3 is one-half of the uncertainty or ±0.35 percent.

9.9.2 Enthalpy-Drop Tests

(*a*) Following the procedures outlined in the Guidance Report, PTC 6 Report-1985, an error analysis for enthalpy-drop tests shows that the overall uncertainty is dependent upon the uncertainty of the instrumentation, the slopes of the enthalpy-pressure and enthalpy-temperature curves, and the magnitude of the enthalpy drop in the turbine section.

(*b*) Table 9.3 gives typical values of uncertainty for both high-pressure and intermediate-pressure turbines. These values were used to vary input data to a computer program and the output information was evaluated to determine their effect upon turbine-section efficiencies. The change in efficiency in the example for each measurement uncertainty is shown in

TABLE 9.2
LOAD-CORRECTION-FACTOR UNCERTAINTIES

Measurement	Variable	Instrument Uncertainty	Instrument Range	Effect of Load-Correction Curve	Correction-Factor Uncertainty, %
Throttle	Pressure	±0.25%	3000 psi	0.0045%/psi	±0.034
	Temperature [Note (1)]	±1.0°F	...	0.01%/°F	±0.007
Hot reheat	Temperature [Note (1)]	±1.0°F	...	0.047%/°F	±0.033
Reheater Pressure Drop	Pressure	±0.25%	1000 psi	0.25%/%	±0.220
	Pressure	±0.25%	1000 psi	0.25%/%	±0.220
Exhaust	Pressure	+0.05 in.	...	1.1%/in.	±0.120
	Probes	+0.10 in.	...		

NOTE:

(1) Two thermocouples specified for each location. Since insturment uncertainty is primarily random uncertainty, the uncertainty in these measurements is inversely proportional to the square root of the number of measurements, or $1°F/\sqrt{2} = 0.7°F$.

TABLE 9.3
TYPICAL ENTHALPY-DROP UNCERTAINTY VALUES

Measurement	Variable	Sensors	Instrument Uncertainty	Instrument Range	Measurement Uncertainty	Change In Efficiency	Enthalpy-Drop Uncertainty
Throttle	Pressure	1-Trans	±0.25%	3000 psi	7.5 psi	0.23%	±0.27%
	Temperature [Note (1)]	2-TC	±1.0°F	...	0.7°F	0.19%	±0.22%
Cold reheat	Pressure	1-Trans	±0.25%	1000 psi	2.5 psi	0.29%	±0.34%
	Temperature [Note (1)]	2-TC	±1.0°F	...	0.7°F	0.22%	±0.26%
Hot reheat	Pressure	1-Trans	±0.25%	1000 psi	2.5 psi	0.14%	±0.15%
	Temperature [Note (1)]	2-TC	±1.0°F	...	0.7°F	0.08%	±0.09%
Crossover	Pressure	1-Trans	±0.25%	500 psi	1.25 psi	0.80%	±0.87%
	Temperature [Note (1)]	2-TC	±1.0°F	...	0.7°F	0.11%	±0.12%

NOTE:

(1) For multiple sensors used for the same measurement, the measurement uncertainty is the individual sensor uncertainty divided by the square root of the number of sensors used to obtain the measurement. For example, if throttle temperature is measured with two identical calibrated thermocouples with uncertainties of 1°F each, the measurement uncertainty would be $1°F/\sqrt{2}$ or 0.7°F.

Table 9.3. To determine the individual uncertainties, as shown in the final column, it is necessary to divide each change in efficiency by the base efficiency.

(c) Uncertainty in turbine section efficiency is the square root of the sum of the squares of the individual uncertainties, or, for the high-pressure turbine,

$$\sqrt{(0.27)^2 + (0.22)^2 + (0.34)^2 + (0.26)^2} = \pm 0.55\%$$

and, for the intermediate-pressure turbine

$$\sqrt{(0.15)^2 + (0.09)^2 + (0.87)^2 + (0.12^2)} = \pm 0.90\%$$

(*d*) Repeatability by para. 3.8.3 is one-half of the uncertainty, or ±0.27% for the high-pressure-turbine efficiency and ±0.45% for the intermediate-pressure-turbine efficiency.

(e) Since uncertainty is dependent upon the range of available energy and the pressure and temperature levels at the end points, Table 9.4 is presented to give guidance on the approximate repeatability levels using the recommended instrumentation for the turbine conditions shown.

TABLE 9.4
APPROXIMATE REPEATABILITY LEVELS FOR REHEAT-REGENERATIVE TURBINES, ENTHALPY-DROP-EFFICIENCY TESTS

Reheat-Regenerative Turbine Steam Conditions	Approximate Repeatability Level For High-Pressure-Turbine Section Efficiency	Approximate Repeatability Level For Intermediate-Pressure Turbine-Section Efficiency
3500 psig/1000/1000/1000°F	0.7–0.75% (VHP)	0.6–0.7%
	0.5–0.6% (HP)	...
3500 psig/1000/1000°F	0.4–0.5%	0.4–0.5%
2400 psig/1000/1000°F	0.35–0.45%	0.2–0.5%
1800 psig/1000/1000°F	0.4–0.45%	0.2–0.5%

9.9.3 Heat-Rate Tests

(*a*) The report on "Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines" presents a table of probable values of the effects of individual measurements on corrected heat rate for reheat-regenerative turbines in the approximate range of the example. Using the information from Table 9.1 of this Report and Table 5.2 from the Guidance Report, the following individual uncertainties were found:

(*b*) Overall uncertainty in corrected heat rate is the square root of the sum of the squares of the individual uncertainties, or ±1.10% for the condensate feedwater flow location and ±1.80% for the final feedwater location.

(*c*) Repeatability for the condensate flow location is ±0.55% and for the final feedwater flow location ±0.90%.

Variable	Uncertainty	Resulting Uncertainty in Correcteed Heat Rate, %
Throttle temperature	±0.7°F	±0.05
Throttle pressure	±0.25%	±0.008
Cold reheat temperature	±0.7°F	±0.03
Cold reheat pressure	±0.25%	±0.016
Hot reheat temperature	±0.7°F	±0.035
Hot reheat pressure	±0.25%	±0.02
Final feed temperature	±0.7°F	±0.084 for final feedwater flow measurement location
		±0.025 for condensate flow measurement location
Power	±0.64%	±0.64% [See para. 9.9.1 (a)]
Primary Flow	±1.68% [Note (1)]	±1.68% for final feedwater flow measurement location
	±0.89%	±0.89% for condensate flow measurement location

NOTE:
(1) See Table 9.1 for descriptive information.

SECTION 10 — TEST FOR CONDENSING TURBINE, REGENERATIVE CYCLE, WITH SATURATED INLET STEAM

10.1 INTRODUCTION

(*a*) A simplified turbine heat-rate test is recommended as the best routine check of the performance of this type of turbine. When deterioration of performance is indicated, additional measurements should be made to locate the source of this deterioration. These additional measurements include readings of all available stage pressures, gland leakages, and heater terminal temperature differences.

(*b*) Recommended test procedures, instrumentation requirements, reading frequencies, and test duration have been chosen to measure critical variables with sufficient accuracy to produce results that should be repeatable within ± 0.7 percent[1] on a day-to-day basis. (See para. 3.8.3. Refer to paras. 4.4.1 through 4.4.5 for factors affecting expected repeatability over longer periods of time.)

(*c*) The example presented in para. 10.8 is based on a nonreheat turbine; however, the same procedure applies equally to turbines utilizing live-steam reheating, and the results are also repeatable within ± 0.7 percent without the use of additional instrumentation.

(*d*) The recommended test will require an estimated minimum of four observers plus a test supervisor. If additional data beyond the minimum required for heat rate determination is desired, additional observers may be needed.

10.2 INSTRUMENTATION REQUIREMENTS

(*a*) Special test instrumentation is recommended for the following critical variables:

- Generator output
- Feedwater flow
- Throttle steam pressure
- Low-pressure turbine exhaust pressure
- Final feedwater temperature
- Throttle steam quality
- First stage pressure

The recommended instruments will measure critical variables with sufficient accuracy for test results that are repeatable to the percentage given in para. 10.1(b) if the recommendations of Section 4 for their installation and use are observed.

(*1*) *Generator Output.* See paras. 4.2.1 through 4.2.7.

(*2*) *Feedwater Flow:*
Primary element: See paras. 4.4.1 through 4.4.4.
Location of primary element: See Location 1 of Table 4.1.
Measurement of differential pressure: See paras. 4.4.6 and 4.4.7.

(*3*) *Pressure.* See para. 4.5

(*4*) *Temperature.* See paras. 4.6.1 through 4.6.4.

(*5*) *Low-Pressure Turbine Exhaust Pressure.* See paras. 4.5.8 through 4.5.10.

(*6*) *Throttle Steam Quality.* See para. 4.7. If the turbine is supplied through multiple leads, duplicate measurements should be made in each lead.

(*b*) Where live-steam reheating is employed, it is desirable to obtain also test measurements of hot-reheat steam pressures and temperatures. These measurements provide supplemental information for analyzing the cause of performance deterioration as typically described in para. 10.7. However, these measurements are not essential to performing the simplified turbine heat rate test herein described.

(*c*) *Secondary Readings.* See para. 4.10.

10.3 INSTRUMENT LOCATIONS

The required instrumentation for a simplified heat rate test is indicated on Fig. 10.1. Also shown are additional measurements which can be used to determine the causes of deterioration in performance.

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[1] See para. 10.9 for derivation of this value.

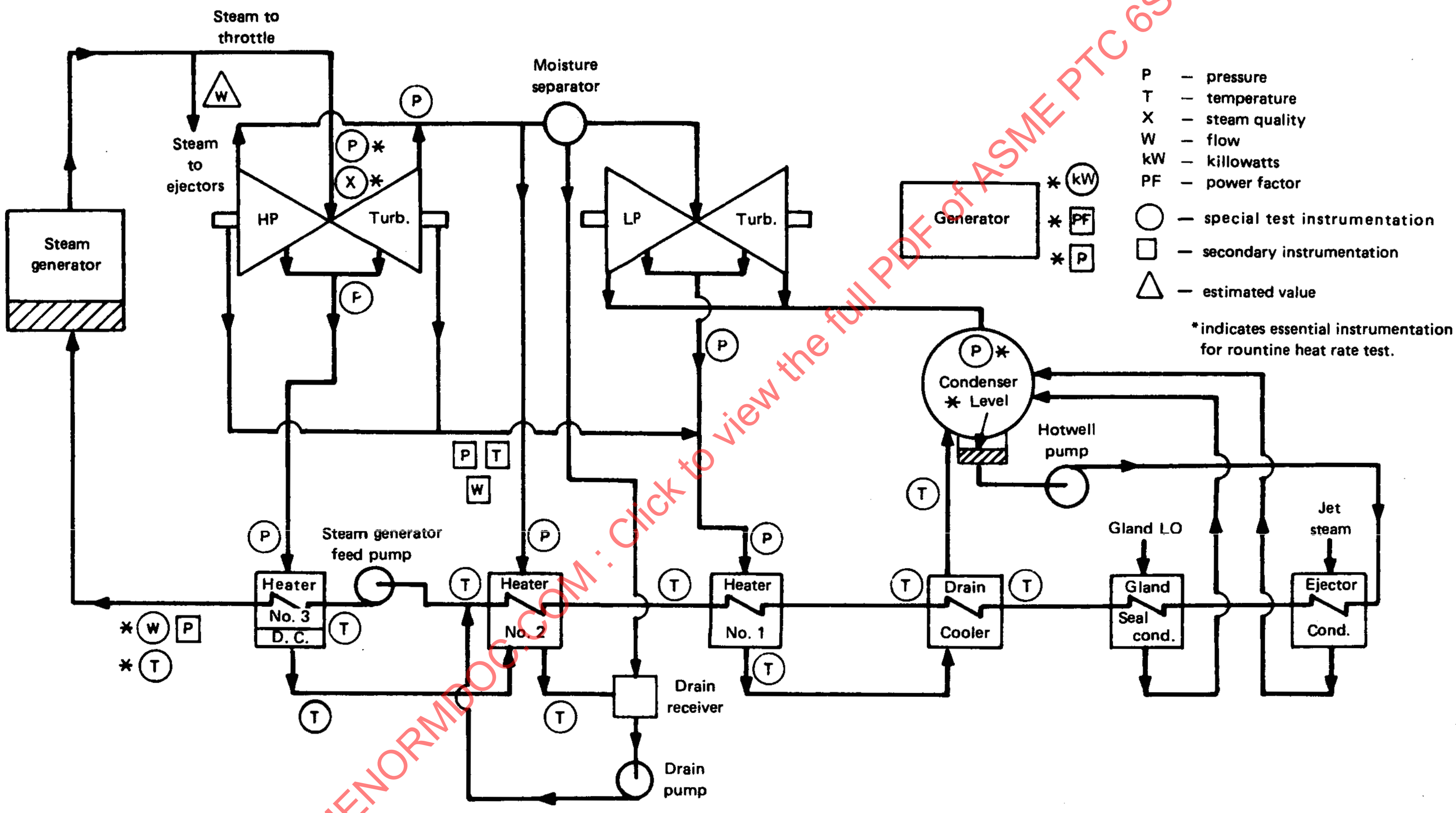

FIG. 10.1 INSTRUMENTATION FOR ROUTINE PERFORMANCE TESTS FOR CONDENSING TURBINE, REGENERATIVE CYCLE, WITH SATURATED INLET STEAM

10.4 ISOLATION PROCEDURE

See para. 3.2.

10.5 CONDUCT OF THE TEST

10.5.1 Test Conditions

(*a*) Isolate the turbine cycle in accordance with para. 3.2.

(*b*) Establish test load so that the turbine is operating at a known governing valve point (see para. 5.1) with operating conditions as close to specified conditions as possible and on load-limit control. In the case of single-valve or throttling turbines, reference measurements should be established to insure that the same percentage openings of the admission valves are established for each test. The unit should be removed from automatic system-load-control devices and made as free of system disturbances as possible.

(*c*) Allow sufficient time to assure stable operating conditions. A minimum of one-half hr stabilization time should be allowed.

10.5.2 Test Duration. Turbine heat rate tests should last two hours. Special tests to determine relative effects upon heat rate of changes to the turbine cycle, which can be performed without changing throttle flow, may last one hour.

10.5.3 Frequency and Coordination of Readings

(*a*) Readings should be coordinated by reliable time measurement in accordance with para. 4.8.

(*b*) Reading frequency should be established to produce a representative average. See para. 3.4.2 for frequency of test readings.

10.6 CALCULATION AND ANALYSIS OF TEST DATA

A typical example of calculating test results is given in para. 10.8.

10.6.1 Definition of Heat Rate. Heat rate, Btu per kWhr, is defined by the following formula with nomenclature given in Section 2 and Fig. 2.1:

$$\text{Heat rate} = \frac{w_1 h_1 - w_{11} h_{11}}{P_g}$$

10.6.2 Test Correction Factors

(*a*) Correction factors in the form of divisors shall be determined from manufacturer's data or prior tests to correct heat rate and output for deviations from specified values of throttle pressure and exhaust pressure.

(*b*) When sufficient test information is available, correction factors may be determined from data developed from prior test analysis to correct heat rate and output for deviation of feedwater heater performance, condensate subcooling in the condenser, and changes in pump-sealing and gland-cooling requirements.

10.6.3 Data Plots and Test Analysis. The presentation and interpretation of test results is discussed in Section 6.

Suggested parameters for performance monitoring of a condensing regenerative turbine supplied with saturated steam are:

(*a*) chronological plots of corrected heat rate and corrected generator output

(*b*) chronological plots of corrected stage pressures

(*c*) when a sufficient number of test results at different valve points are available, corrected heat rate may be plotted against corrected output or corrected throttle flow. A comparison of these plots would show any changes in the timing (or overlap) of the governing valves.

10.6.4 Interpretation of Trend Changes. Increasing cycle heat rate at the same valve point, after correction to specified conditions, indicates a reduction in performance of one or more components of the turbine or its cycle. The basic turbine heat rate test usually does not provide sufficient information to analyze adequately all components of the cycle, and it may be necessary to perform supplementary tests as shown in para. 10.7. Deterioration of component performance can be caused by several factors, such as:

(*a*) accumulation of turbine steam path deposits, or internal turbine damage. This is usually indicated by a change in stage pressures or stage pressure ratios. See paras. 6.24.1, 6.24.3, 6.24.4, 6.24.7, and 6.24.8.

(*b*) increases in leakage flows. (See paras. 6.24.5, 6.24.6 and 6.24.9.)

(*c*) increased feedwater-heater terminal temperature differences resulting from reduced heat transfer coefficient, internal leakage, inadequate venting or failure to control condensate level. (See para. 6.24.2.)

10.7 SUPPLEMENTARY TESTS

Supplementary tests may be required to analyze the cause of performance deterioration. These may be conducted separately or in conjunction with the simplified heat rate test. When conducted separately, stable operating conditions and the same governing valve position that existed during the test should be used as the basis of comparison.

(a) Stage pressures may be measured periodically, preferably at valves wide open. The minimum recommended test duration is 30 minutes. If steam quality differs appreciably from the specified value, additional corrections for moisture content should be made on the basis of data supplied by the manufacturer. See paras. 6.12 to 6.13.4 for corrections due to deviation of throttle pressure from specified value. A comparison of pressure ratios (inlet pressure divided by outlet pressure) for turbine sections or for groups of stages, including the governing valves, is also of value.

(b) Terminal temperature differences (and drain-cooler-approach temperature differences) of each feedwater heater in the heater cycle indicate changes in heater performance which affect heat rate results. Temperature and pressure measurements should be taken at each point of steam and water entry or exit, at the specified valve openings. Test duration should be at least 30 minutes. Usually two observers are required for this test.

(c) Packing leakages can be measured with secondary-flow measuring devices at the specified valve opening under stable conditions. Increased packing-leakage flow indicates deterioration of internal packing clearances, resulting in poorer heat rates. Leakage tests should last at least one hour. They usually require two to three observers.

(d) Phased isolation tests can be used to determine the source of steam bypassing the turbine. Two tests are conducted at constant cycle heat input, one with the cycle operating normally and the other with the cycle isolated, per para. 3.2. The isolation is accomplished in several phases and the gain in power output due to each phase is determined with accurate power measuring instrumentation.

Incremental gains in generation aid in the identification of wasted thermal energy. Steam conditions are measured and the values are used to determine correction factors from manufacturer's data or from prior tests to correct for deviation from specified values.

The instruments required follow the recommendations in para. 10.2 (a).

Each system, such as feedwater heater vents, condensate pump recirculation, etc., is isolated and the unit is allowed to stabilize for at least 30 minutes. Generation is measured to detect any increase due to that isolation phase. After full isolation has been completed, the unit is allowed to stabilize for one hour before a two hour maximum capability test is run.

The thermal output of a nuclear steam supply system must remain constant during the entire phased isolation test in order that the measured power changes are meaningful.

10.8 SAMPLE CALCULATIONS

Specified Conditions	
Throttle steam pressure, psia	464.7
Quality of throttle steam, %	99.75
Exhaust pressure, in. Hg abs.	1.0
Power factor, %	80.0
Hydrogen pressure, psig	15.0

Reference Heat Rate — 11,050 Btu/kWhr, with maximum valve opening, and with all operating conditions at the specified values.

Test Results	
Throttle steam pressure, psia	467.7
Quality of throttle steam, %	99.8
Enthalpy of throttle steam, Btu/lbm	1203.3
Exhaust pressure, in. Hg abs.	1.26
Feedwater leaving No. 3 heater:	
temperature, °F	335.7
enthalpy, Btu/lbm	306.7
Feedwater flow to steam generator, lbm/hr	1,848,000
Generator Output, kW	146,760
Power factor, %	88.0
Hydrogen pressure, psig	18.0

Changes in stored water during 2 hr test period

Condenser hotwell storage = −7200 (level fall) lbm

Steam generator storage = + 700 (level rise) lbm

Make-up = 0

$$\text{System leakage} = \frac{7200 - 700}{2} = 3250 \text{ lbm/hr}$$

(assumed to be from the steam generator)

Steam to ejector = 750 lbm/hr (estimated on basis of previous test data)

Throttle steam flow = Feedwater flow − system leakage − increase in steam generator storage − steam to ejector

$$= 1{,}848{,}000 - 3250 - \frac{700}{2} - 750$$

$$= 1{,}843{,}650 \text{ lbm/hr}$$

Electrical Losses (From manufacturer's curves)

Output	= 146,760 kW
80% power factor, losses	= −1767 kW
88% power factor, losses	= +1572 kW
Load corrected to 80% power factor	= 146,565 kW

Additional Losses With Increased Hydrogen Pressure
Hydrogen pressure was 18 psig, instead of the specified 15 psig, increasing losses by 29 kW.

Generator output corrected to specified power factor and hydrogen pressure = 146,565 + 29 = 146,594 kW

$$\text{Test heat rate} = \frac{1{,}843{,}650 \times 1203.3 - 1{,}848{,}000 \times 306.7}{146{,}594}$$

$$= 11{,}267 \text{ Btu/kWh}$$

GROUP 2 CORRECTION FACTORS (From manufacturer's curves)

	Heat Rate	Output
Throttle pressure	0.9990	1.0065
Exhaust pressure	1.0110	0.9891
Product of correction factors	1.0100	0.9955

$$\text{Corrected heat rate} = \frac{11{,}267}{1.0100} = 11{,}155 \text{ Btu/kWh}$$

$$\text{Corrected output} = \frac{146{,}594}{0.9955} = 147{,}257 \text{ kW}$$

$$\text{Heat rate above reference} = \frac{(11{,}155 - 11{,}050)}{11{,}050} \times 100$$

$$= 0.95\%$$

$$\text{Corrected throttle steam flow} = \text{test flow} \times \sqrt{\frac{p_s}{p_t} \times \frac{v_t}{v_s}}$$

$$= 1{,}843{,}650 \times \sqrt{\frac{464.7}{467.7} \times \frac{0.9905}{0.9965}}$$

$$= 1{,}832{,}190 \text{ lbm/h}$$

10.9 CALCULATION OF EXPECTED REPEATABILITY

10.9.1 Feedwater Flow Nozzle. Calibrated pipe-wall-tap nozzle in flow section meeting Code requirements. Because of location following the final feedwater heater and feedwater pump there was no inspection.

Base uncertainty	$U_B = \pm 1.25\%$
18-D upstream straight pipe	$U_{LS1} = \pm 0.0\%$
50-section flow straightener	$U_{LS2} = \pm 0.0\%$
β-ratio = 0.5	$U_\beta = \pm 0.0\%$
8-D downstream straight pipe	$U_{DLS} = \pm 0.09\%$

Uncertainty =

$$\sqrt{(U_B)^2 + (U_\beta)^2 + (U_{LS1})^2 + (U_{LS2})^2 + (U_{DLS})^2}$$

Flow uncertainty =

$$\sqrt{(1.25)^2 + (0.0)^2 + (0.0)^2 + (0.0)^2 + (0.09)^2} = \pm 1.25\%$$

10.9.2 Flow Differential. Test manometer, precision bored, compensated scale, without reading aid.

Uncertainty = ± 0.05 in. Hg
Test differential = 30.0 in. Hg

$$\text{Uncertainty} = \frac{0.05}{30.0} \times 100 = \pm 0.17\%$$

$$\text{Flow uncertainty} = \frac{\pm 0.17}{2} = \pm 0.09\%$$

10.9.3 Throttle Steam Pressure. Transducer, medium accuracy, laboratory-calibrated, 0–1000 psi range.

Uncertainty = $\pm 0.10\%$ of full scale

$$\text{Uncertainty} = \frac{\pm 0.10}{100} \times 1000 = \pm 1.0 \text{ psi}$$

Heat rate correction = 0.033%/1.0 psi
Heat rate uncertainty = $0.033 \times 1.0 = \pm 0.03\%$

The error in steam enthalpy for a 1.0 psi uncertainty in pressure is negligible.

10.9.4 Throttle Steam Quality. Uncertainty in determination of steam quality results primarily from the difficulty in obtaining a representative sample. Data to predict the possible error is not available. Design moisture contents for throttle steam are low, however, and a sizeable percentage error in moisture content results in a relatively small error in turbine heat rate. Assuming a 100% error in moisture content:

Uncertainty in moisture content = ±0.2%
Uncertainty in steam enthalpy = ±1.5 Btu/lbm
$h_{1t} - h_{11} = 1203.3 - 306.7 = 896.6$ Btu/lbm

$$\text{Uncertainty in heat rate} = \frac{1.5}{896.6} \times 100 = \pm 0.17\%$$

10.9.5 Final Feedwater Temperature. Calibrated test thermocouple with continuous leads used with ±0.05% potentiometer.

Uncertainty in reading = ±2.0 °F
Uncertainty in enthalpy = ±2.1 Btu/lbm
$h_{1t} - h_{11} = 1203.3 - 306.7 = 896.6$ Btu/lbm

$$\text{Uncertainty in heat rate} = \frac{2.1}{896.6} \times 100 = \pm 0.23\%$$

10.9.6 Exhaust Pressure. One probe located in each exhaust end at points of demonstrated accuracy.

Uncertainty in measurement = ±0.2 in. Hg
Transducer, medium accuracy, laboratory calibrated.
Uncertainty in measurement = ±0.05 in. Hg

Overall uncertainty in measurement =

$$\sqrt{(0.2)^2 + (0.05)^2} = \pm 0.21 \text{ in. Hg}$$

Heat rate correction = 0.28%/0.1 in. Hg

$$\text{Uncertainty in heat rate} = 0.28 \times \frac{0.21}{0.10} = \pm 0.59\%$$

10.9.7 Output. The type calibration curves, volt amperes and power factors of burdens were available for both potential and current transformers. A three-phase electronic watthour meter with high accuracy digital readout, calibrated before test, was used.

Uncertainties
- Potential transformer ±0.20%
- Current transformer ±0.10%
- Watthour meter ±0.15%

$$\text{Overall uncertainty} = \sqrt{(0.20)^2 + (0.10)^2 + (0.15)^2} = \pm 0.27\%$$

TABLE 10.1
SUMMARY

Measurement	Instrument Uncertainty	Heat Rate Uncertainty, %	Square of Heat Rate Uncertainty
Flow nozzle	±1.25%	±1.25	1.5625
Flow differential	±0.05 in. Hg	±0.09	0.0081
Throttle steam pressure	±1.0 psi	±0.03	0.0009
Throttle steam quality	±0.2%	±0.17	0.0289
Final feed temperature	±2.0°F	±0.23	0.0529
Exhaust pressure	±0.21 in. Hg	±0.59	0.3481
Output	±0.27%	±0.27	0.0729
		Sum of Squares =	2.0743

Overall heat rate uncertainty = $\sqrt{\text{sum of squares}} = \pm 1.44\%$

$$\text{Repeatability} = \frac{\text{uncertainty}}{2} = \frac{\pm 1.44}{2} = \pm 0.72\%$$

SECTION 11 — TEST FOR NONCONDENSING NONEXTRACTION TURBINE, WITH SUPERHEATED EXHAUST

11.1 INTRODUCTION

Noncondensing, nonextraction turbines (topping units) usually drive power auxiliaries and some process plant equipment. Tests described in this Section are intended to reveal changes in turbine performance which result from deterioration in blade performance, increased clearance of interstage seals and increased packing clearance. These tests are the internal efficiency enthalpy- drop and the maximum capability test. Packing leak-off flow should be monitored from station instrumentation to provide information on seals and packing clearances. (See also para. 11.7.)

(a) *Internal Efficiency Enthalpy-Drop.* This is the simplest and most accurate routine test. The fundamental requirements are:

(*1*) superheated steam conditions at both the throttle and exhaust end of the turbine;

(*2*) steam conditions (pressure and temperature) as close to design as possible;

(*3*) a load coordinate such as governing valve point, first-stage shell pressure, or steam flow;

(*4*) constant or a specific rpm.

(*b*) *Maximum Capability.* The capability test consists of measuring electrical or mechanical output at a particular governing valve point, preferably valves wide open, and steam pressures and temperatures necessary to apply load correction factors.

Measured electrical output must be corrected for variation from specified values of power factor and hydrogen pressure prior to application of the load correction factors. This involves calculating the test coupling kilowatts and then subtracting the electrical and mechanical losses that would occur at specified power factor and hydrogen pressure.

(c) *Test Procedures, Instrumentation, Duration, and Reading Frequency.*

These are listed below and were chosen to measure critical variables with sufficient accuracy to produce results that are estimated to be repeatable within the following percentages (see para. 3.8.3):

Test	Repeatability
Enthalpy-drop efficiency	±0.50% [Note (1)]
Maximum capability	±0.31% [Note (1)]

NOTE:
(1) See para. 11.9 for derivation of these values.

These tests require an estimated number of three observers, depending on the test set up.

11.2 INSTRUMENT REQUIREMENTS

(*a*) Precision instrumentation is recommended for primary readings for each test as follows:

Reading	Enthalpy Drop	Maximum Capability
Throttle pressure and temperature	x	x
Nozzle bowl pressure	x	x
First-stage pressure	x	x
Exhaust pressure and temperature	x	x
Turbine shaft power	...	x
Generator output	...	x
Turbine shaft speed	x	x

(*1*) *Pressure.* See para. 4.5.

(*2*) *Temperature.* See paras. 4.6.1.2, 4.6.3, and 4.6.4.

(*3*) *Generator Output.* See para. 4.2.

(*4*) *Measurements of Driven Equipment.* Precision instrumentation is necessary to measure the performance of driven equipment and should conform to para. 4.3, and to applicable sections in the Performance Test Codes for the specific equipment. For example, water-flow measurement is a primary precision requirement for a pump, and gas-flow measurement is a primary precision requirement for a compressor.

(*5*) *Shaft Speed.* See para. 4.12.

(*b*) *Station Instruments.* For secondary readings, see para. 4.10.

(*c*) *Manufacturer's Data.* See para. 4.11.

11.3 INSTRUMENT LOCATIONS

Figure 11.1 shows schematic arrangements of primary and secondary instrumentation locations for the noncondensing, nonextracting turbine. The physical configuration of the turbine exhaust piping and steam leak-off lines should be considered before locating the instrumentation to determine the exhaust steam conditions.

11.4 OUTLINE OF ISOLATION PROCEDURE

(*a*) For enthalpy-drop efficiency tests, isolation is not normally required. However, cycle conditions affecting turbine section pressure ratios should be consistent between tests.

(*b*) For maximum capability tests, isolate the turbine cycle as described in para. 3.2.

(*c*) Prior to each capability test, isolation valves should be placed in the desired position and tagged. Performance test results shall be based only on information obtained when the turbine cycle is isolated as set forth in (a) and (b) above.

11.5 CONDUCT OF THE TEST

It is generally advantageous to conduct enthalpy-drop efficiency tests at governing valve points. These particular governing valve positions can easily be repeated over a period of years, permitting periodic tests without excessive throttling loss through partially open governing valves. Corrected maximum capability tests must be run with governing valves at their maximum travel.

11.5.1 Establishing Test Load. For enthalpy-drop efficiency tests, set the turbine load at the desired valve position (see para. 5.1), and for maximum capability tests, set the governing valves at their maximum travel. Manufacturer's data or previous data must be consulted to select load points where the exhaust of the turbine is in the superheated steam region. Turbine steam conditions should be as close to specified as possible and the permissible fluctuation in reading should be within the limits of the Code. The turbine should be on load limit control and the generator or other driven equipment should be removed from automatic system-load control devices to be made free from system disturbances. For maximum capability, the turbine heat cycle shall be isolated in accordance with para. 3.2. Allow a minimum of one-half hr stabilization time.

11.5.2 Test Duration. Tests described in this Section should be of one hour duration. Less time is needed if the test conditions are stable and within permissible fluctuation limits of the Code.

Monitoring, by the test supervisor, of the kilowatt counter reading, and temperature data, will provide quick indication on test progress and whether or not the test needs to be extended.

11.5.3 Frequency and Coordination of Readings. Readings should be coordinated by reliable time measurement in accordance with para. 4.8.

Reading frequency should be established to produce a representative average. (See para. 3.4.2.)

11.6 CALCULATION OF TEST RESULTS

11.6.1 Data Preparation and Calculations. Raw data should be examined for consistency and reliability. Paragraph 3.5 should be consulted as a guide for data reduction and calculation techniques.

11.6.2 Formulas and Sample Calculations. Enthalpy-drop efficiency formulas and sample calculations are given on Fig. 5.3 and in para. 11.8.

Corrected capability sample calculations at valves wide open are given in para. 11.8.

11.6.3 Test Correction Factors. Correction factors can be determined from manufacturer's data or prior tests to correct steam rate and output for deviations from specified values of throttle pressure, throttle temperature, and exhaust pressure.

Correction factor curves supplied by the manufacturer can be verified by test, if necessary.

11.6.4 Data Plots and Test Analyses. Corrected steam rate and power output can be compared on a chronological basis by determining the deviation from acceptance test or guarantees.

Test turbine efficiencies, stage pressures, ratios of pressures across groups of stages and packing-leakage flows can be compared in a similar manner to steam rates.

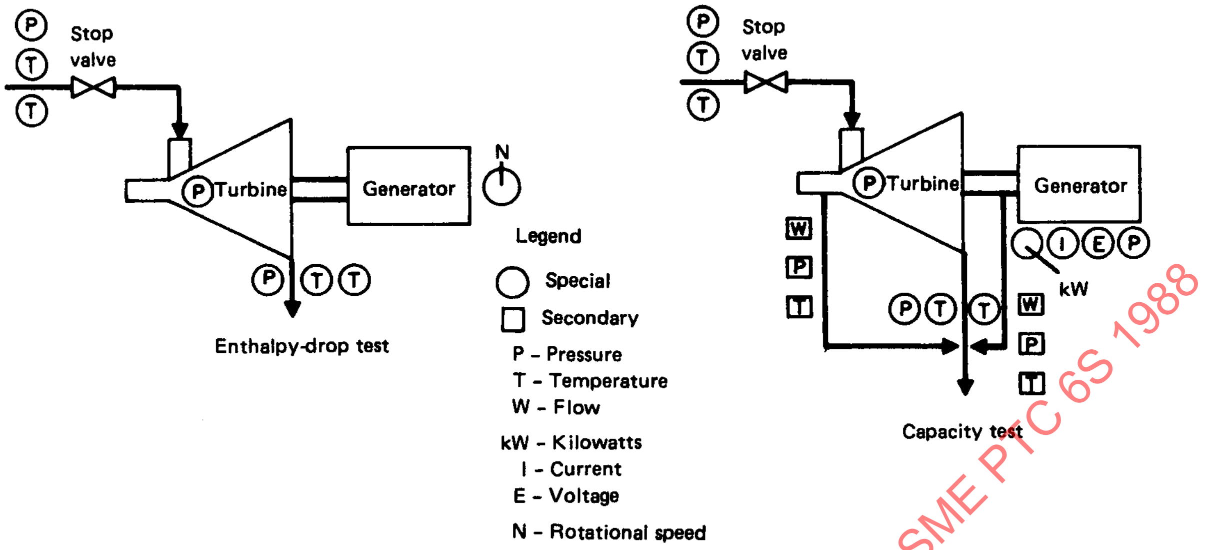

FIG. 11.1 INSTRUMENT LOCATIONS FOR NONCONDENSING, NONEXTRACTION TURBINES

11.7 SUPPLEMENTARY TESTS

Packing leakages can be measured with secondary flow measuring devices in accordance with para. 4.4.8 at valves wide open under stable conditions. The test should last a minimum of one hour and usually requires two or three observers.

Increased packing-leakage flow is indicative of deterioration of internal packing clearances resulting in poorer performance.

11.8 SAMPLE CALCULATIONS

11.8.1 Enthalpy-Drop Efficiency Test Measurements

Initial steam pressure	1247 psia
Initial steam temperature	859.4°F
Initial steam enthalpy	1414.5 Btu/lbm
Exhaust steam pressure	422.5 psia
Exhaust steam temperature	615.1°F
Exhaust steam enthalpy	1314.4 Btu/lbm
Isentropic enthalpy	1286.8 Btu/lbm

$$\text{Enthalpy-drop efficiency} = \frac{h_i - h_o}{h_i - h_s} \times 100$$

$$= \frac{1414.5 - 1314.4}{1414.5 - 1286.8} \times 100$$

$$= 78.39\%$$

11.8.2 Maximum Capability. Correct the measured generator output to the value which would have been obtained had specified values of power factor, voltage, and gas pressure been maintained during the test.

Uncorrected generator output at 0.87 pf and 14.0 psig hydrogen pressure.	18,000 kW

From manufacturer's curves

Loss with 0.87 pf	265 kW
Loss with 0.80 pf	290 kW
Loss with 15.0 psig hydrogen pressure	40 kW
Loss with 14.0 psig hydrogen pressure	36 kW

Test conditions: 265 kW + 36 kW = 301 kW
Specified conditions: 290 kW + 40 kW = 330 kW

Generator output corrected to specified generator operating conditions:

$$18{,}000 + 301 - 330 = 17{,}971 \text{ kW}$$

Gland seal data from secondary measurements:

No. 1 gland leak-off	650 lbm/h, 20 psia 713°F, 1389.9 Btu/lbm
No. 2 gland leak-off	8,000 lbm/h, 20 psia 700°F, 1383.5 Btu/lbm
Turbine mechanical losses	150 kW (From manufacturer's data)

Throttle flow from heat balance around turbine

Heat In

Throttle steam heat = 1414.5y Btu/hr, where y = throttle steam flow, lbm/hr

Heat Out

No. 1 gland leak-off = 650 lbm/hr × 1389.9 Btu/lbm = 903,435 Btu/hr

No. 2 gland leak-off = 8,000 lbm/hr × 1383.5 Btu/lbm = 11,068,000 Btu/hr

Heat equivalent of generator output = 18,000 kW × 3412.142 Btu/kWhr = 61,418,556 Btu/hr

Heat equivalent of generator losses = 301 kW × 3412.142 Btu/kWhr = 1,027,055 Btu/hr

Heat equivalent of turbine mechanical losses = 150 kW × 3412.142 Btu/kWhr = 511,821 Btu/hr

Turbine exhaust steam heat = 1314.4 (y − 8650) Btu/hr

Total heat leaving system = 1314.4 (y − 8650) Btu/hr + 74,928,867 Btu/hr

Heat in = Heat out; 1414.5y = 1314.4(y − 8650) + 74,928,867

Test throttle steam flow = y = 634,958 lbm/hr

Test steam rate = test throttle steam flow/generator output corrected to specified generator operating conditions

$$\text{Test steam rate} = \frac{634{,}958}{17{,}971} = 35.33 \text{ lbm/kWhr}$$

The flow correction factor was determined as follows:

$$\frac{w_s}{w_t} = \sqrt{\frac{p_s}{p_t} \times \frac{v_t}{v_s}} = \sqrt{\frac{1265}{1247} \times \frac{0.5750}{0.5902}} = 0.9941$$

Corrected throttle flow = 634,958 × 0.9941 = 631,212 lbm/hr (286,313 kg/h)

The steam rate correction was determined as follows:
(From manufacturer's curve)

	Specified Operating Conditions	Percent Change	Correction Divisor
Initial pressure	1265 psia	−1.42	0.986
Initial temperature	900°F	+2.4	1.024
Exhaust pressure	440 psia	+0.5	1.005
Product of correction factors			1.0147

Steam rate corrected to specified operating conditions

$$\frac{\text{Test steam rate}}{\text{Correction divisor}} = \frac{35.33}{1.0147} = 34.82 \text{ lbm/kWhr (15.79 kg/kWh)}$$

Generator output corrected to specified turbine generator operating conditions

$$\frac{631{,}212}{34.82} = 18{,}128 \text{ kW}$$

11.9 CALCULATION OF EXPECTED REPEATABILITY (See para. 3.8.3)

(*a*) The repeatability values given in para. 11.1 were derived from uncertainty values for the selected instrumentation of para. 11.2. The derivation of these values is presented in this paragraph in order to acquaint the user with the factors under consideration.

(*b*) Instrument uncertainties are based on the report "Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines," PTC 6 Report-1985 (See Table 11.4).

11.9.1 Enthalpy-Drop Efficiency Tests. The efficiency uncertainty was determined by varying each test measurement uncertainty and determining the change in efficiency, as follows:

Test Measurement	Uncertainty	Change in Efficiency	Enthalpy-Drop Uncertainty	Square of Enthalpy-Drop Uncertainty
Initial pressure	±0.1% of reading	0.12	±0.153	0.023
Initial temperature	±1°F	0.51	±0.652	0.425
Exhaust pressure	±0.1% of reading	0.11	±0.141	0.020
Exhaust temperature	±1°F	0.38	±0.741	0.549
				1.017

Overall combined uncertainty in efficiency = $\sqrt{1.017}$ = ±1.008%. Repeatability = ±1.008/2 = ±0.50%

11.9.2 Capability Tests

(*a*) *Output.* The type calibration curves, volt amperes, and power factors of burdens were available for both potential and current transformers. A three-phase switchboard watthour meter calibrated before test was used. A photoelectric counter was used to minimize the timing uncertainty.

(*b*) *Flow Correction Factor.* The flow correction factor is:

$$\frac{w_s}{w_t} = \sqrt{\frac{p_s}{p_t} \times \frac{v_t}{v_s}}$$

The uncertainty in this factor is due to instrument uncertainty in the measurement of p_t and t_t. The magnitude of this uncertainty is shown in Table 11.1.

(*c*) *Steam Correction Factor.* The steam correction factor is used to correct the steam rate for deviations of test initial pressure and temperature and exhaust pressure. It is obtained from appropriate correction curves as a function of the uncertainty in the measurements and the slopes of the curves involved. Table 11.2 gives values for the example.

(*d*) *Corrected Generator Output.* The generator output corrected to specified turbine-generator operating condition is:

$$\frac{\text{Corrected throttle steam flow}}{\text{Corrected steam rate}}$$

which reduces to:

$$\begin{array}{c} \text{Corrected} \\ \text{generator} \\ \text{output} \end{array} \times \begin{array}{c} \text{Flow} \\ \text{correction} \\ \text{factor} \end{array} \times \begin{array}{c} \text{Steam rate} \\ \text{correction} \\ \text{factor} \end{array}$$

Table 11.3 shows the uncertainty and repeatability determinations for the example.

TABLE 11.1
INSTRUMENT UNCERTAINTY

Instrument	Description	Uncertainty*
Deadweight gage	Area ratio 10:1, uncalibrated	±0.1% of reading
Test manometer	Precision-bored, compensated-scale, without reading aid	±0.05 in.
Potentiometer	Laboratory	±0.03%
Test thermocouple	Continuous leads, calibrated against secondary standard and used with ±0.03% potentiometer	±1°F
Potential transformers	Type calibration curve available, volt-amperes and power factor of burden available	±0.3%
Current transformers	Type calibration curve available, volt-amperes and power factor of burden available	±0.1%
Wattmeter	ANSI** 0.50% accuracy class, calibrated before test	±0.50%

GENERAL NOTE: For multiple sensors used for the same measurement, the measurement uncertainty is the individual sensor uncertainty divided by the square root of the number of sensors used to obtain the measurement.

*See report on "Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines," PTC 6 Report – 1985

**From ANSI C12 - 1975 and ANSI C12.10 – 1978.

TABLE 11.2
FLOW CORRECTION FACTOR UNCERTAINTY

	Measurement Uncertainty	Effect on Specific Volume	Effect on Flow Correction Factor
Initial pressure	±1.25 psi	±0.086%/psi	±0.105%
Initial temperature	±1°F	±0.106%/°F	±0.053%

TABLE 11.3
STEAM RATE UNCERTAINTY

	Measurement Uncertainty	Effect on Steam Rate Uncertainty	Steam Rate Uncertainty
Initial pressure	±1.26 psi	±0.0607%/psi	±0.076%
Initial temperature	±1°F	±0.0985%/°F	±0.099%
Exhaust pressure	±0.44 psi	±0.1900%/psi	±0.085%

TABLE 11.4
COMBINED CAPABILITY UNCERTAINTY

Test Measurement	Flow Correction Factor Uncertainty	Steam Rate Uncertainty %	Capability Uncertainty %	Square of Capability Uncertainty
Potential transformer		±0.30	±0.30	0.090
Current transformer		±0.10	±0.10	0.010
Watthour meter		±0.50	±0.50	0.250
Initial pressure	±0.105%	±0.076	±0.181	0.033
Initial temperature	±0.053%	±0.099	±0.046	0.002
Exhaust pressure		±0.085	±0.085	0.007
				0.392

Overall combined uncertainty in corrected capability = $\sqrt{0.392} = \pm 0.626\%$

Repeatability = ±0.626/2 = ±0.31%

SECTION 12 — TEST FOR NONCONDENSING EXTRACTION TURBINE

12.1 INTRODUCTION

The enthalpy-drop efficiency of turbine sections, capability, and steam-rate tests are recommended as the best routine performance tests for these turbines. When these tests detect performance deterioration, it may be necessary to conduct a complete heat-balance test to pinpoint the cause of the deterioration.

The conduct of capability and steam-rate tests presupposes the measurement of shaft output by methods giving good repeatability. Although turbines of this type are used on many applications, the majority of such machines are used for electrical generation. Therefore, this Section will cover the measurement of electrical generation as the means of determining turbine-shaft output. See applicable Performance Test Code for driven equipment for guides to determining shaft output of turbine drivers for mechanical equipment (fans, pumps, etc.), PTC 19.7 - 1980 for energy-absorbing devices, and PTC 19.13 - 1961 for rotating speed.

The conduct of capability and steam-rate tests requires running the tests without any extraction flows or with accurately repeatable measurements of extraction flows. Where turbine conditions permit, it is desirable to run one test with no extraction and one or more tests with partial or full-extraction flows to determine turbine performance over the full operating range.

(a) *The enthalpy-drop efficiency test* consists of measuring the initial and final steam temperatures and pressures and calculating an internal efficiency from the resulting enthalpies. Although the test is straightforward, measurements must be precise. Instrumentation requirements and specific test procedures are presented in this Section.

(b) *A capability test* consists of measuring electrical output at specified governing valve points, preferably valves wide open, and extraction flows along with steam pressures and temperatures necessary to determine load-correction factors. Measured electrical output must be corrected for variation from specified values of power factor and hydrogen pressure prior to application of the load-correction factors. This involves calculating the test coupling kilowatts and then subtracting the electrical and mechanical losses that would occur at specified power factor and hydrogen pressure.

(c) *A steam-rate test* consists of accurate measurement of throttle steam flow and generator output at given valve points and extraction steam-flow rates, along with steam pressures and temperatures necessary to determine load-correction factors.

(d) *A heat-balance test* consists of a complete turbine heat balance. This test requires precision measurement of all steam flows, all temperatures and pressures, and electrical output.

(e) *Test procedures, instrumentation, and duration* are listed below and were chosen to measure critical variables with sufficient accuracy to produce results that are estimated to be repeatable within the following percentages (see para. 3.8.3):

Type Test	Repeatability[2]
Enthalpy-drop efficiency	±0.6 to ±0.7% depending on enthalpy ranges
Capability	±0.5%
Steam rate	
No extraction flow nozzle[1] for throttle steam flow	±1.8%
Extraction flow nozzle[1] for throttle steam flow (orifice for extraction flow)	±2.4%

NOTES:

(1) Because of the importance of the flow measurement in this test, this value is based on short-time repeatability (day-to-day) only. For expected repeatability over longer periods of time, information given in paras. 4.4.1 through 4.4.5 should be considered. See also report on "Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines," PTC 6 Report-1985.

(2) See Tables 12.1, 12.2, 12.3, and 12.4 for calculation of repeatability.

TABLE 12.1
REPEATABILITY CALCULATIONS FOR ENTHALPY-DROP EFFICIENCY

Measurement	Guidance Reference [Note (1)] Table	Line	Uncertainty [Note (1)]	Change In Efficiency	Enthalpy-Drop Uncertainty, %
Throttle pressure	4.14	2(b)	±2.5 psi	±0.30	±0.39
-temperature	4.18	3	±1.4 °F [Note (2)]	±0.62	±0.82
Extraction					
For HP section pressure	4.14	2(b)	±1.25 psi	±0.36	±0.47
-temperature	4.18	3	±1.4 °F [Note (2)]	±0.69	±0.91
For LP section pressure	4.14	2(b)	±1.25 psi	±0.19	±0.33
-temperature	4.18	3	±1.4 °F [Note (2)]	±0.41	±0.70
Exhaust pressure	4.14	2(b)	±0.25 psi	±0.16	±0.24
-temperature	4.18	3	±1.4 °F [Note (2)]	±0.45	±0.77

Uncertainty in HP section

$$U = \sqrt{(0.39)^2 + (0.82)^2 + (0.47)^2 + (0.91)^2} = \pm 1.37\% \qquad \text{Repeatability} = \pm 0.69\%$$

Uncertainty in LP section

$$U = \sqrt{(0.33)^2 + (0.70)^2 + (0.24)^2 + (0.77)^2} = \pm 1.11\% \qquad \text{Repeatability} = \pm 0.56\%$$

NOTES:
(1) See "Guidance For Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines," PTC 6 Report-1985.
(2) Use two thermocouples to reduce error. For multiple sensors the uncertainty is the individual sensor uncertainty divided by the square root of the number of sensors. Thus $2°F/\sqrt{2} = 1.4°F$.

(*f*) The recommended tests require an estimated number of three observers. Heat-balance tests require a minimum of five observers, plus a test supervisor. Supplementary data and personnel training are two reasons for the higher number of personnel requirements.

12.2 INSTRUMENTATION REQUIREMENTS

(a) Precision instrumentation is recommended for each test as follows:

Reading	Enthalpy-Drop	Capability	Steam Rate
Throttle pressure	x	x	x
Throttle temperature	x	x	x
First-stage pressure	x	x	x
Throttle flow	...	x	x
Extraction pressure	x	x	x
Extraction temperature	x	...	...
Exhaust pressure	x	...	...
Exhaust temperature	x	...	...
Extraction flow and/or exhaust flow	...	x	x
Generator output	...	x	x

(*1*) *Pressure*. See para. 4.5.
Instrumentation varies with repeatability requirements.
Deadweight gage, 100:1 area ratio, uncalibrated.
Transducer, deadweight tester calibrated.
For pressure differential across flow device, use station manometer without reading aid.

(*2*) *Temperature*. See para. 4.6.
Instrumentation varies with repeatability requirements.
Test thermocouple made with continuous leads, calibrated against secondary standard, and used with ±0.03 percent potentiometer. Use two thermocouples at each primary measurement point.
Station recording thermocouple of standard wire, uncalibrated, and used with ±0.30 percent recording potentiometer.

(*3*) *Flow*

(a) *Primary-Flow Measurement*. For installations having a steam generator and turbine operating essentially isolated from other units, feedwater flow provides the most satisfactory measurement of throttle flow, since a calibrated primary element can be used.

TABLE 12.2
REPEATABILITY CALCULATIONS FOR CAPABILITY TEST

Measurement	Guidance Reference [Note (1)] Table	Line	Uncertainty	Capability Uncertainty, %
Throttle pressure	4.15	2	±2%	±0.19
-temperature	4.18	6	±10 °F	±0.28
Extraction pressure	4.15	2	±2%	±0.27
Exhaust pressure	4.15	2	±2%	±0.12
Turbine factor uncertainties	$U = \sqrt{(0.19)^2 + (0.28)^2 + (0.27)^2 + (0.12)^2} = \pm 0.45\%$			
Generator output				
Watthour meter Three-phase calibration	4.3	d	±0.50%	±0.50
Potential transformers	4.4	c	±0.3%/ 2	±0.21
Current transformers	4.4	c	±0.3%/ 2	±0.21
Disc turns timing	Reading		1 sec/180 sec	±0.58
Disc turns counting	Reading		[Note (2)]	0
Generator output uncertainty	$U = \sqrt{(0.50)^2 + (0.21)^2 + (0.21)^2 + (0.58)^2} = \pm 0.82\%$			
Overall capacity uncertainty	$U = \sqrt{(0.45)^2 + (0.82)^2} = \pm 0.94\%$		Repeatability = ±0.5%	

NOTES:
(1) See "Guidance For Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines," PTC 6 Report – 1985.
(2) Data reduction should establish and eliminate miscounting errors.

For headered steam generator installations, it is necessary to measure extraction or exhaust-steam flow, and therefore repeatability is calculated on steam-flow measurements. Each steam-flow measurement decreases the repeatability.

(b) *Feedwater Flow*. Feedwater-flow measurement should be based on the principles set forth in paras. 4.4.1 through 4.4.7.

(c) *Steam Flow*. The requirements for accurate measurement of primary steam flow are the same as for water-flow measurements. (See paras. 4.4.1, 4.4.3, and 4.4.4).

See para. 4.4.5 for a discussion on steam-flow measurement accuracy.

(*i*) In passing through the flow-measuring device, the steam shall have not less than 25°F superheat in the throat to assure dry steam.

(*ii*) The specific weight of steam shall be calculated from an enthalpy obtained from accurate temperature and pressure measurements 10 pipe diameters downstream of the primary element and a pressure measurement one pipe diameter upstream of the primary element.

(*iii*) In some installations it may be necessary to use orifices rather than flow nozzles. While orifices may not have the absolute accuracy of flow nozzles, they produce good repeatability over a long period of time.

Refer to the Report "Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines" for assistance in determining the accuracy of the installation. Before installation, the primary-flow element should be inspected, checked for size, and calibrated, where possible. Calibration should be preferably at the same temperature and pressure as the test conditions. The primary-flow element shall be located immediately upstream of the turbine throttle valve, but with sufficient pipe-diameter

TABLE 12.3
REPEATABILITY CALCULATIONS FOR STEAM-RATE TEST — NO EXTRACTION

Measurement	Guidance Reference [Note (1)] Table	Guidance Reference [Note (1)] Line	Uncertainty [Note (1)]	Steam-Rate Uncertainty, %
Throttle conditions				
Pressure	4.15	2	±2%	±0.19
Temperature	4.19	3	±10 °F	±0.28
Flow differential	4.13	3	±0.1 in.	±0.23
Nozzle factors		No Extraction		
U_B	4.10	Item G [Note (2)]		±3.0
U_β	Fig. 4.6	@ β = 0.6569 uncalibrated curve		±0.5
U_{LS1}	Fig. 4.7	@ 10 and β = 0.6569 =		±0.9
U_{LS2}	Fig. 4.8	@ 30 and β = 0.6569 =		±0.5
U_{DSL}	Fig. 4.9	Ratio = 3 =		±0.0
Throttle flow uncertainty $U = \sqrt{(0.19)^2 + (0.28)^2 + (0.23)^2 + (3.0)^2 + (0.5)^2 + (0.9)^2 + (0.5)^2 + (0)^2} = \sqrt{10.477} = \pm 3.24$				
Generator output				
Watthour meter Single-phase calibration	4.3	d	±1.0%	±1.0
Potential transformers	4.4	d	$\pm 1.5\%/\sqrt{2}$	±1.06
Current transformers	4.5	c	$\pm 0.3\%/\sqrt{2}$	±0.21
Disc turns timing	Reading		1 sec/180 sec	±0.58
Disc turns counting	Reading		[Note (3)]	0
Generator output uncertainty $U = \sqrt{(1.0)^2 + (1.06)^2 + (0.21)^2 + (0.58)^2 + (0)^2} = \sqrt{2.504} = \pm 1.58\%$				

Overall steam rate uncertainty — No Extraction

$$U = \sqrt{(3.24)^2 + (1.58)^2} = \sqrt{12.994} = \pm 3.60\% \qquad \text{Repeatability} = \pm 3.60/2 = \pm 1.80\%$$

NOTES:

(1) See "Guidance For Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines," PTC 6 Report – 1985.

(2) Line G, used as nozzle, had been inspected during normal overhauls. Guidance figures are for absolute accuracy, while their use in this Report is for repeatability. Major variations in accuracy would have been discovered and means developed to correct actual readings to desired accuracy. These calculations should be applied to actual readings for desired accuracy.

(3) Data reduction should establish and eliminate miscounting errors.

spacing and flow straighteners to obtain the desired accuracy. For these tests, good repeatability is of greater importance than absolute accuracy.

In determining repeatability figures, it was assumed throttle steam flow was measured with a flow nozzle and extraction steam flow with an orifice. As indicated by the repeatability differences, the use of extraction during tests decreases both accuracy and repeatability.

(4) Generator Output. See paras. 4.2.2 and 4.2.8. Three-phase switchboard meter, calibrated before test, is satisfactory. Repeatability based on three-phase calibration for capability test and single-phase calibration for steam-rate test.

(5) Secondary Flows. See para. 4.4.8.

(6) Leakage and Storage Changes. See para. 4.9.

(b) Station instruments can be used for secondary readings which have a minor effect upon results provided they are properly maintained and calibrated at regular intervals. Refer to para. 4.10.

(c) For use of previous test and manufacturer's data, refer to para. 4.11.