ASME PTC 10-2022

[Revision of ASME PTC 10-1997 (R2014)]

Axial and Centrifugal Compressors

Performance Test Codes

AN AMERICAN NATIONAL STANDARD

The American Society of Mechanical Engineers

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NOTICE

All ASME Performance Test Codes (PTCs) shall adhere to the requirements of ASME PTC 1, General Instructions. It is expected that the Code user is fully cognizant of the requirements of ASME PTC 1 and has read them before applying ASME PTCs.

ASME PTCs provide unbiased test methods for both the equipment supplier and the users of the equipment or systems. The Codes are developed by balanced committees representing all concerned interests and specify procedures, instrumentation, equipment-operating requirements, calculation methods, and uncertainty analysis. Parties to the test can reference an ASME PTC confident that it represents the highest level of accuracy consistent with the best engineering knowledge and standard practice available, taking into account test costs and the value of information obtained from testing. Precision and reliability of test results shall also underlie all considerations in the development of an ASME PTC, consistent with economic considerations as judged appropriate by each technical committee under the jurisdiction of the ASME Board on Standardization and Testing.

When tests are run in accordance with a Code, the test results, without adjustment for uncertainty, yield the best available indication of the actual performance of the tested equipment. Parties to the test shall ensure that the test is objective and transparent. All parties to the test shall be aware of the goals of the test, technical limitations, challenges, and compromises that shall be considered when designing, executing, and reporting a test under the ASME PTC guidelines.

ASME PTCs do not specify means to compare test results to contractual guarantees. Therefore, the parties to a commercial test should agree before starting the test, and preferably before signing the contract, on the method to be used for comparing the test results to the contractual guarantees. It is beyond the scope of any ASME PTC to determine or interpret how such comparisons shall be made.

FOREWORD

Revisions to test codes are inevitable in an effort to incorporate new technology and lessons learned. As is typical with this type of revision and update, committee members working to achieve an improved code have employed the previous code editions extensively throughout their professional careers. This leads to modifications, additions, and deletions to the code based on firsthand experience. However, the same ultimate goal sought by committees that rewrote previous editions remains. For the ASME PTC 10 Committee, that goal is to provide the best possible guidance and a set of rules to ensure that a compressor tested according to the Code reveals its true performance capabilities that will manifest when applied in the field. It is important to note that an acceptable ASME PTC 10 test simply means the results were obtained with adherence to Code requirements developed via dimensional analysis. The Code does not have rules concerning compressor acceptability to meet project guarantees agreed between vendors and users.

Historic and developing technical literature for compressor performance abounds with recommended changes and improvements to ASME PTC 10. These cover such technical areas as numerical solution algorithms, advances in equation of state accuracy and access, instrumentation, testing logic, and compressor hardware. The committee has reviewed and debated many proposed technical advances and judiciously applied sound engineering judgment tempered by collective experience, varied professional backgrounds, and a healthy dose of guidance from ASME.

Three major changes that previous Code users will notice are worthy of mention. These are

(a) ideal gas considerations replaced by real gas methods

(b) performance calculations based on the Schultz method replaced by an option to select from three polytropic computational methods

(c) replacement and expansion of Reynolds number correction calculations

The first change is driven by industry's need to address fluid conditions well in excess of pressures and temperatures that previously were thought of as being in the near-ideal gas behavior region of a fluid's phase diagram. Nonideal fluid behavior has been a concern and historically has presented numerous discrepancies between predicted and measured compressor performance. These issues are much better understood today, and this revised Code requires all fluids be treated as real rather than ideal. Vast expansion in availability and access to reference-quality fluid equations of state assists in easing this transition.

The second major change has been driven by peer-reviewed published documentation showing the magnitude of relative differences introduced by various polytropic work computational methods. The polytropic methods previously embraced by the Code served analysts well for many years. The historical origins of those methods date back to the 1860s but they were thoroughly documented and expanded by Schultz (1962). Schultz's methods served as the basis for incorporation into ASME PTC 10-1965 and were retained in the 1997 edition. For this revised edition, the use of pv^n to develop a closed-form solution for the integral of polytropic work (vdp) has been abandoned in favor of methods with lower uncertainty over a broad range of fluid conditions. The three methods included in the current Code range from simple to complex with uncertainty decreasing with complexity. Computerized numerical tools render straightforward implementation of any of the three methods. The committee has expended a great amount of energy and resources in determining the methods to include in the Code and proving their ability to yield accurate results over a wide range of fluids and conditions. In addition, the isentropic relations that were included in previous editions of the Code have been deleted. While the isentropic model was necessary to calculate the Schultz correction factor that was then applied to modify the polytropic calculation results, it was also used in some cases to provide alternate isentropic model performance calculations.

The third major change is the replacement and expansion of the Reynolds number correction methods. In actual practice, most vendors and users have replaced the methods adapted from Wiesner and included in ASME PTC 10-1997. Type 2 tests performed according to this Code now apply the International Compressed Air and Allied Machinery Committee (ICAAMC) 1987 method for this subject. Nondimensional test results for polytropic efficiency, polytropic work coefficient, work input coefficient, and flow coefficient will all be corrected accordingly.

While it is not possible to describe all the changes incorporated into this revised Code in this Foreword, users should be aware that many additions, improvements, deletions, and changes have been made. Embracing the resulting Code will provide compressor performance analysts with accurate methods and testing guidance. Several appendices have been provided that will assist in explanations and illustrate sample calculations. Nonmandatory Appendix D, References, has been greatly expanded to provide users with a resource listing that will augment their own individual study of compressor performance.

This Code is available for public review on a continuing basis. This provides an opportunity for additional input from industry, academia, regulatory agencies, and the public-at-large.

ASME PTC 10-2022 was approved by the PTC Standards Committee on September 27, 2022, and was approved as an American National Standard by the American National Standards Institute (ANSI) Board of Standards Review on December 12, 2022.

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Performance Test Codes

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Revisions and Errata. The committee processes revisions to this Code on a continuous basis to incorporate changes that appear necessary or desirable as demonstrated by the experience gained from the application of the Code. Approved revisions will be published in the next edition of the Code.

In addition, the committee may post errata on the committee web page. Errata become effective on the date posted. Users can register on the committee web page to receive e-mail notifications of posted errata.

This Code is always open for comment, and the committee welcomes proposals for revisions. Such proposals should be as specific as possible, citing the paragraph number(s), the proposed wording, and a detailed description of the reasons for the proposal, including any pertinent background information and supporting documentation.

Cases

(a) The most common applications for cases are

(1) to permit early implementation of a revision based on an urgent need

(2) to provide alternative requirements

(3) to allow users to gain experience with alternative or potential additional requirements prior to incorporation directly into the Code

(4) to permit the use of a new material or process

(b) Users are cautioned that not all jurisdictions or owners automatically accept cases. Cases are not to be considered as approving, recommending, certifying, or endorsing any proprietary or specific design, or as limiting in any way the freedom of manufacturers, constructors, or owners to choose any method of design or any form of construction that conforms to the Code.

(c) A proposed case shall be written as a question and reply in the same format as existing cases. The proposal shall also include the following information:

(1) a statement of need and background information

(2) the urgency of the case (e.g., the case concerns a project that is underway or imminent)

(3) the Code and the paragraph, figure, or table number(s)

(4) the edition(s) of the Code to which the proposed case applies

(d) A case is effective for use when the public review process has been completed and it is approved by the cognizant supervisory board. Approved cases are posted on the committee web page.

Interpretations. Upon request, the committee will issue an interpretation of any requirement of this Code. An interpretation can be issued only in response to a request submitted through the online Interpretation Submittal Form at https://go.asme.org/InterpretationRequest. Upon submitting the form, the inquirer will receive an automatic e-mail confirming receipt.

ASME does not act as a consultant for specific engineering problems or for the general application or understanding of the Code requirements. If, based on the information submitted, it is the opinion of the committee that the inquirer should seek assistance, the request will be returned with the recommendation that such assistance be obtained. Inquirers can track the status of their requests at https://go.asme.org/Interpretations.

ASME procedures provide for reconsideration of any interpretation when or if additional information that might affect an interpretation is available. Further, persons aggrieved by an interpretation may appeal to the cognizant ASME committee or subcommittee. ASME does not "approve," "certify," "rate," or "endorse" any item, construction, proprietary device, or activity.

Interpretations are published in the ASME Interpretations Database at https://go.asme.org/Interpretations as they are issued.

Committee Meetings. The PTC Standards Committee regularly holds meetings that are open to the public. Persons wishing to attend any meeting should contact the secretary of the committee. Information on future committee meetings can be found on the committee web page at https://go.asme.org/PTCcommittee.

Section 1
Object and Scope

1-1 OBJECT

The object of this Code is to provide a test procedure to determine the thermodynamic performance of an axial or centrifugal compressor doing work on a gas of known or measurable properties under specified conditions.

This Code is written to provide a test procedure, which will yield the highest level of accuracy consistent with the best engineering knowledge and practice currently available. Nonetheless, no single universal value of the uncertainty is, or should be, expected to apply to every test. The uncertainty associated with any individual ASME PTC 10 test will depend on practical choices made in terms of instrumentation and methodology. Rules are provided to estimate the uncertainty for individual tests.

The expectation of the Code is that a compressor performance test will be executed in a shop or factory environment to provide for the special instrumentation, calibration requirements, meter run designs, and other controlled test conditions needed by the Code. This Code may be applied to the extent that its requirements are satisfied elsewhere, such as at a user's site or field installation.

An important assumption of this Code is that the performance of a compressor may be determined either by testing at conditions that are close to those that are specified, including gas composition, pressures, and temperatures, or by testing at alternative conditions that preserve key design parameters of the compressor. Such alternative conditions may allow the test to be conducted with a suitable test gas, at suitable test pressures and temperatures, at a suitable test speed and a flow rate that preserve similitude between the specified conditions and the scaled test conditions. These alternative conditions require that the ratio of the inlet specific volume versus the discharge specific volume at test conditions and at specified conditions is within permissible tolerance. These alternative conditions also require that the nondimensional flow rate (i.e., the flow coefficient) at the test conditions and at specified conditions is within a permissible tolerance. By maintaining similitude between test conditions and specified conditions, the Code assumes that the results of the test, including flow rate, work, and efficiency, can be converted from test conditions to specified conditions.

1-2 SCOPE

1-2.1 General

The scope of this Code includes instructions on test arrangement and instrumentation, test procedure, and methods for evaluation and reporting of final results.

This Code provides rules for establishing the following quantities, corrected as necessary to represent expected performance under specified operating conditions with the specified gas:

(a) quantity of gas delivered
(b) pressure rise produced
(c) volume reduction ratio
(d) polytropic work
(e) shaft power required
(f) polytropic efficiency
(g) surge point
(h) choke point

Other than providing methods for calculating mechanical power losses, this Code does not cover rotor dynamics or other mechanical performance parameters.

1-2.2 Compressor Arrangements

This Code is designed to allow the testing of single- or multiple-casing axial or centrifugal compressors or combinations thereof, with one or more sections per casing and with sidestreams.

Compressors, as the name implies, are usually intended to produce considerable density change as a result of the compression process. Fans are normally considered to be air- or gas-moving devices and are characterized by minimal density change.

The methods of ASME PTC 10, which provide for the pronounced effects of density and pressure change during compression, have no theoretical lower limit. However, practical considerations regarding achievable accuracy become important in attempting to apply ASME PTC 10 to devices commonly classified as fans. Refer to ASME PTC 11 on fans and ASME PTC 13 on blowers for further information.

1-2.3 Compressor Performance and Polytropic Assumption

Compression flow path physical boundaries shall be set to allow use of standardized performance calculation procedures. For purposes of this Code, compressor performance is defined to be between inlet and discharge flanges for a single compressor section. Additionally, within a section, the compression process is considered to occur along a constant efficiency polytropic path. This Code provides definitions and calculation methods adhering to these stated boundaries, assumptions, and resulting limitations.

1-3 EQUIPMENT NOT COVERED BY THIS CODE

The calculation procedures provided in this Code are based on the compression of a single-phase fluid. They should not be used for a fluid containing suspended solids or any liquid, when nongaseous phase occurs in the supplied fluid or could be formed in the compression process.

This does not preclude the use of this Code on a gas where condensation occurs in a cooler, provided the droplets are removed prior to the gas entering the next section of compression.

The calculation procedures provided in this Code should not be used when a chemical reaction takes place during the compression process. This Code does not cover compressors constructed with liquid-cooled diaphragms or built-in internal heat exchangers (e.g., isothermal compressors).

1-4 TYPES OF TESTS

This Code contains provisions for the following two different types of tests:

(a) A Type 1 test shall be conducted on a gas similar to the specified gas with limited deviations between test conditions and specified operating conditions as defined in Tables 3-2.1-1 and 3-2.1-2.

(b) A Type 2 test permits the use of a substitute test gas and extends the permissible deviations between test conditions and specified operating conditions as defined in Table 3-2.1-2.

1-5 PERFORMANCE RELATION TO GUARANTEE

This Code provides the means to determine the performance of a compressor at specified operating conditions.

Contractual guarantees are outside the scope of this Code. Use of the test results for such guarantees should be agreed on by the parties prior to the test.

1-6 ALTERNATE PROCEDURES

Definitive procedures for testing compressors are described herein. If any other procedure or test configuration is used, it shall be agreed on in writing prior to the test by the participating parties. However, no deviations may be made that will violate any mandatory requirements of this Code when the tests are designated as tests conducted in accordance with ASME PTC 10.

The mandatory rules of this Code are characterized by use of the word "shall." If a statement is of an advisory nature, it is indicated by use of the word "should" or is stated as a recommendation.

1-7 INSTRUCTIONS

ASME PTC 1 shall be followed where applicable. The instructions in ASME PTC 10 shall prevail over other ASME Performance Test Codes where there is any conflict.

1-8 REFERENCES

Unless otherwise specified, references to other Codes refer to ASME Performance Test Codes. Literature references are shown in Nonmandatory Appendix D.

Section 2
Definitions and Description of Terms

2-1 BASIC SYMBOLS AND UNITS

See Table 2-1-1 for a description of symbols and units used in this Code. See Table 2-1-2 for a description of subscripts used in this Code.

2-2 PRESSURE DEFINITIONS

absolute pressure: the pressure measured above a perfect vacuum.

differential pressure: the difference between any two pressures measured with respect to a common reference (e.g., the difference between two absolute pressures).

discharge static pressure: the absolute static pressure that exists at the discharge measuring station.

discharge total pressure: the absolute total pressure that exists at the discharge measuring station. Unless specifically stated otherwise, this is the compressor discharge pressure.

dynamic pressure: the difference between the total pressure and the static pressure at the same point in a fluid.

gauge pressure: the pressure measured directly with the existing barometric pressure as the zero-base reference.

inlet static pressure: the absolute static pressure that exists at the inlet measuring station.

inlet total pressure: the absolute total pressure that exists at the inlet measuring station. Unless specifically stated otherwise, this is the compressor inlet pressure.

static pressure: the pressure measured in such a manner that no effect is produced by the velocity of the flowing fluid.

total pressure: an absolute or gauge pressure that would exist when a moving fluid is brought to rest and its kinetic energy is converted to an enthalpy rise by an isentropic process from the flow condition to the stagnation condition. In a stationary body of fluid, the static and total pressures are equal. Also called *stagnation pressure*.

2-3 TEMPERATURE DEFINITIONS

absolute temperature: the temperature measured above absolute zero. It is stated in degrees Rankine or kelvin. Numerically, the Rankine temperature is the Fahrenheit temperature plus 459.67°F and the kelvin temperature is the Celsius temperature plus 273.15°C.

discharge measured temperature: the temperature recorded by a measuring device at a section boundary discharge stream. Temperature measurement devices located in a flowing stream may not provide a direct reading of either static temperature or total discharge temperature, but rather a value between these two temperatures. See Nonmandatory Appendix G for the relationships between static, measured, and total temperatures in a flowing fluid stream. In a stationary body of fluid, the static, measured, and total temperatures are equal.

discharge static temperature: the absolute static temperature that exists at the discharge measuring station.

discharge total temperature: the absolute total temperature that exists at the discharge measuring station. Unless specifically stated otherwise, this is the compressor discharge temperature.

dynamic temperature: the difference between the total temperature and the static temperature at the measuring station.

inlet measured temperature: the temperature recorded by a measuring device at a section boundary inlet stream. Temperature measurement devices located in a flowing stream may not provide a direct reading of either static temperature or total inlet temperature, but rather a value between these two temperatures. See Nonmandatory Appendix G for the relationships between static, measured, and total temperatures in a flowing fluid stream. In a stationary body of fluid, the static, measured, and total temperatures are equal.

inlet static temperature: the absolute static temperature that exists at the inlet measuring station.

Table 2-1-1
Symbols and Units

Symbol	Description	U.S. Customary Units	Multiply by	SI Units
A	Flow channel cross-sectional area	ft^2	0.0929	m^2
a	Speed of sound	ft/sec	0.3048	m/s
A_h	Flow straightener hole area	in.^{2}	0.000645	m^2
A_p	Flow straightener pipe area	in.^{2}	0.000645	m^2
b	Fluid flow passage tip width	ft	0.3048	m
c	Specific heat (liquid)	Btu/(lbm·°R)	4,186.8	J/(kg·K)
c_p	Specific heat at constant pressure	Btu/(lbm·°R)	4,186.8	J/(kg·K)
$c_{p_{\text{film}}}$	Specific heat of air film	Btu/(lbm·°R)	4,186.8	J/(kg·K)
CTE_{film}	Coefficient of thermal expansion of air film	1/°R	0.556	1/K
c_v	Specific heat at constant volume	Btu/(lbm·°R)	4,186.8	J/(kg·K)
D	Diameter	in.	0.0254	m
d	Diameter of flowmeter element	in.	0.0254	m
e_{rad}	Emissivity	Nondimensional	1	Nondimensional
h	Specific enthalpy	Btu/lbm	2,326	J/kg
h_{conv}	Coefficient of convective heat transfer for casing and adjoining pipe	Btu/(hr·ft^2·°R)	5.678	W/(m^2·K)
k	Ratio of specific heats, c_p/c_v	Nondimensional	1	Nondimensional
ke	Specific kinetic energy ($V^2/2$)	Btu/lbm	2,326	J/kg
ln	Natural logarithm (base e)	Nondimensional	1	Nondimensional
log	Common logarithm (base 10)	Nondimensional	1	Nondimensional
M	Fluid Mach number	Nondimensional	1	Nondimensional
$\dot{m}$	Mass flow rate	lbm/min	0.00756	kg/s
Mm	Machine Mach number	Nondimensional	1	Nondimensional
MW	Molar mass (molecular weight)	lbm/lbmole	1	kg/kmol
N	Rotational speed	rpm	0.01667	1/s
n_r	Total number of test point readings	Nondimensional	1	Nondimensional
n_s	Isentropic volume exponent	Nondimensional	1	Nondimensional
n_{sl}	Seal type reference constant (see Nonmandatory Appendix B)	Nondimensional	1	Nondimensional
n_{step}	Total number of iteration steps	Nondimensional	1	Nondimensional
Nu	Nusselt number	Nondimensional	1	Nondimensional
P	Power	hp	745.7	W
p	Pressure	psi	6,895	Pa (= N/m^2)
p_{crit}	Critical pressure	psi	6,895	Pa
Pr	Prandtl number	Nondimensional	1	Nondimensional
p_r	Reduced pressure, p/p_{crit}	Nondimensional	1	Nondimensional
p_v	Dynamic pressure	psi	6,895	Pa
q	Volume flow rate, capacity	ft^3/min	0.000472	m^3/s
Q_{conv}	Section convective heat transfer	Btu/min	17.58	W
Q_m	Total mechanical losses (equivalent)	Btu/min	17.58	W
Q_{rad}	Section radiative heat transfer	Btu/min	17.58	W
Q_{sb}	Heat transfer from the section boundaries	Btu/min	17.58	W
Q_{sl}	External seal loss equivalent	Btu/min	17.58	W
R	Specific gas constant	(ft-lbf)/(lbm·°R)	5.381	J/(kg·K)
Ra	Raleigh number	Nondimensional	1	Nondimensional
Re	Fluid Reynolds number	Nondimensional	1	Nondimensional
Rem	Machine Reynolds number	Nondimensional	1	Nondimensional
r_f	Recovery factor	Nondimensional	1	Nondimensional
r_p	Pressure ratio	Nondimensional	1	Nondimensional
r_q	Volume flow ratio	Nondimensional	1	Nondimensional

Table 2-1-1
Symbols and Units (Cont'd)

Symbol	Description	U.S. Customary Units	Multiply by	SI Units
r_t	Temperature ratio	Nondimensional	1	Nondimensional
r_v	Specific volume ratio	Nondimensional	1	Nondimensional
s	Specific entropy	Btu/(lbm·°R)	4,186.8	J/(kg·K)
S_c	Heat transfer surface area of exposed compressor casing and adjoining pipe	ft^2	0.0929	m^2
S_x	Standard deviation for a test reading	Varies	...	Varies
T	Absolute temperature	°R	0.5556	K
$Tcond_{film}$	Thermal conductivity of air film	Btu/(hr·ft·°R)	1.730735	W/(m·K)
T_{crit}	Critical temperature	°R	0.5556	K
U	Blade tip speed	ft/sec	0.3048	m/s
u	Specific internal energy	Btu/lbm	2,326	J/kg
V	Fluid velocity	ft/sec	0.3048	m/s
v	Specific volume	ft^3/lbm	0.06243	m^3/kg
Vis_{film}	Dynamic viscosity of air film	lbm/(ft·sec)	1.488	kg/(m·s)
w	Work per unit mass	(ft-lbf)/lbm	2.989	J/kg
X	Compressibility function	Nondimensional	1	Nondimensional
Z	Compressibility factor as used in gas law, $pv = ZRT$	Nondimensional	1	Nondimensional
Δp	Differential pressure	psi	6,895	Pa
Δt	Differential temperature	°F, °R	0.5556	°C, K
∂	Partial derivative	Nondimensional	1	Nondimensional
δ	Absolute difference between an individual observation and the average for a reading	Varies	...	Varies
η	Efficiency	Nondimensional	1	Nondimensional
λ	Friction factor	Nondimensional	1	Nondimensional
μ	Dynamic (absolute) viscosity	lbm/(ft·sec)	1.488	kg/(m·s)
μ_{in}	Work input coefficient	Nondimensional	1	Nondimensional
μ_p	Polytropic work coefficient	Nondimensional	1	Nondimensional
ν	Kinematic viscosity	ft^2/sec	0.0929	m^2/s
π	3.14159	Nondimensional	1	Nondimensional
ρ	Density	lbm/ft^3	16.02	kg/m^3
ρ_{film}	Density of air film	lbm/ft^3	16.02	kg/m^3
Σ	Summation	Nondimensional	1	Nondimensional
σ	Stefan–Boltzmann constant	Btu/(hr·ft^2·$°R^4$)	33.0827	W/(m^2K^4)
τ	Torque	ft-lbf	1.356	N·m
τ	Function of the number of probes being used to measure a single variable and intended to identify observations that are more than two standard deviations away from the mean [Note (1)]	Nondimensional	1	Nondimensional
ϵ	Surface roughness	in.	0.0254	m
Ω	Total work input coefficient	Nondimensional	1	Nondimensional
ϕ	Flow coefficient	Nondimensional	1	Nondimensional

GENERAL NOTE: Although a symbol for one unit mass is not specified, the conversion 1 kg = 2.204622622 lbm is applied by other symbols in this Code.

NOTE: (1) This definition is used in Nonmandatory Appendix C.

Table 2-1-2
Subscripts

Subscript	Description
a	Ambient
c	Casing
cond	Conduction
conv	Convection
corr,η	Correction for efficiency
corr,μ	Correction for polytropic work coefficient
corr,ϕ	Correction for flow coefficient
crit	Fluid's critical point values
csp	Converted to specified conditions
d	Compressor discharge conditions
film	Film
g	Gas
hb	Heat balance
i	Compressor inlet conditions
i	Iterator variable [Note (1)]
in	Flow stream entering defined control volume
in	Input (as in work input coefficient μ_{in})
j	Iteration (step) number, exponent
l	Leakage
ld	Leakage downstream
lu	Leakage upstream
m	Main stream
meas	Measured
mech	Mechanical losses
other	Other losses
out	Flow stream departing defined control volume
p	Polytropic, pipe
par	Parasitic losses
R	Rotor
r	Ratio, reduced
rad	Radiation
sb	Section boundary
sh	Shaft
sp	At specified conditions
static	Static
t	Test
x	Individual observation of an instrument taken during the test run
∞	Critical or at infinity

NOTE: (1) This definition is used in para. 5-4.3.3.

inlet total temperature: the absolute total temperature that exists at the inlet measuring station. Unless specifically stated otherwise, this is the compressor inlet temperature.

static temperature: the temperature determined in such a way that no effect is produced by the velocity of the flowing fluid.

total temperature: the temperature that would exist when a moving fluid is brought to rest and its kinetic energy is converted to an enthalpy rise by an isentropic process from the flow condition to the stagnation condition. In a stationary body of fluid, the static and the total temperatures are equal. Also called *stagnation temperature.*

2-4 OTHER GAS (FLUID) PROPERTIES DEFINITIONS

acoustic velocity: the acoustic velocity for a pressure wave or acoustic wave of infinitesimal amplitude in any medium is given by

$$a = \sqrt{\left(\frac{\partial p}{\partial \rho}\right)_s} = \sqrt{n_s Z R T_{\text{static}}}$$

and is described by an isentropic process. Also called *sonic velocity*.

compressibility function, X: the change of compressibility factor, Z, with temperature at constant pressure.

$$X = \frac{T}{Z}\left(\frac{\partial Z}{\partial T}\right)_p = \sqrt{\frac{k}{ZRn_s}(c_p - c_v)} - 1$$

NOTE: For ideal gases, $X = 0$.

density: the mass of the gas per unit volume. Density is determined at a point once the total pressure and total temperature are known at that point.

dynamic viscosity: the intensity of viscous shear within a fluid. Also called *absolute viscosity*.

fluid Mach number: the ratio of fluid velocity to acoustic velocity.

fluid velocity: the average fluid flow velocity at the location of interest. The average velocity at the measurement station is given by

$$V = \frac{\dot{m}}{\rho_{\text{static}} A}$$

isentropic volume exponent, n_s: the relationship between volume change and pressure change of a real gas along an isentropic path. Specifically

$$n_s = -\frac{v}{p}\left(\frac{\partial p}{\partial v}\right)_s$$

NOTE: For ideal gases, $n_s = k$.

kinematic viscosity: the dynamic viscosity of a fluid divided by the fluid density.

molar mass: the mass of a fluid referred to that of an atom of carbon-12. Also called *molecular weight*.

ratio of specific heats, k: equal to cp/cv.

NOTE: The relationship $c_p - c_v = R$ is not valid for real gases. To calculate the ratio of specific heats, values of c_p and c_v must be independently calculated.

real gas: a fluid that obeys the general equation of state, $pv = ZRT$, in which Z is not always equal to unity and thermodynamic properties are functions of both pressure and temperature.

specific gravity: the ratio of the molecular weight of the gas or gas mixture to that of air.

NOTE: Gas specific gravity can be exactly defined as the ratio of the density of a gas at defined standard conditions to the density of air at the same defined standard conditions.

specific heat at constant pressure, c_p: the change in enthalpy with respect to temperature at a constant pressure. Represented by $c_p = (\partial h/\partial T)_p$.

specific heat at constant volume, c_v: the change in internal energy with respect to temperature at a constant specific volume. Represented by $c_v = (\partial u/\partial T)_v$.

specific volume: the volume occupied by a unit mass of gas. Specific volume is determined at a point once the total pressure and total temperature are known at that point.

2-5 OPERATING CHARACTERISTICS DEFINITIONS

capacity: of a compressor, the rate of flow, determined by delivered mass flow rate divided by inlet total density. For sidestream machines, this definition shall be applied to individual sections.

compressor choke point: the maximum capacity when the machine is run at a given speed and guide vane geometry.

NOTE: Although aerodynamic choke is understood as a limitation of flow as a result of reaching sonic velocity in a complete cross section of at least one aerodynamic flow passage at a given compressor speed or guide vane setting, for purposes of this Code, it is simply understood as a defined maximum capacity (see para. 3-10.7).

compressor surge point: the minimum stable capacity when the machine is run at a given speed and guide vane geometry. The compressor surge point occurs when flow is reduced, the compressor back pressure exceeds the pressure developed by the compressor, and a breakdown in flow results. This immediately causes a reversal in the flow direction and reduces the compressor back pressure.

control volume: a region of space selected for analysis where the flow streams entering and leaving can be quantitatively defined, as can the power input and heat exchange by conduction and radiation. Such a region is in equilibrium for both a mass and energy balance.

flow coefficient: a nondimensional parameter proportional to the compressed mass flow rate divided by the product of inlet density, blade tip speed, and the square of the blade tip diameter. For centrifugal compressors, the blade tip diameter is the maximum blade diameter at the trailing edge of the impeller blades. For axial compressors, the blade tip diameter is the blade tip diameter at the leading edge of the rotor blades. The compressed mass flow rate is the net mass flow rate through the rotor.

leakage ratio: the sectional external leakage mass flow divided by the inlet mass flow.

machine Mach number: defined as the ratio of the blade velocity at the largest blade tip diameter of the first impeller for centrifugal machines or at the tip diameter of the leading edge of the first-stage rotor blade for axial flow machines to the acoustic velocity of the gas at the total inlet conditions.

NOTES:
(1) Machine Mach number is not to be confused with local fluid Mach number.
(2) Although acoustic velocity is calculated with a static temperature, for purposes of this Code, total temperature is used.

machine Reynolds number: defined by the equation $Rem = Ub/v$, where U is the velocity at the outer blade tip diameter of the first impeller or of the first-stage blade tip diameter of the leading edge, v is the total kinematic viscosity of the gas at the compressor inlet, and b is a characteristic length. For centrifugal compressors, b shall be taken as the exit width at the outer blade diameter of the first-stage impeller. For axial compressors, b shall be taken as the chord length at the tip of the first-stage rotor blade. These variables shall be expressed in consistent units to yield a nondimensional ratio.

NOTE: Although kinematic viscosity is calculated at static conditions, for purposes of this Code, total conditions are used.

maximum capacity: the highest flow rate at which the performance curve terminates at a given speed and guide vane geometry.

NOTE: Maximum capacity may be the highest continuous operable flow rate at a given speed and guide vane geometry.

operating range: the variation in stable flow rate between the compressor surge point and the maximum capacity along a constant speed or fixed guide vane geometry.

pressure ratio: the ratio of the absolute discharge total pressure to the absolute inlet total pressure.

pressure rise: the difference between the discharge total pressure and the inlet total pressure.

section: one or more stages having the same mass flow without external heat transfer other than casing heat transfer.

sidestream: a partial flow addition or extraction to the flow of an upcoming section.

specific volume ratio: the ratio of inlet specific volume to discharge specific volume in total conditions. For an inward sidestream section, the inlet specific volume shall be of the fully mixed combination of the net stream leaving the previous section and the added sidestream.

stage:

(a) for a centrifugal compressor, a single impeller and its associated stationary flow passages.

(b) for an axial compressor, a single row of rotating blades and its associated stationary blades and flow passages.

temperature ratio: the ratio of the absolute discharge total temperature to the absolute inlet total temperature.

temperature rise: the difference between the discharge total temperature and the inlet total temperature.

volume flow rate: as used in this Code, the local mass flow rate divided by local total density. Volume flow rate is used to determine the volume flow ratio.

volume flow ratio: the ratio of volume flow rates at two points in the flow path.

2-6 WORK, POWER, AND EFFICIENCY DEFINITIONS

The following definitions apply to a section:

gas power: the sum of the products of mass flow rates and enthalpies of these flows entering and exiting a control volume surrounding the section plus the heat loss from that section to ambient temperature. The gas power is applied to the section by the rotor.

NOTE: Nonmandatory Appendix E provides definitive equations for gas power determination.

gas specific work: the amount of work performed by the rotor to compress and deliver a unit mass of gas. Gas specific work is equal to the enthalpy rise of the compressed gas between the sectional inlet and the discharge conditions.

mechanical losses: the total power consumed by frictional losses in integral gearing, bearings, and shaft end seals.

parasitic losses: the difference between shaft power and gas power for the section or sections of interest.

polytropic compression: a reversible compression process between the inlet total pressure and inlet total temperature and the discharge total pressure and discharge total temperature. The change in the gravitational potential energy is assumed to be negligible. The polytropic process follows a path such that the polytropic efficiency is constant during the process.

polytropic efficiency: the ratio of the polytropic work to the gas specific work.

polytropic work: the reversible work required to compress a unit mass of gas along a polytropic path from the inlet total pressure and inlet total temperature to the discharge total pressure and discharge total temperature. Polytropic work may also be referred to as polytropic head by industry and in literature, but for the purposes of this Code, polytropic work is the preferred term.

polytropic work coefficient: the nondimensional ratio of the polytropic work to the sum of the squares of the blade tip speeds of all stages in a given section.

shaft power: the power delivered to the compressor shaft. Shaft power is the gas power plus the parasitic losses in the compressor. Also called *brake power*.

total specific work: the sum of the gas specific work and the amount of additional specific work performed by the rotor to sustain sectional external leakage and sidestream flows upstream and downstream of the rotor. The additional specific work comprises enthalpy changes associated with sectional external leakage and sidestream flows multiplied by their respective mass flow rates and referenced to the total mass flow rate passing through the rotor.

total work input coefficient: the nondimensional ratio of the total work input to the gas to the sum of the squares of the blade tip speeds of all stages in a given section.

work input coefficient: the nondimensional ratio of the enthalpy rise to the sum of the squares of the blade tip speeds of all stages in a given section.

2-7 MISCELLANEOUS DEFINITIONS

acceptable raw data: raw data that meets the limitations imposed in Sections 3 and 4 and has outliers removed.

corrected raw data: data that is obtained from applying corrections to the acceptable raw data as described in para. 5-4.3.1.

equivalent conditions: for purposes of this Code, the conditions when, for the same flow coefficient, the ratios of the three dimensionless parameters (specific volume ratio, machine Mach number, and machine Reynolds number) fall within the limits prescribed in Table 3-2.1-2. Also called *similitude*.

fluctuation: the highest reading minus the lowest reading divided by the average of all readings, expressed as a percent.

pipe Reynolds number: the Reynolds number for the gas flow in a pipe. The pipe Reynolds number is defined by the equation $Re = VD/v$, where

D = the characteristic length that is the inside pipe diameter at the pressure-measuring station
V = the average velocity at the pressure-measuring station
v = the kinematic viscosity for the static temperature and static pressure at the measuring station

The pressure- and temperature-measuring stations for flow-metering calculations shall be specified as in Section 4. The variables in the Reynolds number shall be expressed in consistent units to yield a dimensionless ratio.

NOTE: Pipe Reynolds number is different from machine Reynolds number. See definition in subsection 2-5.

raw data: the recorded observations of an instrument taken during the test run.

reading: the average of the corrected raw data at any given measurement station.

specified operating conditions: those conditions for which compressor performance is to be evaluated.

speed: the rotational speed of the compressor rotor that comprises a given section.

test operating conditions: the operating conditions prevailing during the test.

test point: three or more readings that have been averaged and that fall within the permissible specified fluctuation.

2-8 INTERPRETATION OF SUBSCRIPTS

Certain values for thermodynamic state and mass flow rate are used in the computation of nondimensional performance parameters such as Mm, Rem, r_v, ϕ, μ_p, μ_i, η_p, and Ω. Unless otherwise specifically stated, the thermodynamic total conditions are used. The subscripts used in these equations are interpreted as described in (a) through (e).

(a) The subscript i on thermodynamic state variables denotes inlet conditions. For single-entry streams, it refers to conditions at the section inlet measurement station. For multiple-inlet streams, it refers to a calculated mixed state. See Nonmandatory Appendix E, subsection E-5.

(b) The subscript d on thermodynamic state variables denotes discharge conditions. It refers to conditions at the mainstream discharge measurement station.

(c) The subscript R is used on mass flow rate to denote the net mass flow rate compressed by the rotor. Determination of the net mass flow rate compressed by the rotor requires that all measured flows and calculated leakages be considered. See Nonmandatory Appendix E, Figure E-3.11-1.

(d) The subscript sp refers to quantities at specified conditions, which are known before the test and do not change as a result of the test.

(e) The subscript csp refers to quantities that are converted from the test results (Table 5-6.1.2-1) to specified conditions and represent the as-built performance of the compressor at specified conditions. See Table 5-6.1.2-2.

Section 3
Guiding Principles

3-1 PLANNING THE TEST

Before undertaking a test in accordance with the rules of this Code, ASME PTC 1 shall be consulted. ASME PTC 1 explains the intended use of the Performance Test Codes and is particularly helpful in the initial planning of the test.

When a test is to be conducted in accordance with this Code, the scope, procedures, and calculation methods to be used shall be determined in advance and documented in the test procedure. Selections of pipe arrangements, test driver, instruments, and test gas, if applicable, shall be made. Estimates of the probable uncertainty in the planned measurements shall be made. The uncertainty calculation method shall be agreed to by the interested parties in advance of the test.

NOTE: Pretest and post-test uncertainties as described by ASME PTC 1 should be calculated and agreed on by the interested parties.

The scope of the test shall be agreed to by the interested parties. This may be dictated in advance by contractual commitments or may be mutually agreed on prior to the start of the test. This Code contains procedures for a single point performance test and gives guidance on determining a complete performance curve.

3-1.1 Specified Conditions

Specified conditions are mass flow rate, inlet pressure, inlet temperature, gas composition, gas physical properties, cooling water temperature (as applicable), operating speed, and guide vane geometry (as applicable).

NOTE: Specified operating speed and guide vane geometry (as applicable) are generally defined to meet the expected discharge pressure of the compressor.

3-1.2 Test Objectives

A detailed written statement of the test objectives shall be developed prior to conducting the test.

3-1.3 Test Facility

A test facility shall be selected. See subsection 1-1.

3-1.4 Test Personnel

The number of test personnel should be sufficient to ensure a careful and orderly observation of all instruments with time between observations to check for indications of error in instruments or observations.

3-1.5 Responsible Individual

An individual shall be designated as responsible for conducting the test.

3-2 TYPES OF TESTS

This Code defines two types of test, which are based on the deviations between test conditions and specified operating conditions.

3-2.1 Type 1 Tests

Type 1 tests are conducted with the specified gas and operating conditions with permissible deviations as specified by Table 3-2.1-1. These limitations are subject to the further restriction that their individual and combined effects shall not exceed the limits of Table 3-2.1-2. A Type 1 test shall be required for specified machine Reynolds numbers below 90,000. No machine Reynolds number correction is applicable for Type 1 tests.

Table 3-2.1-1
Permissible Deviation From Specified Operating Conditions for Type 1 Tests

Variable	Symbol	SI Units	U.S. Customary Units	Permissible Deviation, %
Inlet pressure	p_i	Pa	psia	5
Inlet temperature	T_i	K	°R	8
Speed	N	1/s	rpm	2
Molecular weight	MW	kg/kmol	lbm/lbmole	2
Capacity	q_i	m^3/s	ft^3/min	4

GENERAL NOTES:
(a) A Type 1 test is conducted on a gas with similar composition to the specified gas with limited deviations between test composition and operating conditions and specified composition and operating conditions. Deviations are based on the specified values, where pressures and temperatures are expressed in absolute values.
(b) The combined effect of inlet pressure, temperature, compressibility factor, and molecular weight shall not produce more than an 8% deviation in the inlet gas density.
(c) Efforts shall be made to maintain the volume reduction as close to that at specified conditions as possible through variation of the variables within the deviations allowed.

3-2.2 Type 2 Tests

Type 2 tests are conducted subject to the limits of Table 3-2.1-2 only. The specified gas or a substitute gas may be used. The test speed required is often different from the specified operating condition speed.

3-2.3 Test Type Selection

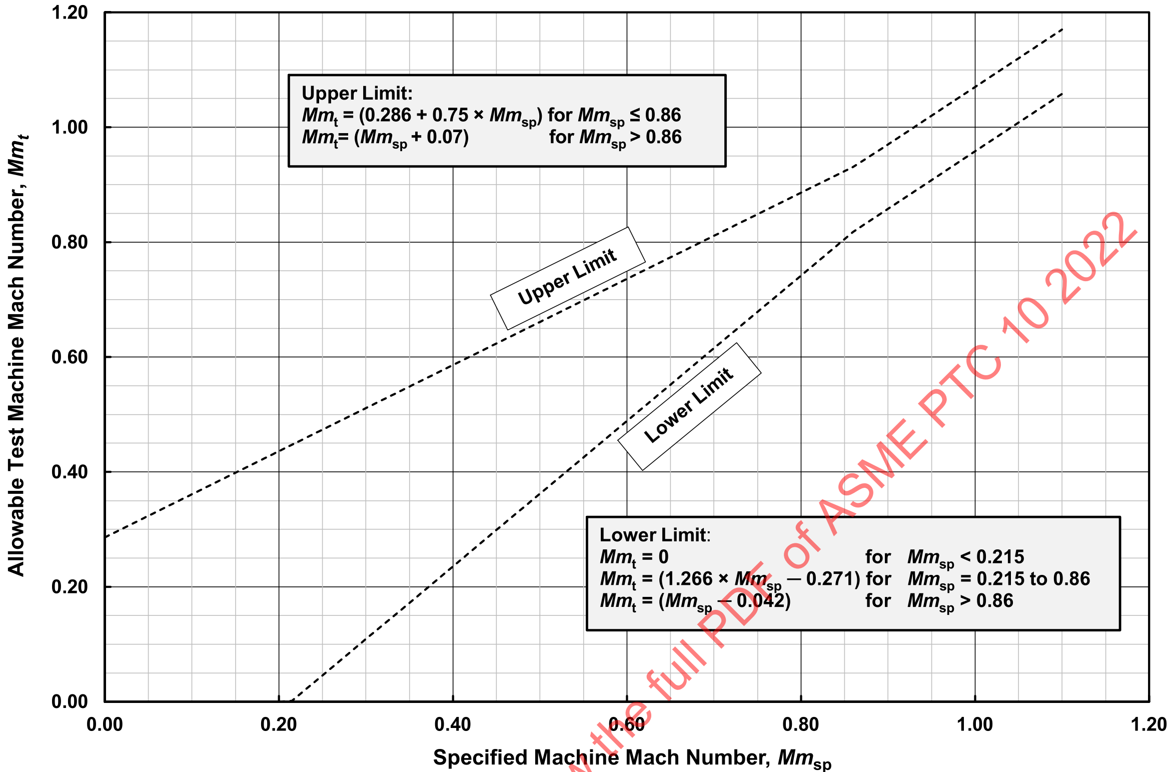
The selection of test type shall be made in advance of the test. In the interest of minimizing uncertainty of test results, it is desirable that test conditions duplicate specified operating conditions as closely as possible. The limits in Table 3-2.1-1 provide maximum allowable deviations of individual parameters for Type 1 tests. The permissible deviations in Table 3-2.1-2 provide maximum allowable deviations of the fundamental nondimensional parameter groupings for both types. The emphasis in conducting either a Type 1 test or a Type 2 test should be toward minimizing these deviations. The most reliable test results would be expected when the deviations in Tables 3-2.1-1 and 3-2.1-2 are minimized. Results of the test shall be validated by applying Table 3-2.1-2 permissible deviations to the converted-to-specified values.

Table 3-2.1-2
Permissible Deviation From Specified Operating Parameters for Type 1 and Type 2 Tests

Parameter	Symbol	Limit of Test Values as Percent of Specified Values [Note (1)] Min.	Max.
Specific volume ratio	v_i/v_d	95	105
Flow coefficient	ϕ	96	104
Machine Mach number	Mm		
Centrifugal compressors		See Figure 3-2.1-1	
Axial compressors		See Figure 3-2.1-2	
Machine Reynolds number	Rem		
Centrifugal compressors [Note (2)]		See Figure 3-2.1-3	
Axial compressors where the machine Reynolds number at specified conditions is less than 100,000 [Note (2)]	…	90	105
Axial compressors where the machine Reynolds number at specified conditions is greater than 100,000	…	10	200

NOTES:
(1) The limits in this table are applied to specified and converted-to-specified values. See para. 3-2.3.
(2) Minimum allowable specified machine Reynolds number is 90,000 for a Type 2 test with Reynolds number correction.

Figure 3-2.1-1
Allowable Test Machine Mach Numbers for Centrifugal Compressors



3-2.4 Calculation Procedures

Calculation procedures for real gases are given in Section 5. See para. 3-4.1 and subsection 5-1.

NOTE: Simplified thermodynamic relationships of ideal gases shall not be used.

3-3 LIMITATIONS

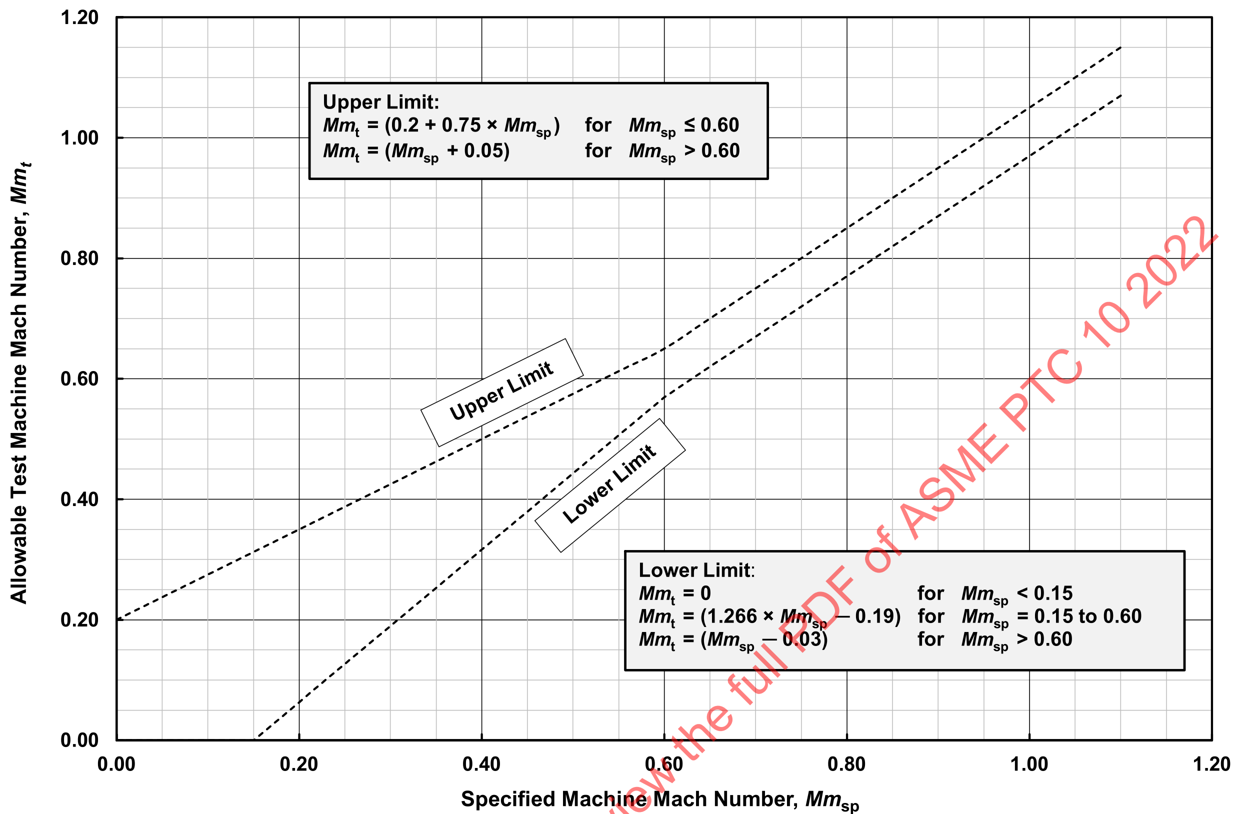
3-3.1 Section Boundaries

The methods in this Code can be applied for conversion of test results to specified operating condition results for compressors that consist of one or more sections. Section boundaries are defined by surfaces of the compressor flow path within the tested section and cross sections of inlet and outlet flow streams. The interior of connected section boundaries forms a section control volume, as depicted in Figure 3-3.1-1. The gas state and flow rate shall be established for each stream, where it crosses the section boundary. In the case where power input is calculated and not directly measured, the power input shall satisfy energy balance for a section control volume. The energy balance shall account for heat transfer crossing the test section boundary. Heat exchangers are excluded from the interior of the section boundaries.

3-3.2 Sidestreams

Compressors with sidestreams can be tested using the procedures for a Type 1 test, provided all conditions, including those at the sidestream, meet the requirements of Table 3-2.1-1. Compressors with sidestreams can also be tested by individual sections using the criteria for a Type 2 test.

Figure 3-2.1-2
Allowable Test Machine Mach Numbers for Axial Compressors



3-3.3 Leakage Ratio

Leakage ratio shall be estimated for both test conditions and specified conditions. If the difference of leakage ratios between test conditions and specified conditions exceeds 0.02, the flow rate corrections for leakage shall be applied to the calculations of capacity and power.

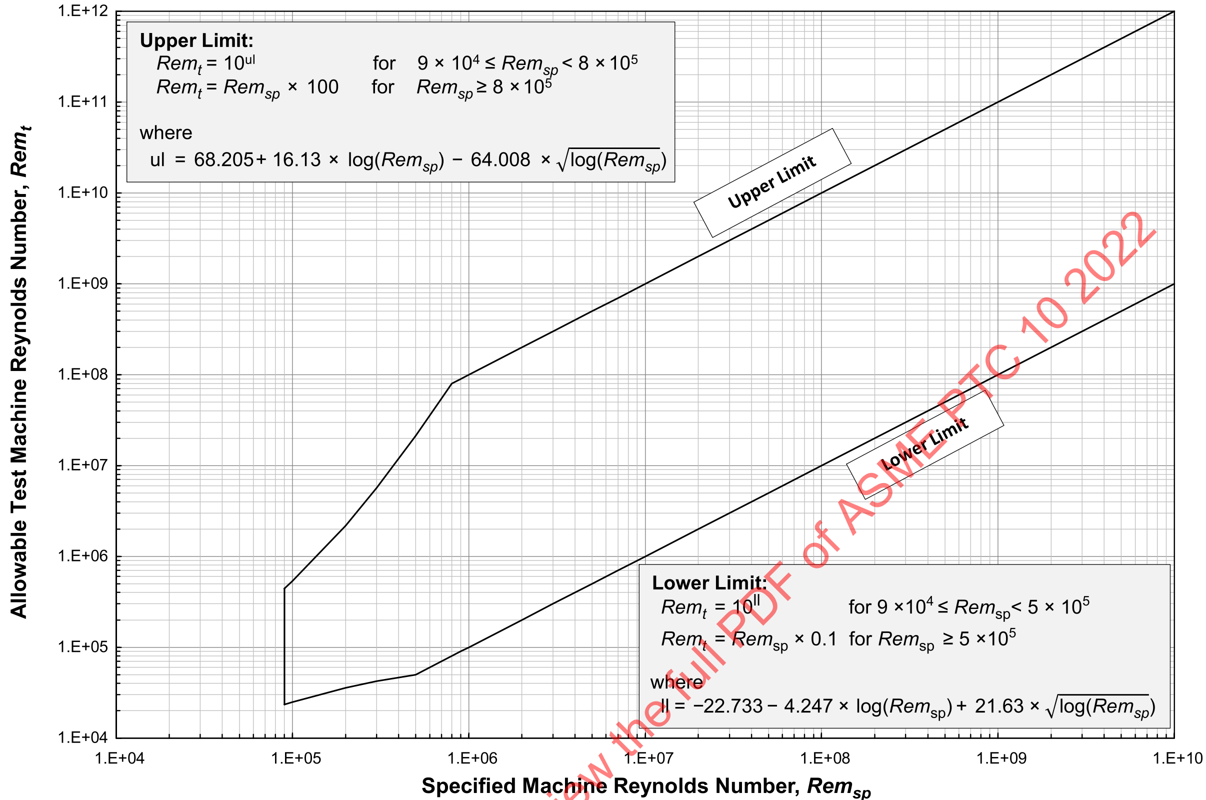
The condition that requires flow rate corrections is represented by the following relation:

$$\left| \left(\frac{\dot{m}_l}{\dot{m}_{\text{in}}} \right)_{sp} - \left(\frac{\dot{m}_l}{\dot{m}_{\text{in}}} \right)_{\text{test}} \right| \geq 0.02$$

These corrected conditions of the tested section shall comply with requirements defined in Table 3-2.1-2.

NOTES:
(1) Specific volume ratio may in practice differ between Type 2 test conditions and specified operating conditions due to leakage differences. For example, as it is common to test at reduced inlet pressure, the reduced differential pressure across a seal to atmosphere could result in zero or negative leakage. Therefore, specific volume ratio equality may not be achieved between Type 2 test conditions and specified conditions. Sample calculations of leakages for a back-to-back compressor are presented in Nonmandatory Appendix B.
(2) Any leakage inside the control volume as shown in Figure 3-3.1-1 from stage shaft and impeller eye seals and seal gas flows are normally not measured, but their effect is included through efficiency impacts.
(3) Any leakage crossing the control volume boundary as shown in Figure 3-3.1-1, such as those related to balance pistons and multisection separation seals, may require or be selected for measurement to improve calculation accuracy.

Figure 3-2.1-3
Allowable Test Machine Reynolds Numbers Departure for Centrifugal Compressors



3-3.4 Mechanical Losses

When efficiency is to be determined by shaft input power measurements, mechanical losses (see para. 5-4.7.4) shall not exceed 10% of the total test power.

NOTE: This will minimize the effect of uncertainties in these mechanical losses on the determination of gas power.

3-3.5 Components Between Sections

Evaluation of performance of components between sections, if any, such as heat exchangers, piping, and valves, is beyond the scope of this Code and shall be agreed on by the parties to the test. The specified operating condition performance of such components or the technique for correction of test results to specified operating conditions shall be agreed on by the parties to the test.

3-3.6 Heat Losses

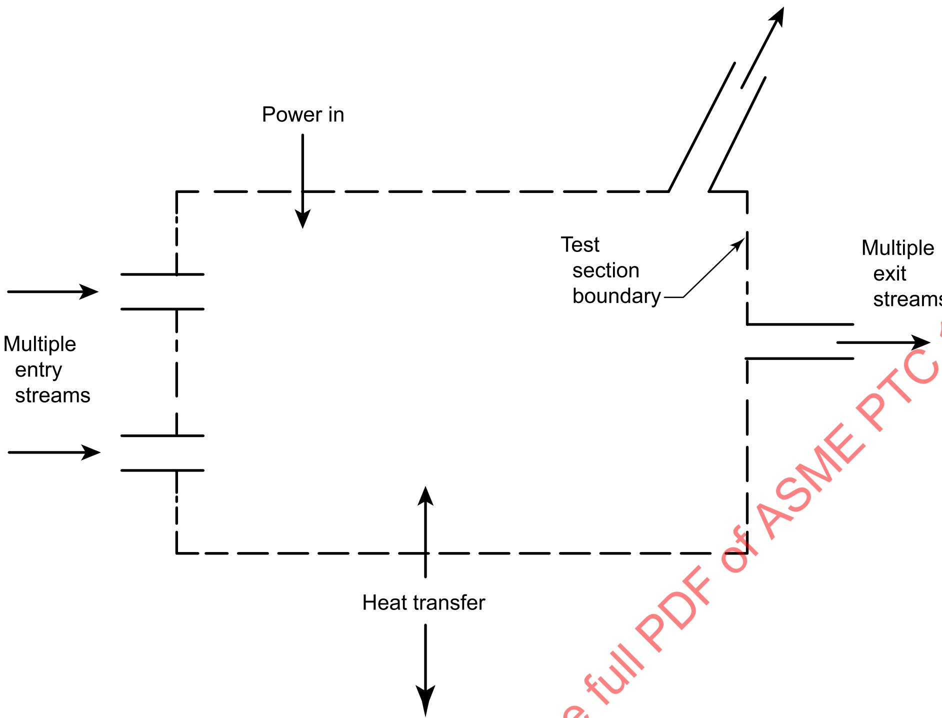
When power is to be determined by the heat balance method, the heat losses (see para. 5-4.7.3) due to radiation and convection, expressed as a percent of the total test power, shall not exceed 5%.

NOTE: This will minimize the effect of uncertainties in the heat loss determination of gas power.

3-3.7 Inlet Gas Superheat Requirement

The inlet gas condition shall have a minimum of 3°C (5°F) of superheat to ensure that there is no condensation before the eye of the first impeller.

Figure 3-3.1-1
Section Control Volume



3-4 TEST GAS AND SPEED

3-4.1 Test Gas Properties

The physical and thermodynamic properties of the specified gas and test gas shall be known. The option of using tabulated data of sufficient resolution to minimize interpolation uncertainties, an equation of state correlation, or experimental determination as a source for these real gas properties shall be agreed on prior to the test.

The following physical and thermodynamic properties of the test gas throughout the expected pressure and temperature range shall be known or accurately determined:

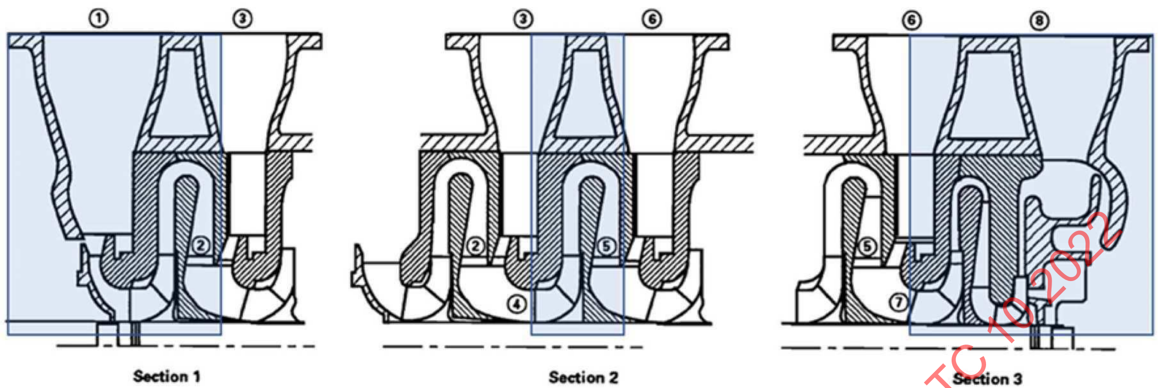
(a) gas composition and molecular weight
(b) specific heat at constant pressure
(c) specific heat at constant volume
(d) ratio of specific heats
(e) compressibility factor
(f) specific volume (density)
(g) dew point
(h) viscosity
(i) isentropic volume exponent
(j) specific enthalpy
(k) specific entropy
(l) acoustic velocity (sonic velocity)

3-4.2 Test Speed

The test speed shall be selected to conform to the limits of Table 3-2.1-2. The test speed shall not exceed the safe operating speed of the compressor. Consideration should be given to critical speeds of rotating equipment when selecting the test speed.

Test pressures and temperatures shall not exceed the maximum allowable pressures and temperatures for the compressor.

Figure 3-5.2-1
Typical Sidestream Sections



Section 1 (Min. 95, Max. 105):

$$r_{q1-2} = \frac{q_1}{q_2} \qquad \frac{(r_{q1-2})_t}{(r_{q1-2})_{sp}} \qquad 95 \qquad 105$$

$$r_{q3-2} = \frac{q_3}{q_2} \qquad \frac{(r_{q3-2})_t}{(r_{q3-2})_{sp}} \qquad 95 \qquad 105$$

Section 2 (Min. 95, Max. 105):

$$r_{q3-2} = \frac{q_3}{q_2} \qquad \frac{(r_{q3-2})_t}{(r_{q3-2})_{sp}} \qquad 95 \qquad 105$$

$$r_{q4-5} = \frac{q_4}{q_5} \qquad \frac{(r_{q4-5})_t}{(r_{q4-5})_{sp}} \qquad 95 \qquad 105$$

$$r_{q6-5} = \frac{q_6}{q_5} \qquad \frac{(r_{q6-5})_t}{(r_{q6-5})_{sp}} \qquad 95 \qquad 105$$

Section 3 (Min. 95, Max. 105):

$$r_{q6-5} = \frac{q_6}{q_5} \qquad \frac{(r_{q6-5})_t}{(r_{q6-5})_{sp}} \qquad 95 \qquad 105$$

$$r_{q7-8} = \frac{q_7}{q_8} \qquad \frac{(r_{q7-8})_t}{(r_{q7-8})_{sp}} \qquad 95 \qquad 105$$

where

subscript 1 = section 1 inlet from flange measurements
subscript 2 = section 1 discharge computed from measurements before sidestream
subscript 3 = section 2 inlet from flange measurements
subscript 4 = section 2 mixed inlet computed
subscript 5 = section 2 discharge computed from internal measurements before sidestream
subscript 6 = section 3 inlet from flange measurements
subscript 7 = section 3 mixed inlet computed
subscript 8 = section 3 discharge from flange measurements

GENERAL NOTE: Permissible deviations of the test volume flow ratio as a percent of the specified values are shown for each section.

3-5 INTERMEDIATE FLOW STREAMS

3-5.1 Section Treatment

Compressors having flows added or removed at intermediate locations (i.e., sidestreams) between the inlet and final discharge are handled by treating the compressor by sections. The gas state and flow rate shall be established for each stream where it crosses the section boundary.

3-5.2 Compressors With Sidestreams

In compressors with sidestreams, the specified volume flow ratio and test volume flow ratio shall be maintained for each section. Permissible deviations from these ratios are listed in Figure 3-5.2-1.

3-5.2.1 In the first section of a sidestream compressor, the ratio of inlet volume flow rate q_1 to discharge volume flow rate q_2 for the test conditions shall be held to within ±5% of the volume flow ratio at the specified conditions. This is the same tolerance that is permitted for the specific volume ratio on a conventional compressor section in Table 3-2.1-2.

3-5.2.2 In the second and all subsequent sections of a sidestream compressor, the ratio of the sidestream volume flow (e.g., q_3 for the second section) to the preceding section discharge flow (e.g., q_2 for the second section) for the test conditions shall be held to within ±5% of the corresponding volume flow ratio at the specified conditions.

This requirement ensures that the relationship between the test flow velocity at the sidestream flange and the preceding discharge is maintained similar to the corresponding velocities at the specified conditions. As a result, the test total pressures at both locations will also have a similar relationship that accurately simulates the performance at specified conditions. The equations to calculate these flow ratios are listed in Figure 3-5.2-1.

3-5.2.3 In the second and all subsequent sections of a sidestream compressor, the volume flow ratio of the flow after the mixing point (e.g., q_4 for the second section) versus the flow at the section outlet (e.g., q_5 for the second section) for the test conditions shall be held to the requirements of Figure 3-5.2-1.

3-5.3 Inward Sidestreams

When the sidestream flow is inward, the discharge temperature of the preceding section shall be measured prior to the mixing of the two streams.

This temperature measurement should be made in a portion of the discharge flow stream to ensure minimizing the effect of the sidestream on the raw data.

Raw data may be affected by heat transfer between a sidestream and a mainstream flow or from recirculation, which may occur within the flow passage.

The discharge temperature of the preceding section is needed to compute the performance and the reference mixed temperature for the next section inlet.

NOTE: It is possible for internal total pressures to exceed flange total pressure due to the higher internal velocities. The higher internal velocities are accompanied by a lower static pressure, which provides a pressure difference for inward flow.

3-5.3.1 Temperature Stratification. It is common for sidestream sectional compressors to have temperature differences between the mainstream and sidestream. It is possible, due to these differences, for thermal flow stratification to exist within the compressor sections. This stratification may result in increased uncertainty of measurements of internal temperatures in downstream sections. Under test conditions, the stream temperature differences should be maintained as close to specified as practical.

3-5.3.2 Performance Definition. The sectional work, efficiencies, and pressures are defined between the flanges in each section.

The only internal measurements needed are the sectional discharge temperatures for computing the mixed temperature conditions and sectional performance. When internal temperature measurements are taken by means of total temperature probes, their orientation shall be aligned to the flow angle expected at the specified point.

The total pressure used for calculating the sectional performance is assumed to be equal to the sidestream flange total pressure.

The internal mixed temperature should be computed on a mass enthalpy basis for obtaining the inlet temperature for subsequent sections.

3-5.4 Extraction Sidestreams

When the intermediate flows are removed (i.e., bleed-off) from the compressor, they will cross a section boundary.

The internal temperature and pressure can be assumed to be equal to the external flange temperature and pressure of the primary internal stream. The ratio of flow rate restrictions in Figure 3-5.2-1 shall also apply to outward flowing sidestreams.

3-5.5 Multisection Compressors

Each section of a multisection compressor shall have its own performance curve defined by a number of test points as per para. 3-10.5.1.

NOTE: Due to deviations in volume flow ratio between a sidestream flange and preceding section discharge at individual test points, it may be difficult to generate smooth sectional performance curves from actual test data. The resulting performance curves may be corrected after test by adjusting flow ratio within the permitted tolerances and using calculated values for sidestream total pressure loss. In such cases, required corrections shall be identified and mutual agreement of these modifications shall be obtained.

3-5.5.1 The specified performance point for each section shall be tested in accordance with the volume flow ratio limitations defined in para. 3-5.2.

3-5.5.2 The additional test points along each section curve shall also comply with the requirements of para. 3-5.2.

NOTE: It is recognized that in some cases performance limitations at the extents of a section curve may not be able to satisfy these conditions. In such cases, required deviations in the volume flow ratio tolerances shall be identified prior to the test and mutual agreement of these modified deviations shall be obtained.

3-6 SAFETY

3-6.1 Compliance

The party providing the test site shall be responsible for complying with any governmental codes, regulations, ordinances, directives, or rules that are applicable to the safety of test personnel and equipment at the test site.

3-6.2 Test Gas Compliance

The following shall be in compliance with local regulations and prudent practice:
(a) the test gas with regard to flammability and/or toxicity.
(b) the test site with regard to overpressure protection. Consideration shall be given to the need for relief valves.

3-6.3 Closed-Loop Testing

Test gases used in a closed loop shall be continuously monitored to avoid combustible mixtures.

3-6.4 System Protection

The party providing the test site will be responsible for establishing the requirements of system protection. The requirement of alarms and/or automatic shutdown devices for high temperature, low oil pressure, compressor overspeed, or other possible malfunctions shall be reviewed and documented in the test plan.

3-7 PIPING

3-7.1 Piping Arrangements

Piping arrangements required to conduct a test under this Code are detailed in Section 4. Permissible alternatives are described for convenience and suitability. A selection suitable for the prevailing test conditions shall be made and described in the test report. The design of the test loop shall allow steady-state operation for all specified test points.

3-7.2 Straight Lengths of Piping

Minimum straight lengths of piping at the inlet, discharge, and on both sides of the flow-measuring device are specified in Section 4.

NOTE: When compressors are treated as a number of individual sections, these piping requirements apply to each section. Such piping between sections may not occur naturally in the design. When it does not, the parties to the test should elect by mutual agreement to
(a) install additional piping between the sections.
(b) take measurements in the available space. Consideration shall be given to any compromise in measurement accuracy and its effect on the final test objective.
(c) remove components such as external heat exchangers and replace them with the required piping. When this alternative is selected, it is important that the removal of the component have a negligible effect on the section entry or exit flow field so as not to affect the section performance parameters.

3-7.3 Intercooler Performance and Pressure Drop

Where external intercooler performance and pressure drop are known for the specified operating conditions, or are determined on a separate test, the compressor may be tested as separate sections, and the combined performance may be computed by the method described in Section 5.

3-8 INSTRUMENTATION

Test instruments shall be selected, calibrated, and installed in accordance with the requirements of Section 4.

3-9 PRELIMINARY TEST

3-9.1 Pretest Inspection

If pretest inspection is of interest to either party, it shall be conducted per the requirements of ASME PTC 1 and paras. 3-9.1.1 through 3-9.1.3.

3-9.1.1 The compressor shall be operated for sufficient time at the required conditions to demonstrate acceptable mechanical operation and stable values of all measurements to be taken during the test.

3-9.1.2 All instrument observations pertinent to the test shall be taken during the preliminary test.

3-9.1.3 A set of calculations shall be made using the preliminary test data to ensure that the correct test speed has been selected, that the test parameters required in Table 3-2.1-1 or Table 3-2.1-2, as applicable, were obtained, and that the overall performance values are reasonable.

3-10 TEST OPERATION

3-10.1 Stabilization Time

The compressor shall be operated for a sufficient period of time at each test point to demonstrate that all variables have stabilized (see subsection 3-11).

3-10.2 Test Readings

When all variables have stabilized, the first set of readings of all essential instruments shall be taken (see para. 5-4.3). Three or more sets of readings shall be taken during each test point.

3-10.3 Test Point Duration

The minimum duration of a test point, after stabilization, shall be 5 min from the start of the first set of readings to the end of the last reading.

3-10.4 Two-Point Test

When a test is only to verify a single specified condition, the test shall consist of two test points that bracket the specified flow coefficient within a range of 96% to 104%.

3-10.5 Multipoint Test

If performance curves are required to verify the complete compressor range of operation and one specified point is required, then a multipoint test shall be performed as per paras. 3-10.5.1 and 3-10.5.2.

3-10.5.1 A minimum of five points shall be used to complete the performance curve at the fixed test speed calculated for the specified point and/or the fixed vane settings. A point shall be taken at the specified flow coefficient. The additional points shall consist of at least one point near surge, one point in the overload range (preferably 105% or greater of the specified flow coefficient), and two points spaced approximately equally between the previously defined measurement points to complete the performance curve.

3-10.5.2 If performance curves are required to verify the complete compressor range of operation and more than one specified point is required, then the parties shall agree which specified points require a multipoint test per para. 3-10.5.1 and which specified points, if any, require a test per para. 3-10.4. A multipoint test shall be performed per para. 3-10.5.1 for at least one specified point.

3-10.6 Determination of Surge

The flow at which surge occurs can be determined by slowly reducing the flow rate at the test speed until indications of unstable or pulsating flow appear. When the surge flow has been identified, the flow should be increased slightly until stable operation is restored so that a complete set of performance data may be taken. This process shall be repeated a second time to demonstrate the repeatability of the initial setting.

NOTES:
(1) The severity of surge will vary widely as a function of gas density, pressure ratio, type of compressor, and capacitance of the piping system.
(2) Surge may be identified by noise, fluctuations in the differential pressure of the flow nozzle, or a drop and/or fluctuation of the pressure and/or temperature.
(3) It should be understood that a surge flow established in a shop test may not define the surge conditions that will occur in the field due to differences in piping configuration and system response.

3-10.7 Maximum Capacity Determination

The maximum capacity can be determined by gradually opening the discharge throttle valve while maintaining speed, guide vane setting, and inlet pressure until the flow remains essentially constant with decreasing discharge pressure.

3-10.7.1 If the compressor is to be tested with an open discharge, the maximum capacity may be determined by gradually opening the inlet valve while holding speed, guide vane setting, and discharge pressure constant.

3-10.7.2 If maximum capacity is to be determined, the test facilities shall be designed so as not to limit maximum flow.

3-11 TEST STABILIZATION

3-11.1 Test Readings

Stabilization of the test variables shall demonstrate that any small variations of the test readings do not result in unacceptable changes of the calculated results, primarily polytropic work and efficiency.

3-11.2 Calculated Values

Stabilization is demonstrated by a series of calculated values of polytropic efficiency, with the difference between its maximum and minimum values not exceeding 0.005 over a minimum period of 5 min.

3-12 INCONSISTENCIES

3-12.1 Outliers

Where four independent instruments are used to measure an individual data point value such as a pressure or temperature, and any recorded observation is inconsistent as determined by the outlier method described in ASME PTC 19.1, Nonmandatory Appendix A, the data point value shall be determined from the average of the remaining observations. See para. 5-4.1.

3-12.2 Fluctuation Tolerances

All readings for each test point shall be within the fluctuation tolerances listed in Table 3-12.2-1. Readings exceeding the stated tolerances shall be discarded; however, removing outliers typically avoids this situation.

3-13 ERRORS AND UNCERTAINTIES

3-13.1 Uncertainty Analysis

It should be recognized that the results of the test calculations are subject to error caused by the inaccuracies of the test instrumentation systems and/or procedures. It is recommended that an uncertainty analysis be made prior to the test to ensure that the test objectives can be met. The detailed procedures are given in Section 7 and ASME PTC 19.1.

3-13.2 Test Quality

The uncertainty is a measure of the quality of the test and should not be used as a measure of the quality of the machine.

Table 3-12.2-1
Permissible Fluctuations of Test Readings

Measurement	Symbol	SI Units	U.S. Customary Units	Fluctuation, %
Inlet pressure	p_i	Pa	psia	2
Inlet temperature	T_i	K	°R	0.3
Discharge pressure	p_d	Pa	psia	2
Inlet volumetric flow	Q	m^3/s	ft^3/min	0.5
Speed	N	1/s	rpm	0.5
Torque	T	N·m	ft-lbf	0.5
Electric motor input	P	W	hp	1
Molecular weight	MW	kg/kmol	lbm/lbmole	[Notes (1), (2)]
Cooling water inlet temperature	T	K	°R	0.3
Line voltage	...	Volts	volts	2

GENERAL NOTES:
(a) See para. 5-4.3.3.
(b) A fluctuation is the percent difference between the minimum and maximum test readings divided by the average of all acceptable readings.
(c) Permissible fluctuations apply to Type 1 and Type 2 tests.

NOTES:
(1) The allowable fluctuation for the application of specific gravity meters is 0.25%.
(2) See para. 4-9.3 regarding gas chromatographs.

3-14 TEST LOG SHEETS

The test log sheet shall identify the compressor manufacturer, model, and serial number. Test location, driver identification, test instruments used, test date, and time shall be listed. Raw data as observed for each test point shall be recorded on the test log sheet as well as the time of each set of data. Corrections and corrected readings shall be listed separately in the test report.

At the completion of the test, the log sheets shall be signed by the representatives of the interested parties. Copies of the complete log sheets shall be furnished to the interested parties. The test report shall be completed in accordance with the instructions in Section 6.

Section 4
Instruments and Methods of Measurement

4-1 METHODS

The choice of methods provided in this Code depends on the compressor, the specified gas, and the type of test selected.

4-2 INSTRUMENTATION

The Performance Test Code Supplements in the ASME PTC 19 series on instruments and apparatus provide authoritative information concerning instruments and their use and should be consulted for such information. The selection of instrumentation shall be determined by the uncertainty limit requirements of the test as well as suitability for the test site conditions. The instrument selection shall be justified by calculation that the uncertainty in results meets the stated test objectives.

Instrumentation is required to determine the inlet and discharge gas states, flow rates, compressor speeds, and adjustable guide vane positions. Depending on the method selected, additional instrumentation may be required to determine test power.

4-3 PIPING

4-3.1 Pressure- and Temperature-Measuring Stations

The locations of the pressure- and temperature-measuring stations have specific relation to the compressor inlet and outlet openings. The pipe sizes shall match these openings. Minimum lengths of straight pipe are mandatory for certain pressure and temperature measurement stations and for certain flow devices. Pipe arrangements and allowable exceptions are described in this Section. Appropriate selections shall be made and described in the test procedure and report.

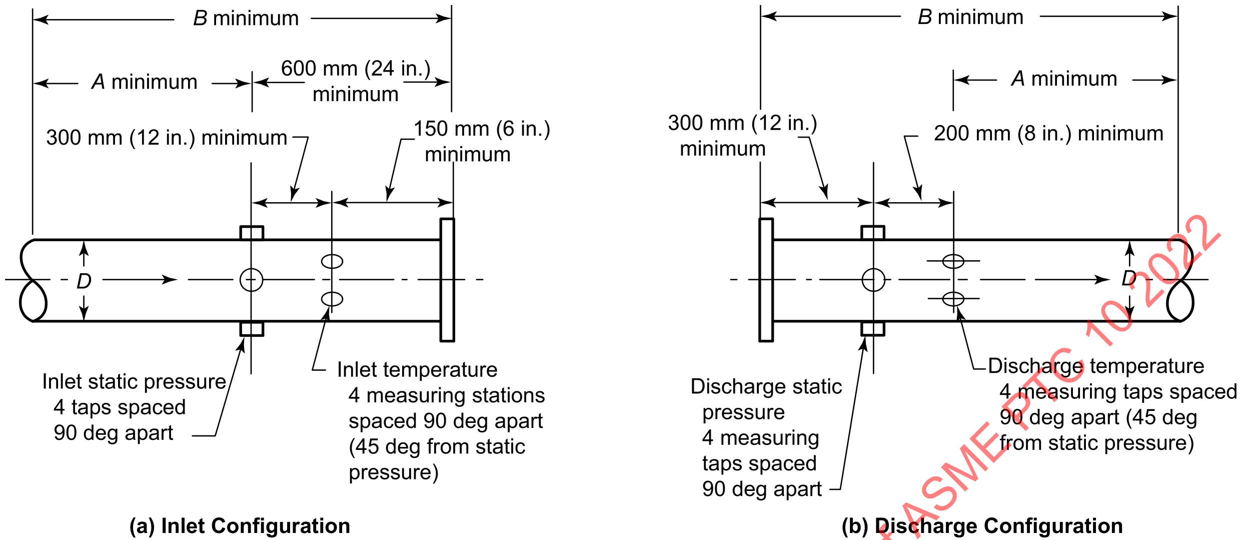
4-3.2 Inlet Piping

Typical inlet piping required for compressors is outlined in Figure 4-3.2-1. The minimum straight length of inlet pipe is determined by what is upstream of the inlet opening. The four static pressure taps shall be a minimum of 600 mm (24 in.) upstream of the inlet opening. Downstream of the pressure taps are four temperature taps displaced 45 deg from the pressure taps and at least 300 mm (12 in.) downstream.

In special cases when atmospheric conditions satisfy the requirements, the compressor may be run without an inlet pipe as shown in Figure 4-3.2-2. The inlet opening shall be protected with a screen and bellmouth suitably designed to eliminate debris and minimize entrance losses (see subsection 4-4). The total inlet pressure is equal to atmospheric pressure. Temperature-measuring devices shall be located on the screen to measure the temperature of the air stream at the compressor inlet.

For compressors with an axial inlet, the impeller may, under some conditions, produce a vortex at the pressure station, which could cause substantial error in the measurement of inlet pressure. Users of this Code, by agreement, may use vanes suitably designed for low pressure loss to prevent swirl at the pressure taps. The static pressure stations shall not be less than four pipe diameters upstream of the compressor flange as shown in Figure 4-3.2-3.

Figure 4-3.2-1
Inlet and Discharge Configuration



Inlet Opening Preceded by	Minimum Dimension *A*	Minimum Dimension *B*	Discharge Opening Followed by	Minimum Dimension *A*	Minimum Dimension *B*
Straight run	2*D*	3*D*	Straight run	2*D*	3*D*
Elbow	2*D*	3*D*	Elbow	2*D*	3*D*
Reducer	3*D*	5*D*	Reducer	3*D*	5*D*
Valve	8*D*	10*D*	Valve	3*D*	5*D*
Flow device	3*D*	5*D*	Flow device	8*D*	10*D*

GENERAL NOTES:
(a) For open inlet, see Figure 4-3.2-2. For open discharge, see Figure 4-3.3-2.
(b) For vortex-producing axial inlet, see Figure 4-3.2-3. For diffusing volute with unsymmetrical flow, see Figure 4-3.3-1.

Figure 4-3.2-2
Open Inlet

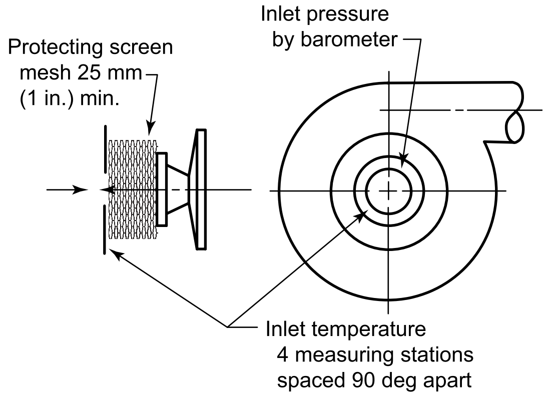
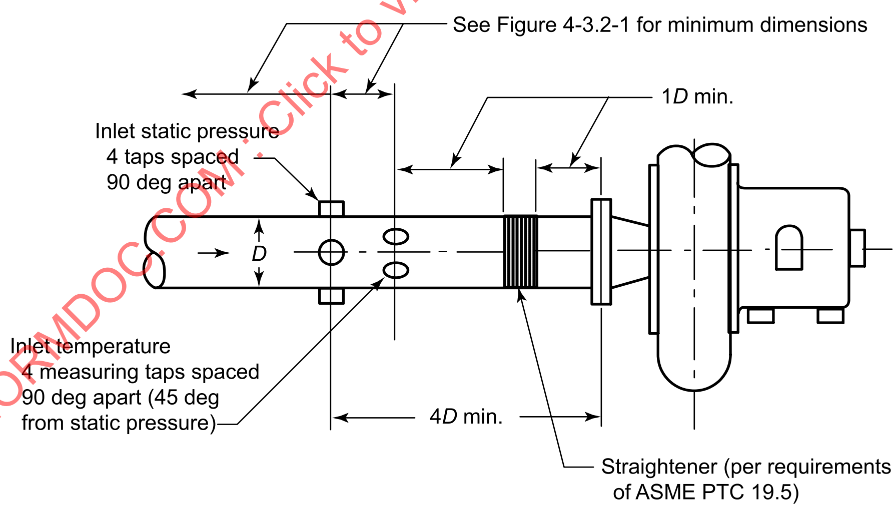


Figure 4-3.2-3
Vortex-Producing Axial Inlet



4-3.3 Discharge Piping

Typical discharge piping required for compressors is outlined in Figure 4-3.2-1. The minimum straight length of discharge pipe required before and after the instrumentation is specified. The four static pressure taps are a minimum 300 mm (12 in.) downstream of the discharge opening. The pressure taps are followed by the four temperature taps displaced 45 deg from the pressure taps and at least 200 mm (8 in.) downstream.

When the compressor has a volute that produces unsymmetrical flow at the discharge opening, the static pressure taps shall be a minimum of six diameters downstream as shown in Figure 4-3.3-1. The other minimum dimensions are specified in Figure 4-3.2-2.

An open discharge without pipe is shown in Figure 4-3.3-2.

Figure 4-3.3-1
Diffusing Volute Discharge With Nonsymmetric Flow

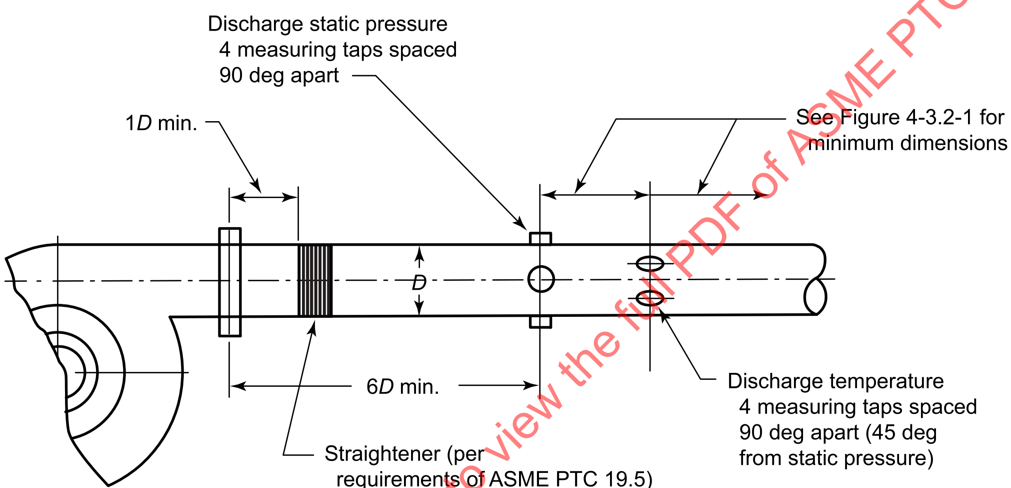
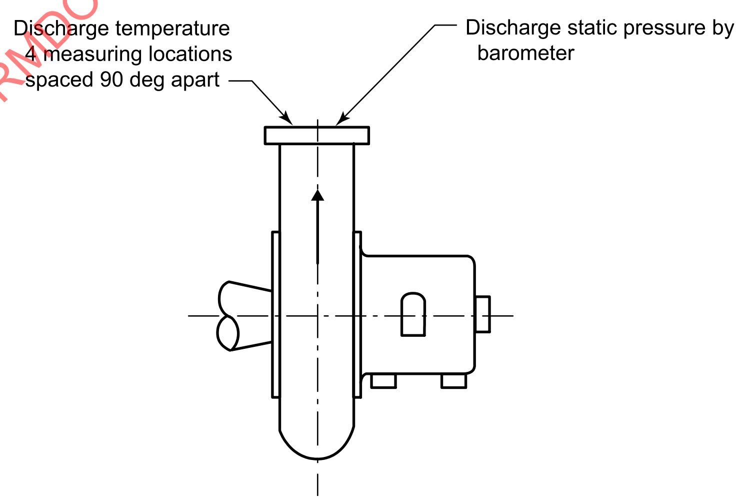


Figure 4-3.3-2
Open Discharge



GENERAL NOTE: When discharge velocity pressure exceeds 5% of total pressure, use discharge pipe arrangement. See Figure 4-3.2-1.

4-3.4 Typical Piping Arrangement

Figures 4-3.4-1 and 4-3.4-2 show a typical arrangement for testing with a general closed loop and closed loop with sidestreams.

Figure 4-3.4-1
Typical Closed Loop

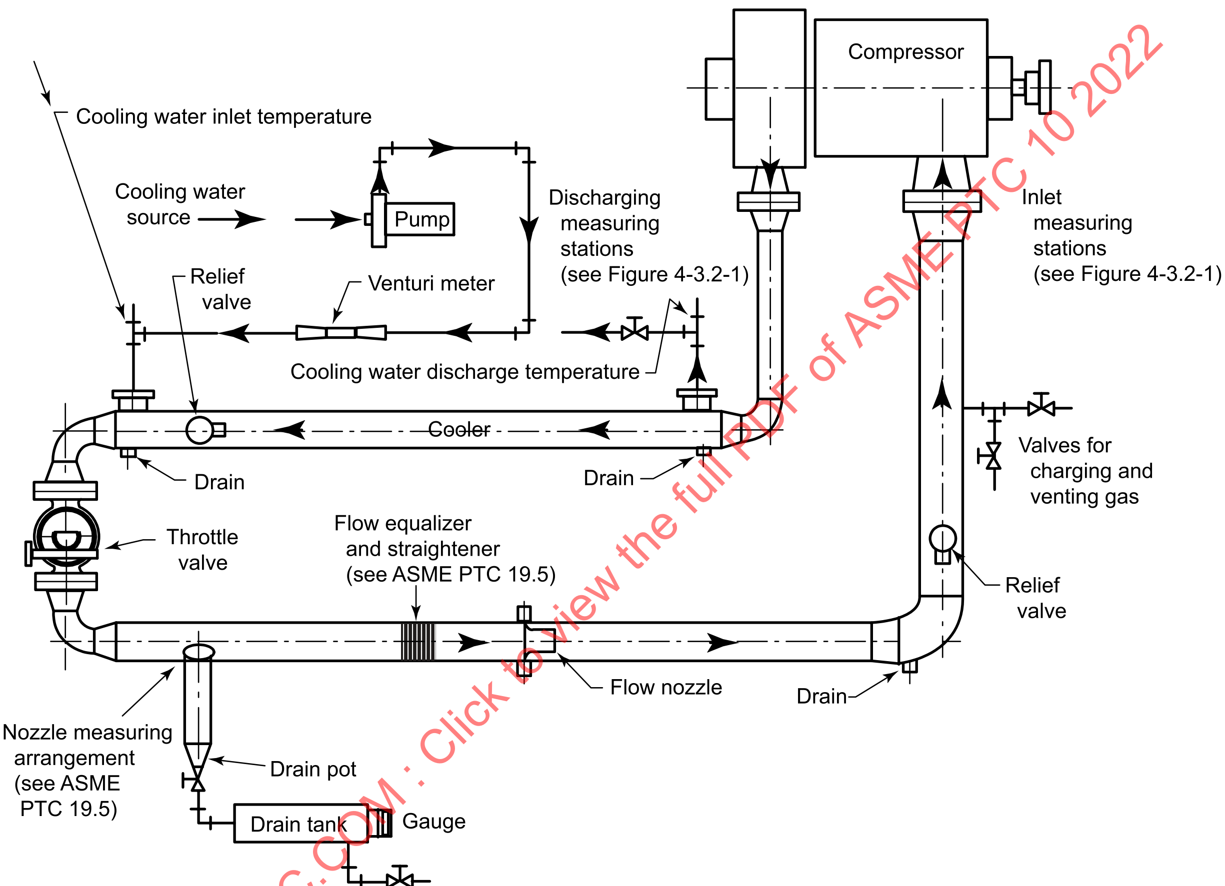
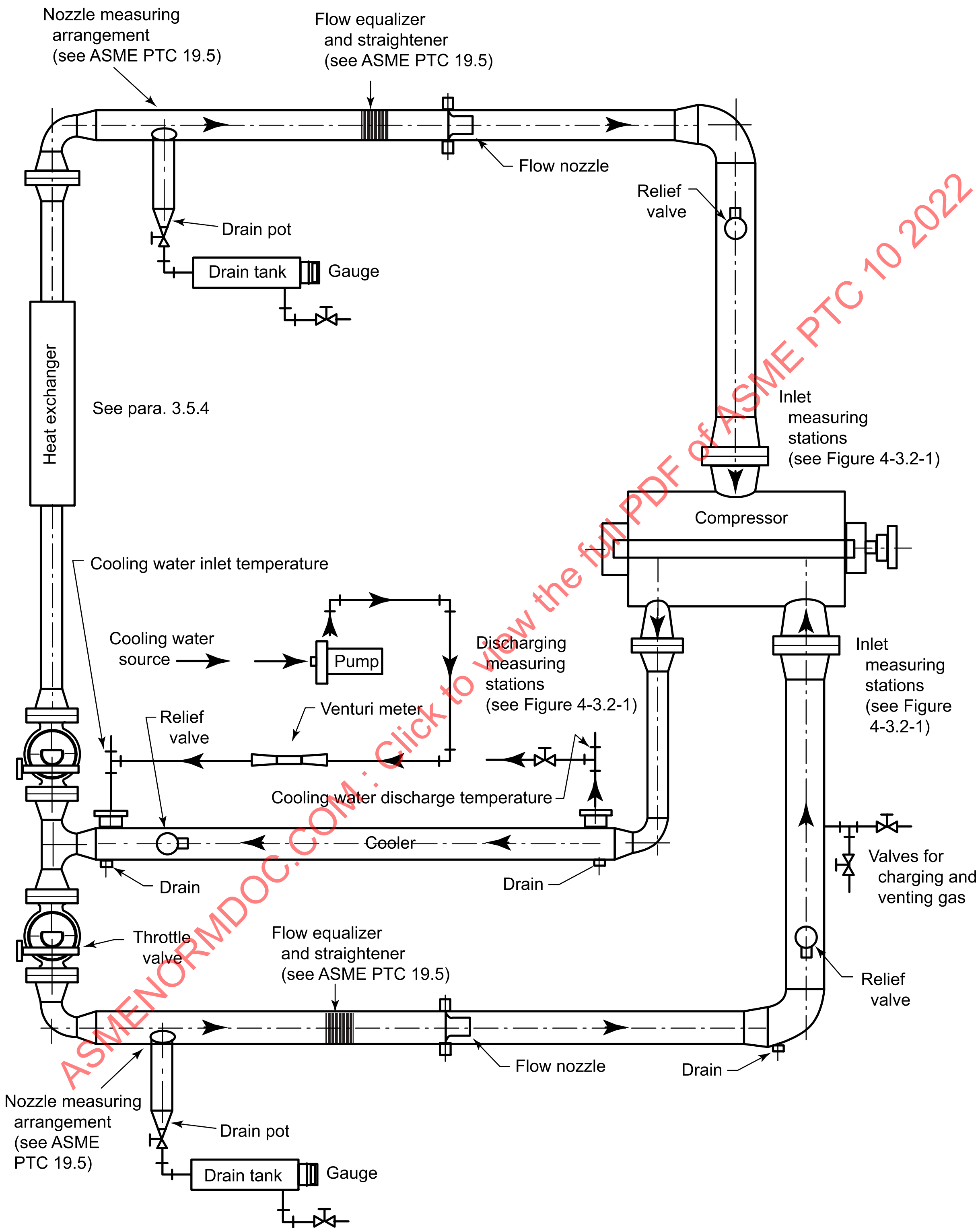


Figure 4-3.4-2
Typical Closed Loop With Sidestream



4-4 PROTECTIVE SCREENS

Compressors operating with an open inlet shall be protected with a screen or filter that is suitable for the conditions. In general, a screen on the inlet shall be strong enough to prevent collapse in the event of accidental clogging. The mesh of a screen shall be selected to prevent entry of foreign matter that might damage the compressor and impair its performance. Reliable tests cannot be made on atmospheric air laden with dust, oil-fog, paint spray, or other foreign matter that may foul the flow passage of the compressor. Protective screens shall have an open area at least two times that of the compressor inlet or the nozzle pipe. When screens with very small mesh or filters are used, inlet pressure shall be measured by static taps as provided in Figure 4-3.2-1 for straight pipe. Where screens or filters are used in a closed loop, precautions such as measurement of the differential pressure are recommended.

4-5 FLOW CONDITIONERS

Flow conditioners (e.g., flow straighteners and equalizers) shall be installed per the requirements of ASME PTC 19.5.

4-6 PRESSURE MEASUREMENTS

4-6.1 Pressure Instrumentation

Instrumentation to measure pressure shall comply with the requirements provided in ASME PTC 19.2 unless otherwise modified in this Code. See Table 3-12.2-1 for permissible fluctuations in test measurements.

4-6.2 Gauge Lines

Gauge lines shall be designed to remain dry (i.e., self-draining or heated and insulated) or to hold a consistent and predictable level of liquid. The measured pressure shall be compensated to account for the liquid column.

4-6.3 Transducers

Transducers shall be selected with pressure ranges appropriate for the expected test pressures.

4-6.4 Operational Stability

Operational stability shall be verified as part of the measurement system operating procedures for any automated data collection equipment. See para. 3-10.1.

4-6.5 Dynamic Pressure

Dynamic pressure shall be computed based on the average velocity. See para. 5-4.3.5.

4-6.6 Raw Data Observations

Individual raw data observations of static pressure shall be recorded from four stations spaced 90 deg apart in the same plane, perpendicular to the flow in the pipe. See para. 5-4.1 for processing of static pressure raw data observations. Total pressure probes may be used to measure pressure at the same stations at which the static measurements are made.

4-6.7 Inlet Pressure Measurement

Inlet pressure is the total pressure prevailing at the inlet of each section. Inlet pressure is the sum of the static pressure and the dynamic pressure. Static pressure shall be measured as specified for inlet pipes in Figure 4-3.2-1 or Figure 4-3.2-3. Where no inlet pipe is used, as in Figure 4-3.2-2, the inlet total pressure shall be measured by a barometer.

4-6.8 Discharge Pressure Measurement

Discharge pressure is the total pressure prevailing at the discharge of each section. Discharge pressure is the sum of the static pressure and the dynamic pressure. Static pressure shall be measured as illustrated in Figure 4-3.2-1. When no discharge pipe is used, as illustrated in Figure 4-3.2-2, the discharge static pressure shall be measured by a barometer. If the dynamic pressure (based on discharge opening area) exceeds 5% of the static pressure, an open discharge shall not be used.

4-6.9 Total Pressure Measurement

Total pressure may be directly measured by the use of total pressure probes inserted into the flow stream (such probes shall be properly oriented or directionally compensated to ensure proper measurement). The total pressure measurement may be affected by the gas velocity at the probe versus the average gas velocity at the measurement station. In the event of significant unresolved differences from the total pressure deduced from the static pressure and average velocity, the static pressure–based result shall prevail.

4-6.10 Ambient Pressure and Temperature

Ambient pressure and temperature shall be recorded at the beginning and end of each test point. The measuring instrument shall be located at the site of the test and shall be protected from weather, direct sunlight, and fluctuating ambient conditions. Precautions shall be taken to prevent a pressure drop near the barometer, which may be caused by strong winds, compressor intakes, or ventilating fans.

4-6.11 Internal Pressure Measurements

When the parties to the test require the section performance to be based on internal pressure measurements, the parties shall agree to the needed internal pressure measurement locations and the manner in which the measurements will be taken.

NOTE: Due to the many configurations of the internal passages in sidestream compressors, this Code cannot specify precisely where or how internal pressure instrumentation may be placed. As a guide, multiple pressure probes (either static or total) should be inserted in the mainstream flow. These probes should be located so the incoming sidestream does not affect the raw data (see Figure 4-7.6-1). It may be difficult to make accurate internal pressure measurements at a stage discharge since this is normally a region of high velocity with local variations of velocity, flow angle, and pressure. This measurement uncertainty should be reflected in the error analysis and in the value of the uncertainty assigned to these stations.

4-7 TEMPERATURE MEASUREMENTS

4-7.1 Temperature Instrumentation

Instrumentation to measure temperature shall comply with the requirements provided in ASME PTC 19.3 unless otherwise modified in this Code. See Table 3-12.2-1 for permissible fluctuations in test measurements. Temperature shall be measured by thermocouples, resistance temperature detectors (RTDs), thermistors, or other devices with equivalent accuracy. The range of the devices' scales, sensitivity, and required accuracy shall be chosen for each of the significant measurements according to the particular need.

The temperature-measuring devices shall extend a sufficient distance into the fluid stream to minimize unavoidable conduction of heat. The devices need not be perpendicular to the wall.

Precaution shall be taken to avoid insertion of the temperature-measuring device into a stagnant area when measuring the temperature of a flowing medium.

NOTE: The following general precautions are recommended when making any temperature measurement:

(a) The instrument installation should ensure that the effects of radiation, convection, and conduction between the temperature-sensitive element and all external thermal bodies (pipe wall, external portions of thermowells and thermocouple sheaths, etc.) shall have a negligible effect on the temperature reading.

(b) Insulation of those parts of a thermowell, thermocouple sheath, etc., that extend beyond the pipe outside diameter may be a means of accomplishing the objective in (a).

(c) In some cases, insulation of the pipe wall near the thermowell or insulation of the section of the pipe upstream of the thermowell may be necessary. Refer to ASME PTC 19.3 and ASME PTC 19.3 TW for more information.

4-7.2 Thermocouples

Junctions of thermocouples shall be silver brazed or welded. The selection of materials shall be suitable for the temperature and the gases being measured. Calibration shall be made with the complete assembly, including the instrument, the reference junction, and the lead wires. If the well is integral with the thermocouple, the well shall also be included in the calibration.

4-7.3 Total Temperature Measurement

Total temperature is the sum of static temperature and dynamic temperature. Normally, the actual temperature measured is a value between static temperature and total temperature. The dynamic temperature is then corrected for the recovery factor and added to the measured observation (see para. 5-4.4). Special temperature probes made to measure total temperature may need little or no correction.

4-7.4 Inlet Temperature Measurement

Inlet temperature is the total temperature prevailing at the inlet of each section. When the compressor is tested with an inlet pipe, four temperature probes shall be spaced 90 deg apart in the same plane and displaced 45 deg from the static pressure sensors (see Figure 4-3.2-1 or Figure 4-3.2-3). When machines are assembled with an open inlet as in Figure 4-3.2-2, inlet total temperature is the atmospheric temperature, and it shall be measured by four instruments attached to the protecting screen. Large variations caused by factors other than instrument error, such as design, may require more than four measuring stations.

4-7.5 Discharge Temperature Measurement

Discharge temperature is the total temperature prevailing at the discharge of each section. When a compressor is assembled for test with a discharge pipe, the instruments shall be located as shown in Figure 4-3.2-1 or Figure 4-3.3-1 and spaced 90 deg apart in the same plane and displaced 45 deg from the pressure taps. Where the compressor is operated without a discharge pipe, four instruments shall be anchored to the discharge opening with a suitable projection into the gas stream.

Large variations caused by factors other than instrument error may require more than four measuring stations.

4-7.6 Internal Temperature Measurements

For inward sidestream compressors, the only internal measurements required are the sectional discharge temperatures for computing the mixed temperature conditions and sectional performance (see para. 3-5.3.2). The parties shall agree to the needed internal temperature measurement locations and the manner in which the measurements will be taken.

NOTE: Due to the many configurations of the internal passages in sidestream compressors, this Code cannot specify where or how internal temperature instrumentation may be placed.

As a guide, multiple temperature probes should be inserted in the mainstream flow. These probes should be located so that the incoming sidestream does not affect the raw data (see Figure 4-7.6-1). It may be difficult to make accurate internal temperature measurements at a stage discharge since this is normally a region of high velocity. This measurement uncertainty should be reflected in the uncertainty analysis and in the value of the uncertainty assigned to these stations.

4-8 CAPACITY MEASUREMENTS

4-8.1 Flow Measurement Instrumentation

Instrumentation to measure flow shall comply with the requirements provided in ASME PTC 19.5 unless otherwise modified in this Code. See Table 3-12.2-1 for permissible fluctuations in test measurements. Flow may be measured by using a flow nozzle, concentric orifice, Venturi tube, Coriolis meter, or alternative devices of equal or better accuracy. The interested parties shall mutually agree on the type of metering device to be used. The choice and details shall be stated in the test procedure and report.

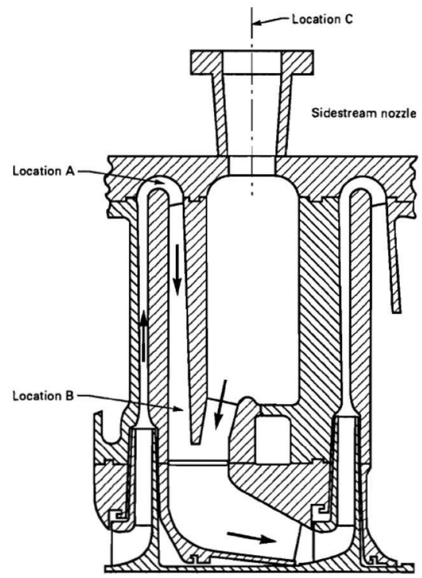
4-8.2 Flow-Measuring Device for Flow Sections

The flow-measuring device may be located on either the inlet side or the discharge side of the compressor. The device shall be used to determine the capacity (see subsection 2-5), which excludes losses by shaft leakage, balancing pistons, condensation, and any specific leakage that may be inherent in the compressor design. Multiple devices shall be used for multiple inlet or discharge flow sections.

4-8.3 Open Inlet Flow Nozzle

If a flow nozzle is used with an open inlet, a protecting screen shall be used in accordance with the instructions of subsection 4-4. Upstream total pressure is equal to the barometric pressure.

Figure 4-7.6-1
Typical Inward Sidestream Cross Section



GENERAL NOTE: Mainstream temperature and, if applicable, pressure instrumentation shall be installed between locations A and B for inward flowing sidestream machines. Extraction sidestream machine temperature and pressure measurement shall be installed at location C.

4-8.4 Open Discharge Flow Nozzle

A flow nozzle may be used with an open discharge where it is permitted to discharge the gas to atmosphere (see subsection 3-6).

4-9 GAS COMPOSITION

4-9.1 Gas Composition Evaluation

The gas composition measurement shall be taken during the performance test at least three times as follows:

(a) before starting the test.

(b) after completing the test.

(c) before performing any measurement for the specified point. In the case where the gas is made up of three or fewer components, consistency of gas composition during the test can be proven by the use of a specific gravity meter measurement for the specified point.

Gas composition shall be evaluated through either automatic gas chromatograph (see para. 4-9.3) sampling or by taking regular gas samples.

4-9.2 Gas Composition Evaluation for Performance Curve

If it is required to develop the performance curve, there is no need to perform the gas composition evaluation for any test point other than the specified point at any fixed speed or fixed vane setting (see para. 3-10.5). Deviations of specific gravity between test readings shall be within 0.25%.

4-9.3 Gas Chromatographs

Use of gas chromatographs to determine test gas composition and molecular weight is subject to a number of variables associated with the specific design of the instrument and sampling system. Measurement uncertainty and repeatability shall be demonstrated through measurements of highly accurate samples of similar composition to the test gas prior to and after completion of the performance test.

4-9.4 Test Loop Design

Except for open-loop air tests, the design of the test loop shall prevent air ingress and changes in gas composition during the test.

4-9.5 Test Loop Condensation

Precautions shall be taken when testing with a closed loop to eliminate all liquids from the gas stream and instrumentation impulse lines. When dealing with gas mixtures subject to variation, samples shall be taken at each test point and shall be analyzed by spectrographic or chromatographic methods. The sample shall be taken from the piping such that there is no condensation before the compressor or the sampling points. This analysis shall consist of identification of the constituents, a measure of mole percent of each, and evaluation of the molecular weight. If the test gas is air, no samples are necessary. However, relative humidity or dewpoint shall be measured during each test point.

Instrument lines shall be designed to ensure the effect of potential condensation is prevented.

NOTE: Although the gas under test conditions may not exhibit condensation, the gas in the instrument lines may be cooler (i.e., room temperature) and, under some conditions, condensation could occur.

4-10 SPEED MEASUREMENT

4-10.1 Continuous Speed Measurement

Instruments shall be selected to provide a continuous indication of speed measurement, where variable-speed drivers are used.

4-10.2 Speed Measurement Instrumentation

The speed of rotation shall be measured using magnetic speed pickups, shaft encoders on the machine rotor, or the key phasor probes of the compressor.

4-11 TIME MEASUREMENT

The date and time of day at which test readings are taken shall be recorded on all data records.

4-12 METHODS OF SHAFT POWER MEASUREMENT

4-12.1 Shaft Power Input

The shaft power input at the compressor coupling or the drive shaft may be measured directly by torque meters or evaluated from the following:

(a) measurement of electrical input to a driving motor

(b) a heat balance method

NOTE: A direct shaft measurement method may be used as an independent measurement of absorbed power to verify shaft power calculated from other measurement methods.

4-12.2 Power Measurement Methods

The precautions, limitations, and permissible applications for each of these methods are described separately. The selected method of shaft power measurement used shall be by mutual agreement between the parties involved in the testing and shall be documented in the test procedure and report.

4-13 SHAFT POWER BY TORQUE MEASUREMENTS

Torque may be directly measured by devices installed in a drive shaft interposed between the driver and the compressor. For tests under this Code, torque meters shall be a type suitable for calibration. The torsion member shall be selected for readability and accuracy at the speed and load prevailing during the test.

4-14 SHAFT POWER BY ELECTRICAL MEASUREMENTS

4-14.1 Motor Shaft Power

The shaft power input to a motor-driven compressor may be computed from measurements of the electrical input to the motor terminals under certain conditions. The power requirement of the compressor should be more than 75% of the motor rating. The output of a motor shall be calculated by subtracting losses from the measured electrical input or as the product of input and motor efficiency. Motor efficiency shall be determined by an input–output test, where output is measured on a calibrated dynamometer or other appropriate device. For motor efficiency determination, the supply line voltage used for calibration shall be the same as that used for the compressor test.

4-14.2 Motor Efficiency Determination

Motor efficiency determination by input–output measurements may not be practical for large motors. For large motors, the loss method may be used. The segregated losses of an induction motor shall include friction and windage, core loss, I^2R loss of the rotor and the stator, and a load loss in accordance with applicable standards such as IEEE 112 for induction motors and IEEE 115 for synchronous motors.

4-14.3 Motor Power Input Measurement

The electric power input to the motor shall be measured by the instruments connected at the motor terminals. The detailed instructions for the measurement of electrical power are as given in ASME PTC 19.6 and IEEE 120. The indicating electric meters should be selected to read above one-third of the scale range.

4-14.4 Transformers

Calculations of electrical power shall include calibration corrections for the meter and current transformers. The transformers shall be measured for ratio and phase angle at the load conditions prevailing during the test.

4-15 SHAFT POWER BY HEAT BALANCE MEASUREMENTS

4-15.1 Shaft Power Computation

Shaft power may be computed from measured values of the flow rate, gas properties at inlet and discharge [and internal locations when required by this Code (see para. 4-7.6)], heat exchange through the casing, mechanical losses, and gas leakage loss from the shaft seals.

4-15.2 Mechanical Losses

Methods to account for mechanical losses are discussed in subsection 4-17. External heat loss from the casing shall be evaluated in accordance with subsection 4-16.

4-15.3 Precautions and Limitations

The heat balance method shall be used with the following precautions and limitations:

(a) When the temperature rise is less than 28°C (50°F), the inlet and discharge temperatures shall be measured with instruments suitably selected and applied to provide combined accuracy within 1% of the temperature rise.

NOTE: Consideration should be given to direct measurement of the temperature rise (such as with differential thermocouples).

(b) Evidence of nonuniform temperature distribution more than 2% of the temperature rise at either the inlet or the discharge measurement station requires one of the following procedures be used at the offending measurement station:

(1) Apply insulation to the piping upstream of the temperature measurement station to minimize thermal gradient. If successful, the temperature measurement installation need not be changed.

(2) Move the temperature measurement station away from the compressor and add pipe insulation. This might be particularly effective when temperature stratification causes the problem at a compressor discharge.

(3) Measure temperature at multiple locations along each of two diametral lines spaced 90 deg apart at the same pipe cross section. The measured temperature is the average of the individual measurements (see para. 5-4.3 for processing of raw data).

(c) In sidestream machines, where internal temperature measurements are to be made, four locations should be used where practical. Determination of the number of probes or measurement stations is dependent on the geometry of the compressor.

Definition of the number, type, and location of required internal instrumentation shall be defined in the test agenda. In all cases, the upstream temperatures of the two streams mixing internally shall be measured. See para. 3-5.3.1 for requirements on temperature stratification.

NOTE: A measurement of the downstream mixed temperature would be unreliable and should not be used for calculation purposes due to inherent poor internal mixing conditions in a machine.

(d) The heat losses due to radiation and convection expressed as a percent of total tested shaft power shall not exceed 5% (see subsection 4-16).

4-16 HEAT LOSS

4-16.1 Heat Loss Minimization

When using the heat balance method to determine power, heat loss may be minimized by the application of a suitable insulating material. If the compressed gas temperature rise is less than 28°C (50°F), the inlet piping, compressor casing, and exit piping shall be insulated at least to the measuring station.

NOTE: The external heat loss from the compressor section casing and relevant connecting piping may be computed with acceptable accuracy from measurements of the exposed surface area, the average temperature of the surface, and the ambient temperature near the surface. Where a hot surface temperature varies widely, as in large multistage compressors, it is advisable to divide the casing into arbitrary sections and determine the area and temperature of each separately, thus obtaining an approximate integrated average temperature for the section's total surface area.

4-16.2 Cooling Fluid Measurements

Where cooling occurs between the discharge and inlet of measuring stations as part of the compressor design, measurement of temperatures and flow rates of the specified cooling fluids are required. Examples are compressors incorporating interstage coolers or aftercoolers as part of the compressor package being tested.

4-17 MECHANICAL LOSSES

4-17.1 Heat Produced by Mechanical Losses

When practical, the heat produced by the mechanical losses (e.g., integral gears and bearings) shall be determined from the temperature rise of the lubricating fluid or cooling fluid. The quantity of fluid flowing shall be determined by calibrated flowmeters. The mechanical losses and the frictional loss in the seals, if used, shall be determined and included in the total mechanical losses. Where the mechanical losses are well known and documented, the calculated values or the values determined from prior testing may be used by agreement by the test parties.

4-17.2 Gear Losses

Where speed-changing gears (not part of the compressor) are used between a driver and a compressor, and shaft power is measured on the input side of the gear, it is necessary to subtract the friction and windage loss of the gear to obtain the shaft power input to the compressor. The gear power loss to the lubricating fluid may be determined by measuring the flow rate and the temperature rise. The additional external loss to the atmosphere may be determined by the methods of subsection 4-16. When gear loss measurements are made on an independent gear test, care should be taken to ensure that the load, lubricating oil temperature, viscosity, and flow rates are similar to those for the compressor test.

4-18 INSTRUMENT CALIBRATION AND UNCERTAINTY

4-18.1 Calibration

All instruments used for measurement shall be currently certified by comparison with any applicable standard before the test. Those instruments subject to change in calibrations due to use, handling, or exposure to injurious conditions shall be compared again with standards after the test. Due consideration shall be given to temperature, humidity, lighting, vibration, dust control, cleanliness, electromagnetic interference, and other factors affecting the calibration. Where pertinent, these factors shall be monitored and recorded, and, as applicable, compensating corrections shall be applied to calibration results obtained in an environment that departs from acceptable conditions.

Table 4-18.2-1
Typical End-to-End Measurement Uncertainty

Measurement	Uncertainty
Pressure	0.25%
Temperature	0.3°C (0.5°F)
Volume flow	0.5%
Gas molecular weight (for specific gravity meters, also see para. 4-9.3 for gas chromatographs)	1.0%
Torque	1.0%
Speed	0.1%
Relative humidity (for open-loop air tests)	2.0%

The number of calibration points depends on the magnitude of the measurement's sensitivity factor relative to the tested parameter. The calibration shall bracket the expected measurement values as closely as possible. All instruments should be calibrated such that the expected values are approached from both a higher value and a lower value. This approach will minimize hysteresis effects.

4-18.2 Measurement Uncertainty

Typical end-to-end measurement uncertainties for adequate precision are listed in Table 4-18.2-1. Further specification should be in accordance with ASME PTC 19.1. The uncertainty of test results may be evaluated based on specific test requirements for uncertainty analysis and improvements may be made when necessary.

4-18.3 Temperature Instrumentation Calibration

Temperature measurement devices shall be calibrated with certified standards at 20% intervals for the measurement range. The standard shall be suitable for the measurement range of the instruments to be calibrated. Procedures described in ASME PTC 19.3 shall be followed for checking the accuracy of temperature-measuring instruments.

4-18.4 Electrical Power Instrumentation Calibration

Instruments for measuring electric power such as watt meters, ammeters, and voltmeters shall be calibrated with primary standards. The zero adjustments shall be checked. The instruments shall be examined for pivot friction, and instruments showing pivot friction shall not be used. Dynamometer types may be calibrated on either alternating current (AC) or direct current (DC). Current transformers shall be measured for transformation ratio and phase angle at the range of burdens prevailing in the circuit during the test. The transformation ratio of potential transformers shall be measured at the approximate primary voltage and frequency prevailing during the test. Procedures described in ASME PTC 19.6 shall be followed.

4-18.5 Torque Meter Calibration

Torque meters shall be calibrated by applying torque with certified standard weights, load cells, or other appropriate devices spaced to cover the working range.

4-19 HUMIDITY MEASUREMENT

For open-loop tests, the moisture content of inlet air shall be measured directly with a hygrometer or indirectly by measuring the adiabatic wet-bulb temperature. The measurement location shall be downstream of any inlet-conditioning device and preferably in close proximity to the dry-bulb temperature measurement. The measurement location shall be shielded from direct sunlight.

4-20 TORQUE MEASUREMENT

Determination of torque with a torque meter may be used as an alternative to calculate test shaft power.

4-21 DATA ACQUISITION SYSTEM

4-21.1 Collection

Data acquisition shall be carried out in accordance with accepted practices and procedures as discussed in ASME PTC 19.22. A data collection system shall be designed to accept multiple instrument inputs and shall be able to sample and record data from all the instruments. Refer to para. 5-4.1. The data collection systems shall be time synchronized to provide consistent time-based data sampling and recording. All data acquisition systems shall have adequate frequency response and bandwidth for measurement of the desired parameter.

4-21.2 Processing

The data processing system shall have the ability to process each input collected during the test and to calculate test point values in accordance with Section 5.

4-21.3 Calibration

Methods for calibrating data acquisition systems shall be in accordance with ASME PTC 19.22, Section 5 and shall be recorded in the test report. Uncertainty of data acquisition systems shall be in accordance with ASME PTC 19.22, Section 6.

Section 5
Computation of Results

5-1 CALCULATION PROCEDURE

The process of establishing compressor performance from test data involves a number of calculation steps. This procedure is presented in the following order:

Step 1. Select and define an equation of state to be used to determine fluid thermodynamic properties.
Step 2. Select and define the polytropic computational method.
Step 3. Calculate the appropriate test speed and conditions if a Type 2 test is to be performed.
Step 4. Process the raw test data.
Step 5. Calculate the test performance.
Step 6. Express the test performance in a nondimensional form.
Step 7. Apply the Reynolds number corrections.
Step 8. Use the corrected nondimensional expressions to convert performance to specified operating conditions.

The important subject of uncertainty is treated separately. The format of this Section is intended to guide the user in the basic calculation procedure and to present the necessary equations. Nonmandatory Appendix E is provided as a background theory source with further explanation of the equations.

5-2 COMPUTATIONAL METHODS: CHOICE OF METHODS

Three methods to determine compressor performance parameters are defined in paras. 5-2.2 through 5-2.4. These methods shall be used for both Type 1 and Type 2 test conditions. Nonmandatory Appendix C provides example problems.

The choice of calculation method will impact the overall uncertainty of the test. Refer to Section 7.

5-2.1 Method Selection

The party conducting the test shall select and document the choice of one of the three calculation methods used for the performance calculations during the testing.

NOTE: See subsection 7-4 for test method uncertainties.

5-2.2 Sandberg–Colby Method

The Sandberg–Colby method requires only measured performance parameters at the compressor section inlet and discharge and the gas properties. Using only the values of gas specific enthalpy and entropy derived at inlet and discharge conditions along with the associated temperatures, the polytropic work and efficiency are calculated from the relations provided in Table 5-2.2-1.

Table 5-2.2-1
Sandberg–Colby Method Polytropic Relations

Performance Parameter	Relation at Test or Specified Conditions
Polytropic work	$w_p = (h_d - h_i) - \left(\frac{T_i + T_d}{2}\right)(s_d - s_i)$
Polytropic efficiency	$\eta_p = \frac{w_p}{(h_d - h_i)} = 1 - \left(\frac{T_i + T_d}{2}\right)\left[\frac{(s_d - s_i)}{(h_d - h_i)}\right]$

5-2.3 Huntington Method

The Huntington method, as described in Table 5-2.3-1, uses an iterative procedure to calculate polytropic work and efficiency based on measured performance parameters at the compressor section inlet and discharge, the gas properties, and one calculated intermediate point along a polytropic path.

5-2.4 Sandberg–Colby Multistep Method

The Sandberg–Colby multistep method, as described in Figure 5-2.4-1, uses an iterative procedure to calculate polytropic work and efficiency based on measured performance parameters at the compressor section inlet and discharge, the gas properties, and a large number of equal-pressure-ratio pressure steps along a polytropic path.

5-2.5 Tabulated Properties and Equations of State Methods

Thermodynamic property evaluations and polytropic work and efficiency calculations shall be performed by computer-based software. Use of a demonstrated accurate software-based equation of state for thermodynamic property evaluations is required. However, in some cases, it may be acceptable to create thermodynamic property look-up tables based on an appropriate equation of state and expected gas composition to improve computational efficiency.

Test uncertainty shall include an evaluation of uncertainty associated with the interpolation of property values from look-up tables in addition to the uncertainty of direct property evaluations by the used equation of state.

NOTE: The use of look-up tables may be considered if the same look-up table is used for the predicted performance and during the design stage. The evaluation of uncertainties involved in using a look-up table, which has not been used for the predicted performance, is out of the scope of this Code.

5-3 TYPE 2 TEST GAS AND TEST SPEED SELECTION

5-3.1 Test Gas Selection

The gas to be used in establishing the performance of the compressor to be tested can be the specified gas or a gas that allows for similarity testing at equivalent conditions. See Table 3-2.1-2.

5-3.2 Test Speed Selection

The volume ratio limitation of Table 3-2.1-2 may be met by controlling the test speed. Also see para. 3-4.2 for additional considerations in selecting test speed. The appropriate test speed is estimated from

$$\frac{N_t}{N_{sp}} = \sqrt{\frac{w_{p,t}}{w_{p,sp}}}$$

where

$$w_{p,t} = \left[(h_d - h_i) - \left(\frac{T_i + T_d}{2}\right)(s_d - s_i)\right]_t$$

and

$$w_{p,sp} = \left[(h_d - h_i) - \left(\frac{T_i + T_d}{2}\right)(s_d - s_i)\right]_{sp}$$

with the restriction that

$$r_{v,t} = r_{v,sp}$$

NOTE: The calculation of the test speed can be affected by the actual Reynolds number correction factor (specified conditions versus test conditions). However, this has a small effect on speed selection and as a simplification is not used.

Table 5-2.3-1
Huntington Method Polytropic Relations

Step	Step Description at Test or Specified Conditions
1	Calculate intermediate point pressure: $$p_3 = \sqrt{p_d p_i}$$
2	Calculate first estimate of intermediate temperature: $$T_{3,j} = \sqrt{T_d T_i}$$ for $j = 1$
3	Calculate intermediate values of compressibility factor, $Z_{3,j}(p_3, T_3,j)$; specific entropy, $s_{3,j}(p_3, T_3,j)$; and constant pressure specific heat, $Cp_{3,j}(p_3, T_3,j)$.
4	Calculate the coefficients A, B, and C: $$A = Z_i - B$$ $$B = \frac{\left(Z_i + Z_d - 2Z_{3,j}\right)}{\left[\left(p_d/p_i\right)^{0.5} - 1\right]^2}$$ $$C = \frac{\left[Z_d - A - B \times \left(p_d/p_i\right)\right]}{\ln\left(p_d/p_i\right)}$$
5	Calculate a revised estimate of the intermediate point entropy: $$s_{3,j+1} = s_i + (s_d - s_i) \times \left\{ \frac{\frac{A}{2}\ln\left(p_d/p_i\right) + B\left[\left(p_d/p_i\right)^{0.5} - 1\right] + \frac{C}{8}\left[\ln\left(p_d/p_i\right)\right]^2}{A\ln\left(p_d/p_i\right) + B\left[\left(p_d/p_i\right) - 1\right] + \frac{C}{2}\left[\ln\left(p_d/p_i\right)\right]^2} \right\}$$
6	Calculate a revised estimate of the intermediate point temperature: $$T_{3,j+1} = T_{3,j}\exp\left[\frac{(s_{3,j+1} - s_{3,j})}{Cp_{3,j}}\right]$$
7	Iterate on the value of the intermediate temperature, T_3, by returning to Step 3 with the revised temperature until $$\frac{T_{3,j+1} - T_{3,j}}{T_{3,j+1}} \leq 10^{-6}$$ [Note (1)]
8	Once the value of the intermediate temperature, T_3, is found, calculate the polytropic efficiency and work as follows: *(a)* polytropic efficiency $$\eta_p = \left\{1 + \frac{(s_d - s_i)/R}{A\ln\left(p_d/p_i\right) + B\left[\left(p_d/p_i\right) - 1\right] + \frac{C}{2}\left[\ln\left(p_d/p_i\right)\right]^2}\right\}^{-1}$$ *(b)* polytropic work $$w_p = \eta_p(h_d - h_i)$$

NOTE: (1) Convergence normally occurs within three to five iterations of the intermediate temperature.

Figure 5-2.4-1
Sandberg–Colby Multistep Numerical Integration Method

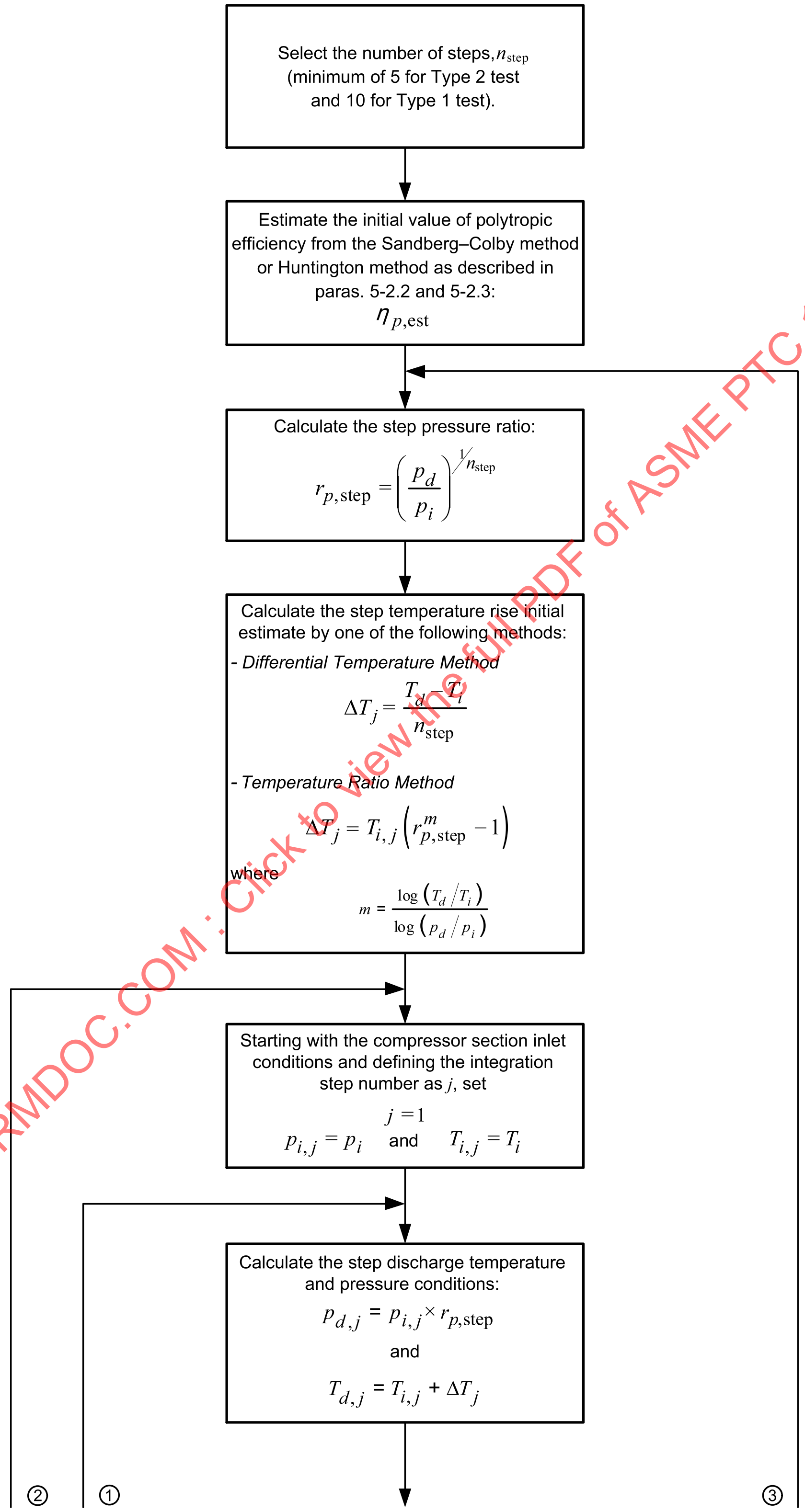


Figure 5-2.4-1
Sandberg–Colby Multistep Numerical Integration Method (Cont'd)

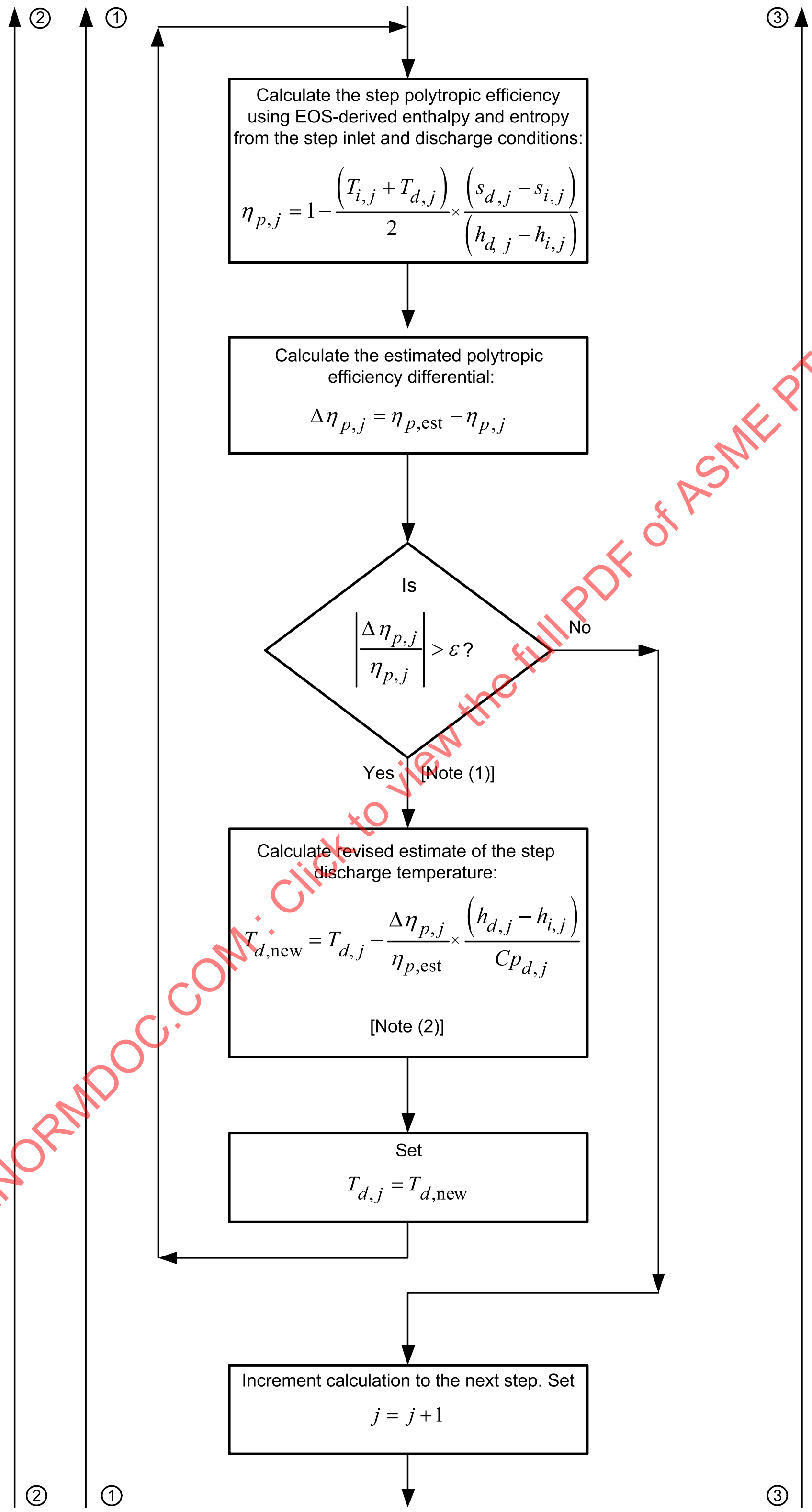


Figure 5-2.4-1
Sandberg–Colby Multistep Numerical Integration Method (Cont'd)

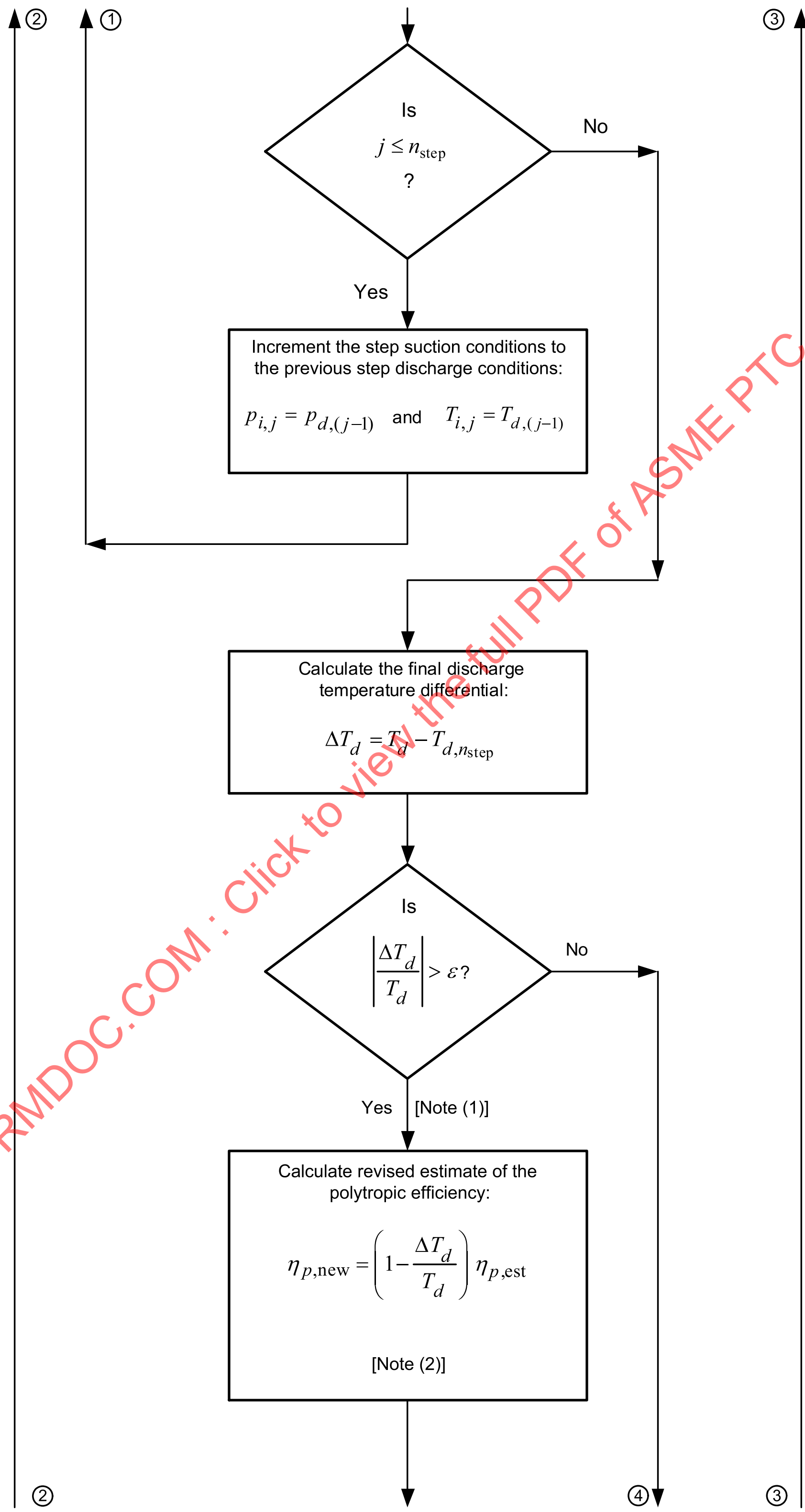
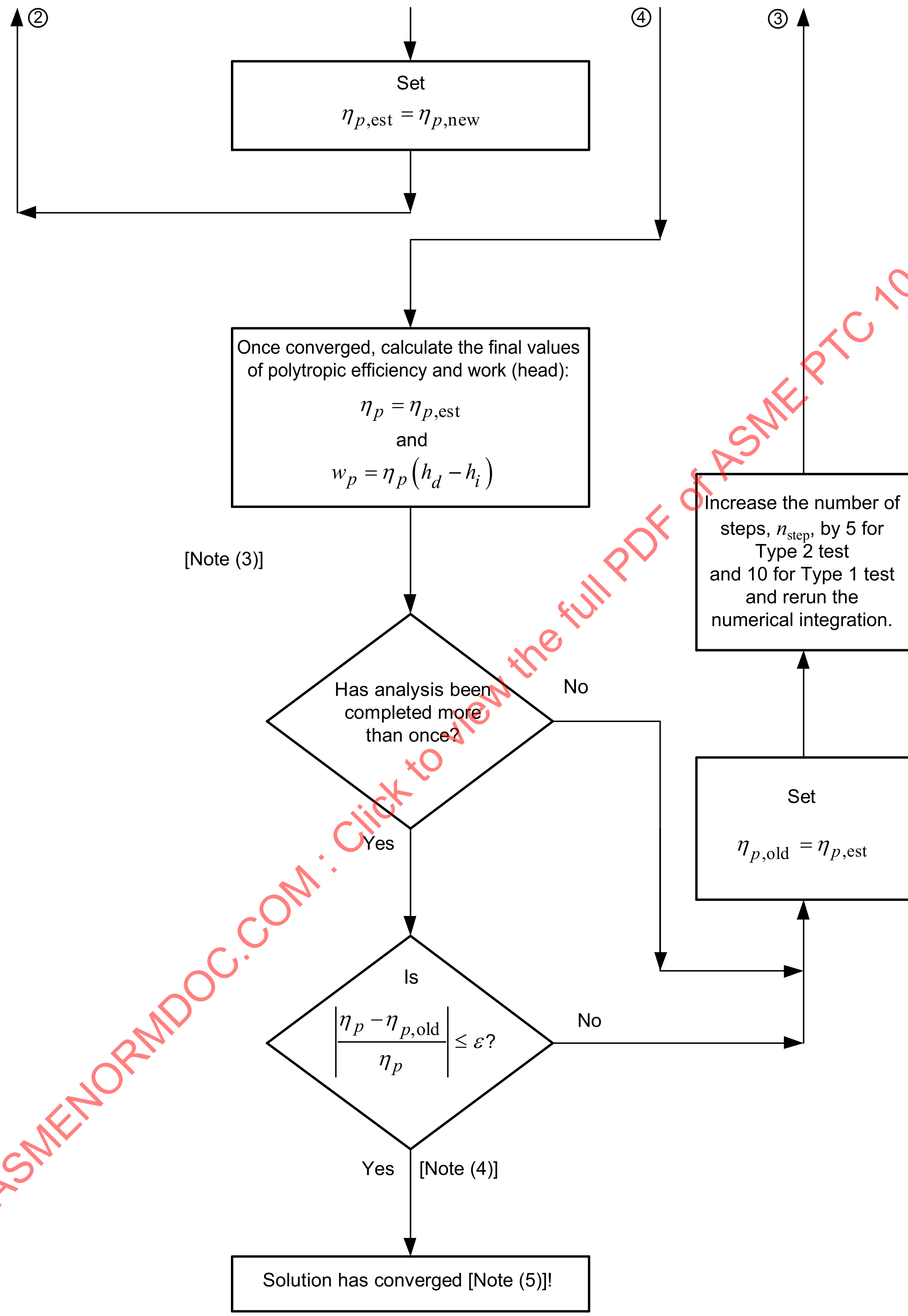


Figure 5-2.4-1
Sandberg–Colby Multistep Numerical Integration Method (Cont'd)



NOTES:
(1) Convergence tolerance, ε, should be held to 1×10^{-5} or less.
(2) Alternative root-finding algorithms may be successfully used in lieu of this scheme.
(3) Alternatively, a single multistep analysis can be provided with the identified number of steps from Table 7-4-1 with a high probability (>95%) of listed maximum error. Uncertainty for Type 2 tests will likely be less than these values.

Figure 5-2.4-1
Sandberg–Colby Multistep Numerical Integration Method (Cont'd)

NOTES (Cont'd)
(4) Analyses of several example cases including uncertainties associated with the pressure and temperature tolerances allowed in Table 4-18.2-1 applied to the overall uncertainty evaluation in accordance with the methods provided in Section 7 and illustrated in Nonmandatory Appendix H demonstrate that a convergence tolerance of 1×10^{-4} is adequate to eliminate added uncertainty due to the numerical integration method.
(5) Convergence normally occurs between 10 and 25 total steps, depending on the complexity of the compression path and thermodynamic properties.

5-3.3 Test Speed Validation

When the actual test conditions differ from the estimated values, the most appropriate test speed will depart from the previously estimated test speed. The test speed is acceptable when the permissible deviation from specified operating parameters is within the limits of Table 3-2.1-2.

NOTE: Predicted compressor performance characteristics are generally used to estimate the test conditions and the test speed. To the extent that the actual performance differs from the predicted performance, the test speed and conditions may need to be adjusted to achieve the tolerances of Table 3-2.1-2. After the test conditions are confirmed, the predicted performance characteristics do not affect the test results.

5-4 CALCULATIONS FOR TEST CONDITIONS

5-4.1 Test Data Acquisition

Operating data taken at test conditions is processed by the following sequence:

(a) Gather information.
  (1) Determine the following items:
    (-a) number of probes
    (-b) number of readings (minimum of three)
    (-c) recording timing and sequence (see subsection 3-10)
  (2) Record ambient conditions for pressure and temperature.

(b) Record observations for a test point as follows:
  (1) Record observation for each probe (gauge data) for first reading.
  (2) Repeat (1) for all subsequent readings in sequence according to timing and stability requirements.
  (3) Convert observations to absolute values.

(c) Remove outliers as per ASME 19.1, Nonmandatory Appendix A, as follows:
  (1) Average all observations for each reading.
  (2) Calculate S_x (standard deviation) for each reading.
  (3) Obtain τ from ASME PTC 19.1, Nonmandatory Appendix A.
  (4) Calculate the product τS_x.
  (5) Calculate δ (absolute difference of value) of each observation from the average.
  (6) If δ is greater than or equal to τS_x, then the observation is an outlier.
  (7) Mark and remove outliers.

(d) Sum the remaining data values for a reading and determine the arithmetic average. Repeat for each reading.

(e) Remove excessive fluctuations as follows:
  (1) Calculate the fluctuation for each reading without outliers (see para. 5-4.3.3).
  (2) Determine the acceptable fluctuation (see Table 3-12.2-1).
  (3) If the actual fluctuation is less than the allowed fluctuation, then the reading is acceptable.
  (4) Remove unacceptable readings.

(f) Determine test point value as follows:
  (1) Average all remaining acceptable readings without outliers.
  (2) Subtract the ambient pressure (or absolute temperature) to get the final data point gauge pressure (or temperature).
  (3) Convert to total conditions (see para. 5-4.3.5).

NOTE: Refer to ASME PTC 19.1 for definition of S_x, τ, and δ.

5-4.2 Raw Data Acceptability

The observed data shall be validated for compliance with the limitations imposed in Sections 3 and 4. Outliers shall be excluded from observed data by the modified Thompson method found in ASME PTC 19.1, Nonmandatory Appendix A. See Nonmandatory Appendix C for observed data treatment example calculations. Observed data so validated will be acceptable raw data.

5-4.3 Processing Raw Data

Acceptable raw data shall be processed to provide values to be used in the computation of results.

5-4.3.1 Calibrations and Corrections. Applicable instrument and system calibrations shall be applied to the acceptable raw data, which becomes the corrected raw data. The need for corrections and calibrations arises from both the indicating system components and measurement technique.

5-4.3.2 Data Conversion. The corrected raw data is then averaged from the total number of observations (raw data) at each measurement station. This averaged data becomes the reading. Refer to Nonmandatory Appendix C.

5-4.3.3 Fluctuation. Three or more readings are used to obtain the test point. The allowable fluctuation of the readings is shown in Table 3-12.2-1. The fluctuation is computed by taking the differences of the highest reading and the lowest reading and dividing by the average of all the readings.

$$\%\ \text{Fluctuation} = \frac{\text{Highest Reading} - \text{Lowest Reading}}{\frac{1}{n_r}\sum_{i=1}^{n_r}(i\text{th Reading})} \times 100\%$$

where

n_r = total number of readings

5-4.3.4 Test Point Data. After removing raw data outliers and any readings with excessive fluctuations, the remaining readings are summed and averaged. This average becomes the test point data and shall be converted to total conditions for use in performance calculations.

5-4.3.5 Total Conditions. Unless otherwise stated, total condition values shall be used for the computational procedure. Static test point data shall be converted to total condition values. This does not preclude final presentation in terms of static conditions.

The relationship between static properties and total properties is velocity dependent. Average total properties are estimated herein from the average velocity at the measurement station.

The average velocity at the measurement station is given by

$$V = \frac{\dot{m}}{\rho_{\text{static}} A}$$

5-4.4 Test Pressure and Temperature

Two methods are provided in paras. 5-4.4.1 and 5-4.4.2 to determine the total pressure and total temperature at the test conditions to include the effects of velocity at the measurement location. The party conducting the test may select and document the choice of one of the two specified methods. Nonmandatory Appendix C provides example problems.

NOTE: The choice of calculation method is not expected to impact the overall uncertainty of the test.

5-4.4.1 Rigorous Method. Figure 5-4.4.1-1 details the steps required for the rigorous method to calculate the test total pressure and test total temperature. This method relies on the use of an equation of state to determine accurate fluid properties to compute pressures and temperatures for known state conditions. This rigorous method should result in the most accurate determination of the total pressure and total temperature.

5-4.4.2 Alternate Mach Number Method. Figure 5-4.4.2-1 details an alternative process to determine the test total pressure and test total temperature based on the fluid Mach number and other fluid properties at the measurement location.

Figure 5-4.4.1-1
Rigorous Method to Calculate the Test Total Pressure and Temperature

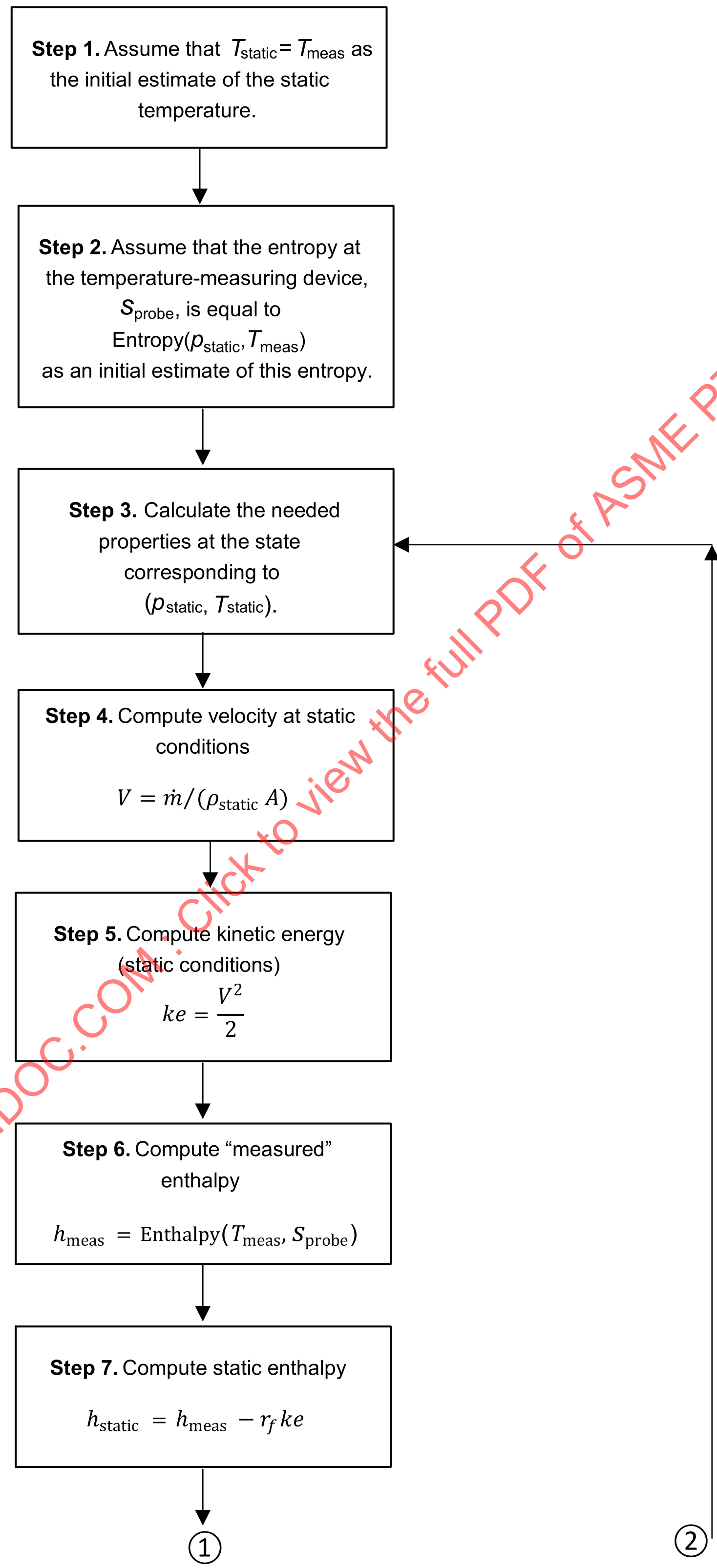
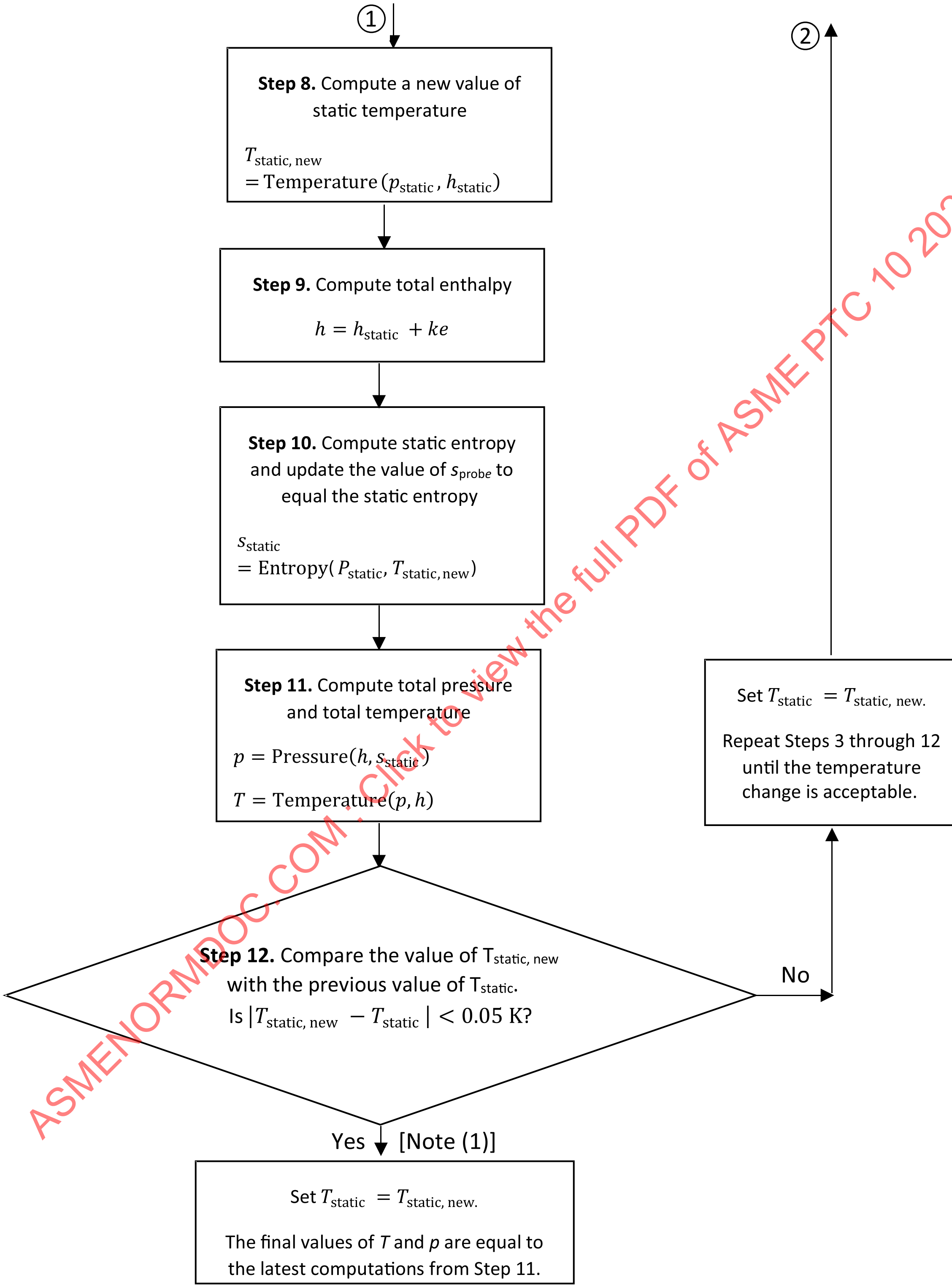


Figure 5-4.4.1-1
Rigorous Method to Calculate the Test Total Pressure and Temperature (Cont'd)



NOTE: (1) Unless the value of r_f is near zero, repeat Steps 3 through 12 at least once regardless of the value of $|T_{\text{static, new}} - T_{\text{static}}|$.

Figure 5-4.4.2-1
Alternative Mach Number Method to Calculate the Test Total Pressure and Temperature

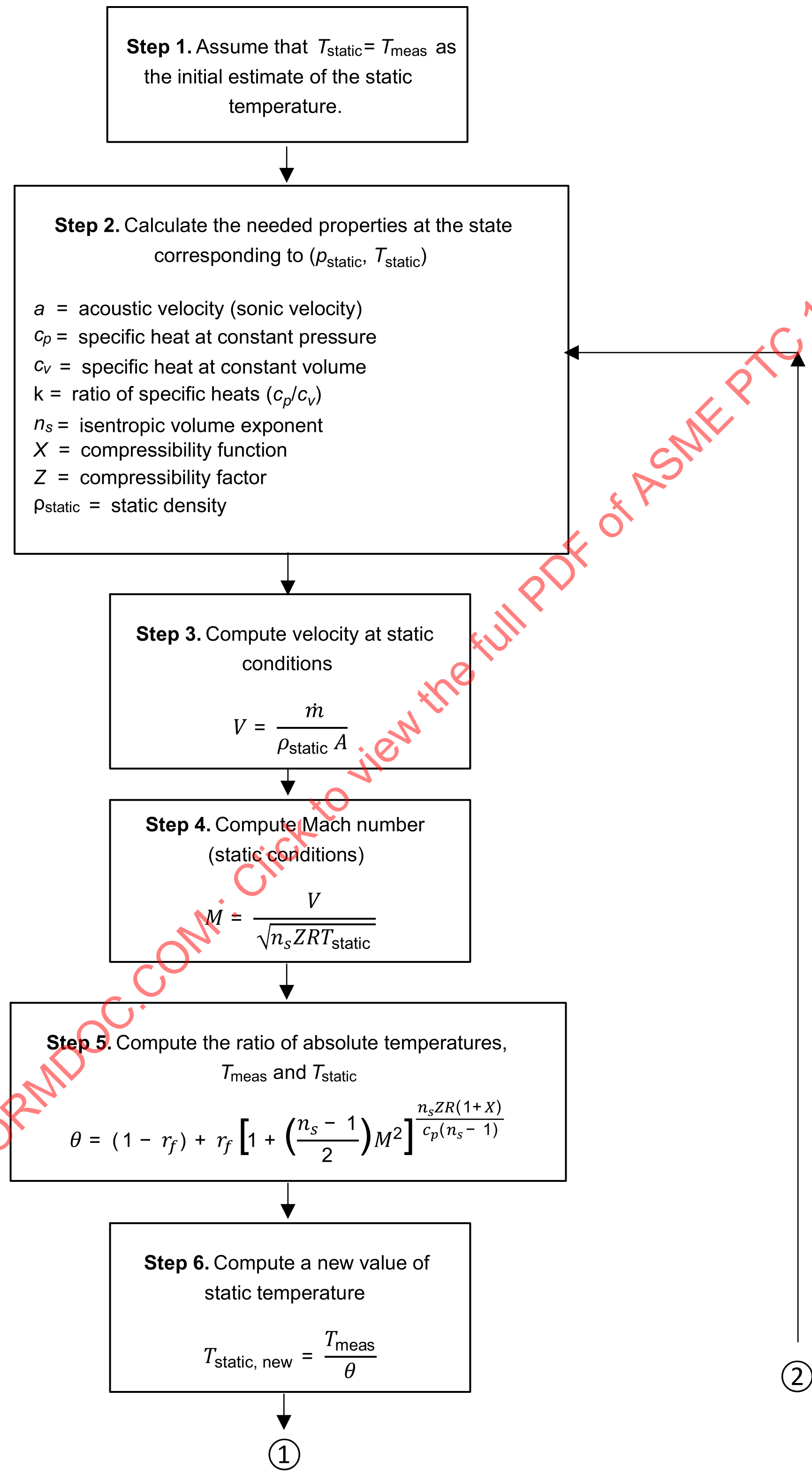
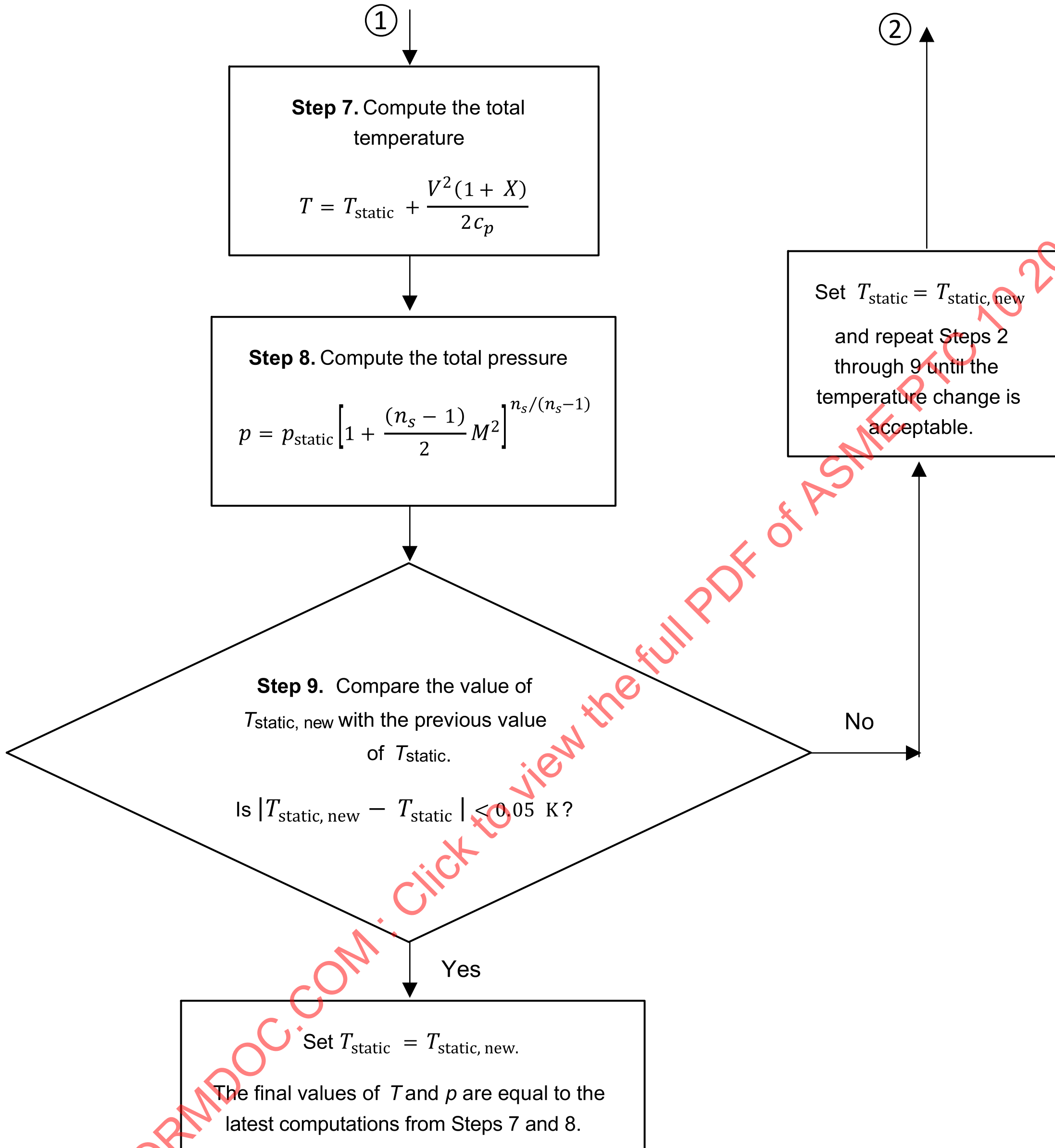


Figure 5-4.4.2-1
Alternative Mach Number Method to Calculate the Test Total Pressure and Temperature (Cont'd)



5-4.4.3 Recovery Factor. The recovery factor, r_f, is defined in terms of enthalpy as

$$r_f = \frac{(h_{\text{meas}} - h_{\text{static}})}{(h - h_{\text{static}})}$$

NOTE: Nonmandatory Appendix G includes a more detailed description of the recovery factor. Refer to ASME PTC 19.3.

5-4.4.4 Test Total Pressure and Total Temperature. The test total pressure and test total temperature are determined from the measured pressure, temperature, and recovery factor using either the rigorous or alternative Mach number method of Nonmandatory Appendix G.

5-4.5 Test Density and Specific Volume

The test total density is calculated from the test total pressure and test total temperature as

$$\rho_t = \left[\frac{p}{ZRT}\right]_t$$

The test total specific volume is the reciprocal of the test total density

$$v_t = \frac{1}{\rho_t}$$

5-4.6 Test Flow Rate

The measured flow rate is calculated according to the formulas applicable to the indicating instrument used. In some cases, flows such as leakages may be wholly calculated rather than measured when mutually acceptable methods are available (refer to Nonmandatory Appendix B applicable to a back-to-back compressor).

5-4.6.1 Mass Flow Rate. Test flow rates may be expressed as mass rate of flow at the station of interest.

5-4.6.2 Volume Flow Rate. This Code uses a flow rate definition in the calculation process that has the units of volume flow rate. It is

$$q = \frac{\dot{m}}{\rho}$$

where

$\dot{m}$ = mass flow rate

ρ = total density

This definition is consistent with the use of total properties in the calculation procedure. It does not represent the actual local volume flow rate because it is based on total density rather than static density. All references to calculated volume flow rate imply this definition unless otherwise stated.

5-4.7 Test Power

The calculation of test power depends on the method of measurement. Both shaft power and gas power may be of interest.

5-4.7.1 Shaft Power Method. When power input is measured by an instrument such as a torque meter or by electrical power measurement, the shaft power is calculated using the appropriate formula. Gas power is calculated by subtracting the parasitic losses from the shaft power (see para. 5-4.7.4 for parasitic losses).

$$P_{g,t} = P_{sh,t} - P_{\text{par},t}$$

5-4.7.2 Heat Balance Method. Gas power is calculated from the first law of thermodynamics applied to the compressor section of interest, yielding

$$P_{g,t} = \left(\sum\nolimits_{\text{out}} \dot{m}h - \sum\nolimits_{\text{in}} \dot{m}h\right) + Q_{sb}$$

where

$\left(\sum_{out} \dot{m}h - \sum_{in} \dot{m}h\right)$ = the balance of the products of mass flow rate and enthalpy for each flow stream departing and entering the control volume associated with a section

Q_{sb} = the total amount of heat departing the control volume associated with the section

Shaft power is the sum of gas power plus any parasitic losses

$$P_{sh,t} = P_{g,t} + P_{\mathrm{par},t}$$

NOTE: See Nonmandatory Appendix E, para. E-3.11.

5-4.7.3 Casing Section Heat Transfer. The external heat loss or heat gain from the section may be computed from measurements of the exposed surface area, the average temperature of the surface, and the ambient temperature.

Section Convection Heat Transfer:

$$Q_{\mathrm{conv}} = h_{\mathrm{conv}}\, S_c\, [T_c - T_a]$$

Section Radiation Heat Transfer:

$$Q_{\mathrm{rad}} = \sigma\, S_c e_{\mathrm{rad}}\left[T_c^4 - T_a^4\right]$$

Section Boundary Heat Transfer:

$$Q_{sb} = Q_{\mathrm{conv}} + Q_{\mathrm{rad}}$$

where

e_{rad} = casing surface emissivity
h_{conv} = convection heat transfer coefficient
S_c = heat transfer surface area of exposed compressor
T_a = ambient temperature near compressor casing
T_c = casing surface temperature
σ = Stefan–Boltzmann constant

Where the casing surface temperature varies widely, the accuracy of this calculation may be improved by calculating small areas of the surface separately and summing the results. See para. 4-16.1.

NOTE: The convective heat transfer coefficient is specific to each individual installation. It should be agreed on by all parties involved prior to testing and should be included in the test procedure and report (see Nonmandatory Appendix C, para. C-3.12).

5-4.7.4 Parasitic Losses. Parasitic losses are the difference between shaft power and gas power for the section or sections of interest. Parasitic losses comprise mechanical losses and other power requirements that do not contribute to the energy rise of the gas in the section of interest

$$P_{\mathrm{par}} = P_{\mathrm{mech}} + P_{\mathrm{other}}$$

(a) Mechanical Losses. Mechanical losses are always considered to be parasitic losses. Those losses caused by lubricated gears, bearings, and/or oil seals can be estimated from the lubricating oil temperature rise. Other mechanical losses from devices such as dry gas seals, nonpressurized bearings, or one or more couplings that do not contribute to the lubricating oil temperature rise may be determined separately.

$$P_{\mathrm{mech}} = P_{\mathrm{mech}_{\text{lube oil}}} + P_{\mathrm{mech}_{\text{non-lube oil}}}$$

The portion of the mechanical loss evident in the lubricating oil temperature rise is given by

$$P_{\mathrm{mech}_{\text{lube oil}}} = \dot{m}c\Delta t$$

where

c = average specific heat of the lubricating or sealing fluid
$\dot{m}$ = mass flow rate of the lubricating or sealing fluid
Δt = temperature rise of the lubricating or sealing fluid

(b) Other Parasitic Losses (P_{other}). When the shaft power method is used, power supplied to drive auxiliary equipment is treated as parasitic. Power supplied to sections of a multisection compressor other than the section being tested is also considered parasitic. When the heat balance method is used and total shaft power is defined to include power to drive auxiliary equipment, the auxiliary power requirement is treated as parasitic.

5-5 NONDIMENSIONAL PARAMETERS

The following nondimensional parameters are calculated for the test conditions to provide verification that the limits of Table 3-2.1-2 have been met.

5-5.1 Machine Mach Number

The machine Mach number of a section is given by

$$Mm = U/a_i$$

where the acoustic velocity of the gas at total inlet conditions is

$$a_i = \sqrt{\mathrm{n}_{s,i}\mathrm{Z}_i\mathrm{RT}_i}$$

5-5.2 Machine Reynolds Number

The machine Reynolds number of a section is given by

$$Rem = Ub/\nu_i$$

(a) For centrifugal compressors, the variables are defined as follows:

b = fluid flow passage tip width of the first impeller
U = velocity at the largest blade tip diameter of the first impeller of a section
ν_i = kinematic viscosity of the gas at total inlet conditions of this section

(b) For axial compressors, the variables are defined as follows:

b = chord at tip of the first-stage rotor blade
U = velocity at the tip diameter of the leading edge of the first-stage rotor blade of a section
ν_i = kinematic viscosity of the gas at total inlet conditions of this section

5-5.3 Specific Volume Ratio

The specific volume ratio of a section is the ratio

$$r_v = v_i/v_d$$

5-5.4 Volume Flow Ratio

The volume flow ratio between any two points x and y within the compressor flow path is given by

$$r_q = \frac{q_x}{q_y} = \frac{\left(\frac{\dot{m}_x}{\rho_x}\right)}{\left(\frac{\dot{m}_y}{\rho_y}\right)}$$

For compressors without sidestreams, the inlet volume to discharge volume flow ratio is limited by the specific volume ratio limit. For sidestream compressors, the volume flow ratio limits of Figure 3-5.2-1 also apply. (See subsection 3-5.)

5-5.5 Flow Coefficient

The flow coefficient of a section is given by

$$\phi = \frac{4\ \dot{m}_R}{\rho_i \pi U D^2}$$

where

$\dot{m}_R$ = mass flow rate that enters the rotor and is compressed
N = the rotational speed, 1/s
U = blade tip speed, calculated as follows:
= πND

(a) For centrifugal compressors, D is the maximum diameter of the blade trailing edge of the first impeller in a section.
(b) For axial compressors, D is the maximum diameter of the first-stage rotor blade in a section.

NOTE: The mass flow rate $\dot{m}_R$ differs from the measured mass flow rate by the amount of leakage and sidestream flow that may occur between the rotor entry and the flow measurement station. Refer to Nonmandatory Appendix E, para. E-3.11 for the calculation of $\dot{m}_R$ and Figure E-3.11-1 for a schematic representation of the mainstream and leakage flows of a section.

5-6 CALCULATIONS FOR SPECIFIED OPERATING CONDITIONS

Performance at specified conditions is calculated by the following procedures. Certain additional nondimensional parameters are calculated for the test conditions and extended to specified conditions.

5-6.1 Single-Section Compressor

5-6.1.1 Description. The single-section compressor from inlet to outlet measurement stations experiences no gas cooling other than radiation and convection. No gas flow is added or removed other than that lost through seal or balance piston leakage. No phase change occurs.

5-6.1.2 Calculation Procedure for Single-Section Compressors

Step 1. Calculate the following values:
(a) flow coefficient
(b) polytropic work coefficient
(c) polytropic efficiency
(d) work input coefficient
(e) total work input coefficient

The equations needed to do this are shown in Tables 5-2.2-1, 5-2.3-1, and 5-6.1.2-1 and Figure 5-2.4-1, and are explained in detail in Nonmandatory Appendix E. The assumption column of Table 5-6.1.2-2 shows the relationship between the test condition values and specified condition values.

Step 2. Begin by conducting a performance test. Compressor performance at any specified condition operating point is determined from at least two bracketing test points. To perform the interpolation, the specified operating condition nondimensional parameters are treated as functions of the specified operating condition flow coefficient. The specified operating condition nondimensional parameters for each point may be plotted as shown in Figure 5-6.1.2-1. A smooth curve is drawn connecting the data points. For two points, this is simply linear interpolation. Improved data interpolation may be possible with additional test points and nonlinear curve fitting.

Step 3. Establish the compressor performance in nondimensional terms at the specified operating condition flow of interest. These parameters are subject to correction for the difference in machine Reynolds number between test operating conditions and specified operating conditions, as explained in para. 5-6.3. To establish the compressor performance in nondimensional terms, calculate a specified operating condition flow coefficient from the flow rate, speed, and inlet conditions of interest. The remaining nondimensional performance parameters are defined from the interpolation process in Step 2. This information is simply read from the curves of Figure 5-6.1.2-1 at the flow coefficient of interest. The compressor performance at the specified operating condition point of interest is now defined in nondimensional terms.

Step 4. Calculate the compressor performance in the desired dimensional form. This is done by solving the nondimensional parameter equations for those quantities of interest. Typical equations used to do this are shown in Table 5-6.1.2-2.

Table 5-6.1.2-1
Real Gas Nondimensional Parameters

Parameter	Relation at Test Operating Conditions	Equation No.
Flow coefficient	$\phi_t = \left(\frac{4\,\dot{m}_R}{\rho_i \pi U D^2}\right)_t$	(5-1)
Work input coefficient	$\mu_{\text{in},t} = \left(\frac{h_d - h_i}{\sum U^2}\right)_t$	(5-2)
Polytropic work coefficient	$\mu_{p,t} = \left(\frac{w_p}{\sum U^2}\right)_t$	(5-3)
Polytropic efficiency	$\eta_{p,t} = \left(\frac{w_p}{h_d - h_i}\right)_t$	(5-4)
Total work input coefficient (heat balance method)	$\Omega_{hb,t} = \left[\frac{\dot{m}_d}{\dot{m}_R}\frac{(h_d - h_i)}{\sum U^2} + \frac{\dot{m}_{sd}}{\dot{m}_R}\frac{(h_{sd} - h_i)}{\sum U^2} + \frac{\dot{m}_{ld}}{\dot{m}_R}\frac{(h_{ld} - h_i)}{\sum U^2} + \frac{\dot{m}_{lu}}{\dot{m}_R}\frac{(h_{lu} - h_i)}{\sum U^2} - \frac{\dot{m}_{su}}{\dot{m}_R}\frac{(h_{su} - h_i)}{\sum U^2} + \frac{Q_{sb}}{\dot{m}_R \sum U^2}\right]_t$	(5-5) [Note (1)]
Total work input coefficient (shaft power method)	$\Omega_{sh,t} = \left(\frac{P_{sh} - P_{\text{par}}}{\dot{m}_R \sum U^2}\right)_t$	(5-6)

GENERAL NOTE: Appropriate units and conversions must be chosen to render the parameters nondimensional. Further explanation of these equations is available in Nonmandatory Appendix E.

NOTE: (1) This equation applies to a particular model as presented in Nonmandatory Appendix E, para. E-3.11. Some of the terms may not apply in a different case. The analysis presented in para. E-3.11 may be followed to develop the appropriate equation.

Table 5-6.1.2-2
Conversion of Nondimensional Parameters

Parameter	Mathematical Description at Converted Specified Conditions	Equation No.	Assumptions
Rotor mass flow rate	$\dot{m}_{R,\text{csp}} = \phi_{\text{csp}}\left(\frac{\rho_i \pi U_{sp} D^2}{4}\right)$	(5-7)	$\phi_{\text{csp}} = \phi_t \, Rem_{\text{corr},\,\phi}$
Delivered mass flow rate	$\dot{m}_{d,\text{csp}} = \dot{m}_{R,\,\text{csp}} - \dot{m}_{ld,sp} - \dot{m}_{sd,sp}$	(5-8) [Note (1)]	N/A
Capacity	$q_{\text{csp}} = \frac{\dot{m}_{d,\text{csp}}}{\rho_{i,sp}}$	(5-9)	N/A
Polytropic work per section	$w_{p,\text{csp}} = \mu_{p,\text{csp}} \sum U_{sp}^2$	(5-10)	$\mu_{p,\text{csp}} = \mu_{p,t} \, Rem_{\text{corr},\,\mu}$
Work input per section	$w_{\text{in},\text{csp}} = \frac{w_{p,\text{csp}}}{\eta_{p,\text{csp}}} = \left(\frac{\mu_p}{\eta_p}\right)_{\text{csp}} \sum U_{sp}^2$	(5-11)	$\mu_{p,\text{csp}} = \mu_{p,\,t} \, Rem_{\text{corr},\,\mu}$ $\eta_{p,\,\text{csp}} = \eta_{p,t} \, Rem_{\text{corr},\,\eta}$
Discharge enthalpy	$h_{d,\text{csp}} = w_{\text{in},\text{csp}} + h_{i,sp}$	(5-12)	N/A
Discharge entropy (Sandberg–Colby method basis)	$s_{d,\text{csp}} = \frac{2\left(1 - \eta_{p,\text{csp}}\right)\left(h_{d,\text{csp}} - h_{i,sp}\right)}{\left(T_{d,\text{csp}} + T_{i,sp}\right)} + s_{i,sp}$	(5-13) [Notes (2), (3)]	N/A
Discharge pressure	$p_{d,\text{csp}} = p\left(h_{d,\,\text{csp}}, s_{d,\,\text{csp}}\right)$	(5-14) [Notes (2), (3)]	N/A

Table 5-6.1.2-2
Conversion of Nondimensional Parameters (Cont'd)

Parameter	Mathematical Description at Converted Specified Conditions	Equation No.	Assumptions
Discharge temperature	$T_{d,\text{csp}} = T\left(h_{d,\text{ csp}}, s_{d,\text{ csp}}\right)$	(5-15) [Notes (2), (3)]	N/A
Total work input coefficient (heat balance method)	$\Omega_{hb,\text{csp}} = \frac{\dot{m}_{d,\text{csp}}}{\dot{m}_{R,\text{csp}}}\frac{\left(h_{d,\text{csp}} - h_{i,sp}\right)}{\sum U_{sp}{}^{2}} + \frac{\dot{m}_{sd,sp}}{\dot{m}_{R,\text{csp}}}\frac{\left(h_{sd,sp} - h_{i,sp}\right)}{\sum U_{sp}{}^{2}} + \frac{\dot{m}_{ld,sp}}{\dot{m}_{R,\text{csp}}}\frac{\left(h_{ld,sp} - h_{i,sp}\right)}{\sum U_{sp}{}^{2}} + \frac{\dot{m}_{lu,sp}}{\dot{m}_{R,\text{csp}}}\frac{\left(h_{lu,sp} - h_{i,sp}\right)}{\sum U_{sp}{}^{2}} - \frac{\dot{m}_{su,sp}}{\dot{m}_{R,\text{csp}}}\frac{\left(h_{su,sp} - h_{i,sp}\right)}{\sum U_{sp}{}^{2}} + \frac{Q_{sb,\text{csp}}}{\dot{m}_{R,\text{csp}}\sum U_{sp}{}^{2}}$	(5-16) [Notes (4), (5)]	[Note (6)]
Gas power per section	$P_{g,\text{csp}} = \dot{m}_{R,\text{csp}}\Omega_{hb,\text{csp}}\sum U_{sp}^{2}$	(5-17)	N/A
Shaft power (heat balance method)	$P_{sh,\text{csp}} = P_{g,\text{csp}} + P_{\text{par},sp}$	(5-18) [Note (7)]	N/A
Shaft power (shaft power method)	$P_{sh,\text{csp}} = \dot{m}_{R,\text{csp}}\Omega_{sh,\text{csp}}\sum U_{sp}^{2} + P_{\text{par},sp}$	(5-19) [Note (7)]	$\Omega_{sh,\text{csp}} = \Omega_{sh,t}\frac{Rem_{\text{corr},\mu}}{Rem_{\text{corr},\eta}}$

GENERAL NOTES:

(a) Appropriate units and conversions must be chosen to render the proper dimensional parameters. Further explanation of the application of these equations is available in Nonmandatory Appendix E.

(b) Pressure rise can be calculated using $\Delta p_{\text{csp}} = p_{d,\text{csp}} - p_{i,sp}$.

NOTES:

(1) If a measured and corrected specified value of leakage mass flow rate ($\dot{m}_{ld,sp}$) is available, it should be used instead of the original specified value. See para. 3-3.3.

(2) Equations (5-13) through (5-15) must be solved simultaneously via successive substitution or some comparable iterative method to obtain the correct pressure and temperature at the specified value of discharge enthalpy.

(3) Consistent use of the same or a more rigorous method to determine discharge conditions as that used to evaluate performance test parameters is necessary to minimize uncertainty of results.

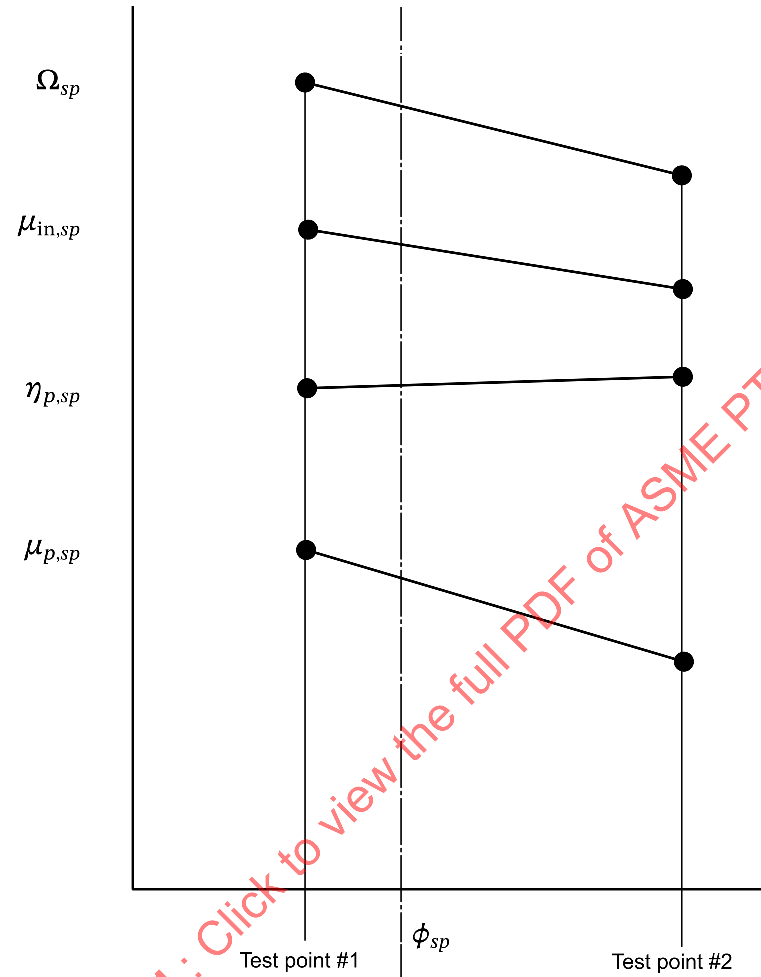
(4) If measured and corrected specified values of leakage and/or sidestream mass flow rates (including $\dot{m}_{sd,sp}$, $\dot{m}_{ld,sp}$, $\dot{m}_{lu,sp}$, and $\dot{m}_{su,sp}$) and associated corrected enthalpies (including $h_{sd,sp}$, $h_{ld,sp}$, $h_{lu,sp}$, and $h_{su,sp}$) of these streams are available, they should be used instead of the original specified values.

(5) Applicable revised casing section heat loss calculated from corrected discharge temperature should also be used.

(6) There is no Reynolds number correction between $\Omega_{hb,t}$ and $\Omega_{hb,\text{csp}}$ for the heat balance method.

(7) If any test-measured parasitic losses are available, they should be used preferentially to those that may be specified. Corrections to any measured mechanical losses for test speed versus specified speed should also be included (see para. 5-6.4).

Figure 5-6.1.2-1
Interpolation for the Specified Condition Flow Coefficient



5-6.2 The Multisection Compressor

5-6.2.1 Description. A multisection compressor is a compressor that may be treated as a number of individual single-section compressors operating in series. The output from each section provides input to the next section. The section boundaries may be drawn to exclude intermediate components such as external heat exchangers.

The following conditions shall be met to treat a compressor as a multisection compressor:

(a) It shall be possible to gather test information for each section as though it were an independent single-section compressor. That is, the test speed, flow rate, and inlet and outlet states shall be available for each section.

In the special case of sidestream mixing internally in a compressor, the inlet mixed condition shall be determined from the states of the incoming streams.

(b) When a component such as an external heat exchanger exists between sections, the performance of that component shall be known for test operating conditions and specified operating conditions.

(c) Differences in the intermediate component performance between test operating conditions and specified operating conditions shall have a minimal and quantifiable effect on the single-section performance, i.e., a minimal and quantifiable effect on the nondimensional performance parameters.

5-6.2.2 Calculation Method for Multisection Compressors. The specified operating condition performance for multisection compressors is calculated from the specified operating condition performance of the individual calculated sections. The basic calculation procedure for each section is the same as for single-section tests. The test data for each section is reduced to the form of nondimensional performance parameters that apply at the specified operating conditions. The performance of the first section is calculated the same as is done for a single-section compressor.

This yields the discharge conditions from the first section. If an intermediate component such as an intercooler exists before the next section entry, the effects on flow rate and gas state are taken into account.

For a heat exchanger, these effects are temperature reduction, pressure drop, and condensate removal. For the case of mixed streams, see Nonmandatory Appendix E, subsection E-5. The resulting condition becomes the specified operating condition gas state at the entry to the second section. The flow coefficient calculated from the known flow rate becomes the interpolating flow coefficient for the second section. The calculation process is repeated through the second section, the remaining intermediate components and sections, and on to the final discharge. It is not necessary for an intermediate component to exist to treat a compressor in multiple sections. The exit of one section and entry of another may coincide.

NOTE: The specified operating condition flow coefficients for the second and subsequent sections are functions of the performance of the preceding sections. This dependence on preceding section performance is an effect commonly referred to as section matching. When the individual section performance curves are steep, and as the number of individual sections increases, the overall compressor performance becomes increasingly sensitive. It is because of this effect that it is important to follow the calculation method presented. What may appear to be small differences between test operating conditions and specified operating conditions in each section may combine to show up as important effects in overall performance. Calculation methods that attempt to make overall corrections without explicit consideration of the section matching effect can lead to erroneous results.

5-6.3 Machine Reynolds Number Correction

5-6.3.1 General. The performance of a compressor is affected by the machine Reynolds number. Frictional losses in the internal flow passages vary in a manner similar to friction losses in pipes or other flow channels. If the machine Reynolds number at test operating conditions differs from that at specified operating conditions, a correction to the test results is necessary to properly predict the performance of the compressor.

The flow patterns of axial and centrifugal compressors are relatively complex. The term "machine Reynolds number" is used to provide a basis for definition in this Code. The machine Reynolds number correction for centrifugal compressors recommended in this Section is based on Strub et al. (1987). The machine Reynolds number correction for axial compressors is based on Carter et al. (1960).

If another method of correction is used it shall be agreed on by the parties prior to the test.

5-6.3.2 Correction Factor. Frictional losses in the flow passage of a compressor section are influenced by both the Reynolds number and the surface roughness of the flow passage. Related but different correction factors for, respectively, efficiency, work coefficient, flow coefficient, work input coefficient, and total work input coefficient for the shaft power method are defined as follows:

$$Rem_{\text{corr},\eta} = \frac{\eta_{p,\text{csp}}}{\eta_{p,t}}$$

$$Rem_{\text{corr},\mu} = \frac{\mu_{p,\text{csp}}}{\mu_{p,t}}$$

$$Rem_{\text{corr},\varphi} = \frac{\varphi_{\text{csp}}}{\varphi_t}$$

$$Rem_{\text{corr},\mu_{in}} = \frac{\mu_{\text{in,csp}}}{\mu_{\text{in},t}} = \frac{Rem_{\text{corr},\mu}}{Rem_{\text{corr},\eta}}$$

$$Rem_{\text{corr},\Omega} = \frac{\Omega_{sh,\text{csp}}}{\Omega_{sh,t}} = \frac{Rem_{\text{corr},\mu}}{Rem_{\text{corr},\eta}}$$

The correction factors for the specified condition are related to the test condition as follows:

(a) *For Centrifugal Compressors*

$$Rem_{\text{corr},\eta} = \frac{1}{\eta_{p,t}} + \left(1 - \frac{1}{\eta_{p,t}}\right)\frac{\left(0.3 + 0.7\frac{\lambda_{sp}}{\lambda_{\infty}}\right)}{\left(0.3 + 0.7\frac{\lambda_{t}}{\lambda_{\infty}}\right)}$$

$$Rem_{\text{corr},\mu} = 0.5 + 0.5Rem_{\text{corr},\eta}$$

$$Rem_{\text{corr},\phi} = \sqrt{Rem_{\text{corr},\mu}}$$

The steps for determining the machine Reynolds number correction for centrifugal compressors are shown in Figure 5-6.3.2-1.

(b) *For Axial Compressors*

$$\frac{1 - \eta_{p,\text{csp}}}{1 - \eta_{p,t}} = \frac{Rem_{sp}}{Rem_{t}}$$

The limitations of Table 3-2.1-2 apply. No corrections are applied for work and flow coefficient for axial compressors.

5-6.3.3 Limits of Application. Since the performance variations increase substantially as the machine Reynolds number decreases, tests of centrifugal compressors designed for operation at low machine Reynolds numbers should be tested at conditions close to those specified. Therefore, the maximum and minimum permissible ratios between Rem_t and Rem_{sp} are shown in Figure 3-2.1-3. Also, see Nonmandatory Appendix F.

5-6.4 Mechanical Losses

When the mechanical losses at specified operating conditions are not known, they may be estimated from the following equation:

$$P_{m,sp} = P_{m,t}\left(\frac{N_{sp}}{N_t}\right)^{j}$$

The exponent j in the preceding equation may vary with the design of bearings, thrust loads, lube oil systems, couplings, etc. The exponent j usually has a value between 2.0 and 3.0. Specific values of the exponent j or alternative calculation methods should be obtained from the associated equipment manufacturer.

Figure 5-6.3.2-1
Machine Reynolds Number Correction for Centrifugal Compressors

Step 1. Obtain the necessary information to determine the correction factors

b = impeller tip width, m

Ra = average roughness, m

U_{sp} = impeller tip speed at specified conditions, m/s

U_t = impeller tip speed at test conditions, m/s

ν_{sp} = kinematic viscosity of specified gas, m^2/s

ν_t = kinematic viscosity of test gas, m^2/s

Step 2. Calculate the machine Reynolds numbers at the specified conditions and proposed test conditions

$$Rem_{sp} = \frac{U_{sp} b}{\nu_{sp}} \qquad Rem_t = \frac{U_t b}{\nu_t}$$

Step 3. Limit the lower value of the machine Reynolds numbers at the specified conditions

If $Rem_{sp} < 90{,}000$ then $Rem_{sp} = 90{,}000$

Step 4. Determine the lower limit for Rem_t

If $Rem_{sp} > 500{,}000$ then

lower limit for $Rem_t = 0.1 \times Rem_{sp}$

ELSE lower limit for $Rem_t = 10^{ll}$

where

$$ll = -22.733 - 4.247° \times \log_{10}(Rem_{sp}) + 21.63 \times \sqrt{\log_{10}(Rem_{sp})}$$

Step 5. Determine the upper limit for Rem_t

IF $Rem_{sp} > 800{,}000$ then

upper limit for $Rem_t = 100 \times Rem_{sp}$

ELSE upper limit for $Rem_t = 10^{ul}$

where

$$ul = 68.205 + 16.13 \times \log_{10}(Rem_{sp}) - 64.008 \times \sqrt{\log_{10}(Rem_{sp})}$$

Step 6. Apply the following limits to Rem_t:

If $Rem_t >$ upper limit for Rem_t
or $Rem_t <$ lower limit for Rem_t
then test conditions are unacceptable. Go back to Step 1 and reselect the test conditions.
ELSE Continue to Step 7.

Step 7. Apply the following limits to relative roughness:

If $Ra < \frac{b}{10^6}$ then $Ra = \frac{b}{10^6}$ (hydraulically smooth)

If $Ra < \frac{b}{20}$ then $Ra = \frac{b}{20}$ (very rough)

Step 8. Calculate critical friction factor at infinite Rem using the von Karman equation

$$\lambda_\infty = [1.74 - 2\log_{10}(\frac{2Ra}{b})]^{-2}$$

Step 9. Solve the combination of the von Karman and Colebrook equations for λ_t and λ_{sp} [Note (1)]

$$\lambda_t = \left[\frac{1}{\sqrt{\lambda_\infty}} - 2\log_{10}\left(1 + \frac{18.7}{Rem_t\sqrt{\lambda_t}}\frac{b}{2Ra}\right)\right]^{-2}$$

$$\lambda_{sp} = \left[\frac{1}{\sqrt{\lambda_\infty}} - 2\log_{10}\left(1 + \frac{18.7}{Rem_{sp}\sqrt{\lambda_{sp}}}\frac{b}{2Ra}\right)\right]^{-2}$$

Step 10. Calculate the correction factors [Note (2)]

$$Rem_{\mathrm{corr},\eta} = \frac{1}{\eta_{p,t}} + \left(1 - \frac{1}{\eta_{p,t}}\right)\frac{\left(0.3 + 0.7\frac{\lambda_{sp}}{\lambda_\infty}\right)}{\left(0.3 + 0.7\frac{\lambda_t}{\lambda_\infty}\right)}$$

$$Rem_{\mathrm{corr},\mu} = 0.5 + 0.5Rem_{\mathrm{corr},\eta}$$

$$Rem_{\mathrm{corr},\phi} = \sqrt{Rem_{\mathrm{corr},\mu}}$$

Step 11. The work input coefficient, μ_{in}, is equal to μ/η and therefore the correction factor for μ_{in} is

$$Rem_{\mathrm{corr},\,\mu_{\mathrm{in}}} = \frac{\mu_{\mathrm{in,csp}}}{\mu_{\mathrm{in},t}} = \frac{Rem_{\mathrm{corr},\mu}}{Rem_{\mathrm{corr},\eta}}$$

Figure 5-6.3.2-1
Machine Reynolds Number Correction for Centrifugal Compressors (Cont'd)

NOTES:
(1) The equations of Step 9 must be solved by successive substitution or another rooting technique. If using successive substitution, use initial guesses of $\lambda_t = \lambda_{sp} = \lambda_\infty$. If using a rooting technique, find the zeroes of the following rearrangements of equations of Step 9:

$$\frac{1}{\sqrt{\lambda_t}} + 2\log_{10}\left(1 + \frac{18.7}{Rem_t}\frac{b}{2Ra}\frac{1}{\sqrt{\lambda_t}}\right) - \frac{1}{\sqrt{\lambda_\infty}} = 0$$

$$\frac{1}{\sqrt{\lambda_{sp}}} + 2\log_{10}\left(1 + \frac{18.7}{Rem_{sp}}\frac{b}{2Ra}\frac{1}{\sqrt{\lambda_{sp}}}\right) - \frac{1}{\sqrt{\lambda_\infty}} = 0$$

(2) In Step 10, if the shaft power method is used, the correction factor for the total work input coefficient is

$$Rem_{\mathrm{corr},\Omega} = \frac{\Omega_{sh,\mathrm{csp}}}{\Omega_{sh,t}} = \frac{Rem_{\mathrm{corr},\mu}}{Rem_{\mathrm{corr},\eta}}$$

Section 6
Report of Test

6-1 CONTENTS

The report of test shall include applicable portions of the information shown in subsection 6-2 and may include other data as necessary.

Copies of the original test data log, certificates of instrument calibration, driver (motor or other type) efficiency data as needed, description of test arrangement and instrumentation, and any special written agreements pertaining to the test or the computation of results shall be included.

When tests are run over a range of operating conditions, the results shall also be presented in the form of performance test data and corresponding curves. The curves and data shall be clearly marked to denote use of static conditions or total conditions.

6-2 TYPICAL REPORT INFORMATION

6-2.1 General Information

A typical report contains the following information:

(a) date or dates and time of test
(b) location of test
(c) manufacturer
(d) manufacturer's serial numbers and complete identification
(e) party or parties conducting the test
(f) representatives of interested parties
(g) detailed written statement of the test
(h) agreement made by parties to the test
(i) reference to the agreed test procedure
(j) test objectives

6-2.2 Description of Test Calculations

A description of test calculations within a typical report contains the following information:

(a) type of test
  (1) Type 1 or Type 2
  (2) number of test points (refer to paras. 3-10.4 and 3-10.5)
(b) computational method (refer to subsection 5-2)
(c) method of determining fluid properties (see subsection 5-2)
(d) loss assumptions
  (1) losses addressed, e.g., heat transfer from casing (see para. 5-4.7.3), parasitic losses (see para. 5-4.7.4)
  (2) losses not addressed (e.g., coupling windage losses)
(e) leakage rate corrections and assumption basis (see para. 3-3.3)

6-2.3 Description of Test Installation

A description of the test installation within a typical report contains the following information:

(a) installation details (e.g., type of compressor, configuration)
  (1) number of stages
  (2) diameter of each impeller
  (3) tip width of the first impeller
  (4) arrangement of casing or casings and piping

(5) inlet pipe and discharge pipe sizes
(6) arrangement of intercoolers, if used
(b) description of lubricating system and lubricant properties
(c) type of shaft seals
(d) type and arrangements of driver
(e) description of cooling system and coolant properties, if used
(f) description of bearings (e.g., hydrodynamic bearings, magnetic bearings)

NOTE: Any losses associated with magnetic bearings are out of the scope of ASME PTC 10.

6-2.4 Specified Conditions and Test Conditions

A typical report contains the following specified conditions and test conditions:
(a) gas composition
(b) basis for evaluation of gas properties (see subsection 5-2)
(c) inlet gas state
(1) total pressure and static pressure
(2) total temperature and static temperature
(3) total density and static density
(4) relative humidity, if applicable
(d) gas flow rate
(1) inlet or discharge mass flow rate
(2) inlet or discharge volume flow rate
(e) discharge static pressure and discharge total pressure
(f) speed

6-2.5 Performance at Specified Conditions

A typical report contains the following performance at specified conditions:
(a) polytropic work
(b) polytropic efficiency
(c) gas power
(d) calculated shaft power
(e) discharge total temperature
(f) pressure ratio
(g) volume flow ratio
(h) flow coefficient
(i) machine Reynolds number
(j) machine Mach number

6-2.6 Setup of Instruments and Methods of Measurement

A typical report contains the following information about the setup of instruments and methods of measurement:
(a) one or more piping and instrumentation diagrams of the test arrangement, including locations of all measuring stations
(b) description of agreed departures from this Code
(c) instruments used for the measurement of flow rate (flowmeter dimensions and details), pressure, temperature, speed, composition of gas, density, torque, and power
(d) procedures and facilities used for the calibration of instruments
(e) calibration data and certificates
(f) instrument accuracy
(g) method of determining power losses, if any, between the power measurement device and the compressor input shaft
(h) description of sampling and analysis methods for test gas

6-2.7 Test Points Data (See Subsection 2-7)

A typical report contains the following test points data (after all calibrations and instrument corrections have been applied):

(a) test point number
(b) timeline of data collection (see para. 3-10.3)
(c) speed
(d) inlet static (or total) temperature
(e) barometer reading
(f) ambient temperature at barometer
(g) measured inlet static (or total) pressure
(h) ambient dry-bulb temperature, if required
(i) ambient wet-bulb temperature, if required
(j) test gas inlet dew point temperature (see para. 3-3.7)
(k) test gas discharge dew point temperature, if applicable
(l) gas composition (see subsection 4-9)
(m) gas density, if measured (see subsection 4-9)
(n) discharge static (or total) pressure
(o) discharge static (or total) temperature
(p) flowmeter data, typically
  (1) pressure differential across the flowmeter
  (2) static pressure on the upstream side of the flowmeter
  (3) static temperature on the upstream side of the flowmeter
  (4) flowmeter throat diameter
(q) gas power and shaft power
(r) torque, if measured
(s) lubricant flow rate, if applicable
(t) lubricant inlet temperature, if applicable
(u) lubricant outlet temperature, if applicable
(v) mean compressor casing (or insulation, if applicable) surface temperature
(w) ambient temperature near casing
(x) casing surface area
(y) leakage flow rates

6-2.8 Computed Results for Each Test Point (See Paras. 3-10.4 and 3-10.5)

A typical report contains the following computed results for each test point:
(a) test point number
(b) barometric pressure
(c) gas composition (see subsection 4-9)
(d) mass flow rate
(e) inlet static conditions
  (1) pressure
  (2) temperature

NOTE: An iterative solution may be required for temperature.

  (3) others as needed
(f) inlet volume flow rate
(g) average inlet velocity at the measurement location
(h) inlet total conditions
  (1) pressure
  (2) temperature

NOTE: An iterative solution may be required for temperature.

  (3) compressibility factor
  (4) density
  (5) enthalpy
  (6) entropy
  (7) others as needed
(i) capacity
(j) discharge static conditions
  (1) pressure

(2) temperature

NOTE: An iterative solution may be required for temperature.

(3) others as needed

(k) discharge volume flow rate

(l) average discharge velocity at the measurement location

(m) discharge total conditions

(1) pressure

(2) temperature

(3) compressibility factor

(4) density

(5) enthalpy

(6) entropy

(7) others as needed

(n) leakages

(1) mass flow rate

(2) enthalpy

(3) entropy

(o) sidestreams, if applicable

(1) total pressure

(2) total temperature

(3) compressibility factor

(4) density

(5) enthalpy

(6) entropy

(7) others as needed

(p) average mixed gas state properties for sidestream applications, if required

(q) rotor mass flow rate (see Table 5-6.1.2-2)

(r) mechanical losses

(s) heat transfer losses

(t) gas power

(u) shaft power

(v) polytropic work

(w) polytropic efficiency

6-2.9 Computed Test Performance Parameters

A typical report contains the following test performance parameters:

(a) polytropic work coefficient

(1) polytropic work

(2) blade tip velocity of each impeller

(3) polytropic work coefficient

(b) polytropic efficiency

(c) work input coefficient

(d) total work input coefficient

(1) energy lost or gained via leakage

(2) energy lost or gained via sidestream flows

(3) energy lost or gained via casing heat transfer

(e) parasitic losses (see para. 5-4.7.4)

(1) mechanical loss

(2) other loss

(f) flow coefficient

(g) specific volume ratio

(h) volume flow ratio (if applicable, see para. 3-5.2)

(i) machine Mach number

(j) pressure ratio

6-2.10 Machine Reynolds Number Correction

A typical report contains the following information about the machine Reynolds number correction:
(a) test operating condition machine Reynolds number
(b) specified operating condition machine Reynolds number
(c) machine Reynolds number correction for flow coefficient
(d) machine Reynolds number correction for head
(e) machine Reynolds number correction for polytropic efficiency
(f) the following specified operating conditions:
  (1) flow coefficient
  (2) work input coefficient
  (3) polytropic work coefficient
  (4) polytropic efficiency
  (5) total work input coefficient

6-2.11 Computed Results for Specified Operating Conditions

A typical report contains the following information about the computed results for specified operating conditions (speed and inlet gas state are given):
(a) flow rate
  (1) capacity
  (2) inlet and/or discharge mass flow rate
  (3) inlet and/or discharge volume flow rate
  (4) leakage flow rate
  (5) condensate from process gas, if present
  (6) sidestream flow rates
(b) discharge conditions
  (1) static pressure and total pressure
  (2) static temperature and total temperature
  (3) compressibility factor
  (4) static density and total density
(c) work-related terms
  (1) polytropic work
  (2) polytropic efficiency
  (3) gas power
  (4) shaft power

6-2.12 Uncertainty Analysis

An uncertainty analysis, if required, shall be included in the report of the test.

6-2.13 Summary of Results

The summary of results includes a comparison of test results with expected values.

Section 7
Test Uncertainty

7-1 GENERAL

Refer to ASME PTC 19.1 for test uncertainty and treatment of errors.

The methodology of ASME PTC 19.1 is the standard for ASME PTC 10 tests. See Nonmandatory Appendix H for sample uncertainty calculations.

7-2 SCOPE OF UNCERTAINTY ANALYSIS

The scope of the uncertainty analysis required for a given test is intimately related to the test objectives. The scope of uncertainty analysis is subject to agreement by the parties to the test. Such agreements shall be made prior to undertaking the test.

7-3 METHODS OF ASME PTC 19.1

The uniqueness of ASME PTC 10 test objectives precludes exhaustive treatment of uncertainty in this Code. It is anticipated that the user will refer to ASME PTC 19.1 for detailed information to apply to individual tests. The uncertainty analysis can thereby be tailored to meet the individual test objectives.

ASME PTC 19.1 presents a step-by-step calculation procedure to be conducted before and after each test.

7-4 TEST METHOD UNCERTAINTY

In addition to measurement uncertainty, the calculation of polytropic work and efficiency are not exact and contribute additional uncertainty for any uncertainty analysis using ASME PTC 19.1. The maximum expected uncertainty for each method described in subsection 5-2 is listed in Table 7-4-1. These values may be used with any uncertainty analysis using ASME PTC 19.1.

Table 7-4-1
Maximum Expected Uncertainty

Calculation Method	Maximum Expected Uncertainty, %
Sandberg–Colby method	0.38
Huntington method	0.02
Sandberg–Colby multistep method (10 steps)	0.004
Sandberg–Colby multistep method (20 steps)	0.001

GENERAL NOTE: More than 70 different example cases referenced in the open literature were evaluated (Evans and Huble, 2017a), including a majority that were highly nonideal gas conditions. The listed uncertainties are based on a 95% confidence level (two standard deviations) for the evaluated cases. These uncertainty differences will be reduced for typical Type 2 test conditions. Refer to Nonmandatory Appendix C for further details.

NONMANDATORY APPENDIX A
USE OF TOTAL PRESSURE AND TOTAL TEMPERATURE TO DEFINE COMPRESSOR PERFORMANCE

A-1 GENERAL

A compressor's performance characteristics that depend on thermodynamic properties for their definition are, under the provisions of this Code, based on total (stagnation) conditions. This procedure can cause confusion if the principles involved are not kept clearly in mind. Compressor performance may be specified at static pressures and static temperatures or at total pressures and total temperatures, as desired, and the following explanation serves to point out the differences between the two.

A-2 ENERGY EQUATION

When the first law of thermodynamics, written as the general energy equation, is applied to a compressor section with the system boundaries defined as the interior wall of the casing and the transverse planes across the inlet and discharge flanges in the absence of leakage, the following expression results:

$$h_{\text{static},\,i} + \frac{V_i^2}{2} + y_i + w_{sh} = h_{\text{static},\,d} + \frac{V_d^2}{2} + y_d + \frac{Q_{sb}}{\dot{m}} \tag{A-2-1}$$

Subscripts i and d refer to the static inlet conditions and static discharge conditions, respectively. The inlet and discharge flanges may be considered to be at the same elevation so that y_i and y_d, the elevation heads, become equal. Solving eq. (A-2-1) for w_{sh} gives

$$w_{sh} = \left(h_{\text{static},\,d} + \frac{V_d^2}{2}\right) - \left(h_{\text{static},\,i} + \frac{V_i^2}{2}\right) + \frac{Q_{sb}}{\dot{m}} \tag{A-2-2}$$

This result involves static enthalpies determined by static pressures and static temperatures.

A-3 TOTAL CONDITIONS

In terms of total conditions, eq. (A-2-2) becomes

$$w_{sh} = h_d - h_i + \frac{Q_{sb}}{\dot{m}} \tag{A-3-1}$$

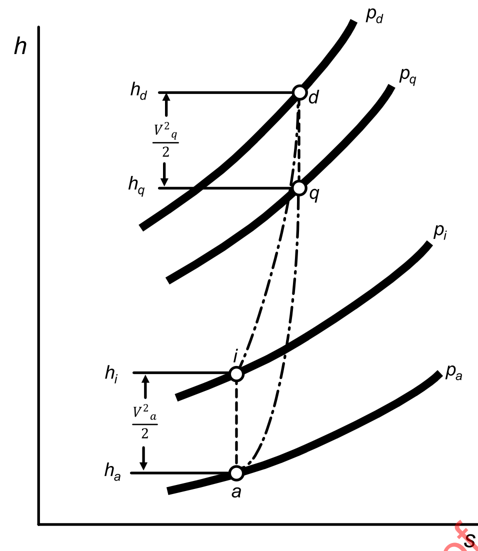
Subscripts i and d refer to total inlet conditions and total discharge conditions, respectively, as determined by total pressures and total temperatures. In the total process

$$h_i = h_{\text{static},\,i} + \frac{V_i^2}{2} \tag{A-3-2}$$

$$h_d = h_{\text{static},\,d} + \frac{V_d^2}{2} \tag{A-3-3}$$

The difference between static conditions and total conditions is shown graphically on an h–s diagram in Figure A-4-1.

Figure A-4-1
Compressor State Points, Static and Total



A-4 COMPRESSION PROCESS

As shown in Figure A-4-1, the process of compression takes place between states i and d. Some calculations regarding the internal compression process might require the use of static states intermediate to i and d. However, as shown by eqs. (A-2-1) through (A-3-3), use of the total properties for the external energy balance of the compressor is an excellent approximation for the following two reasons:

(a) Total enthalpy h_i (at inlet total pressure p_i) is equivalent to the static enthalpy $h_{\text{static},i}$ (at inlet static pressure $p_{\text{static},i}$) plus kinetic energy

$$\frac{V_i^2}{2}$$

(b) Total enthalpy h_d (at discharge total pressure p_d) is equivalent to the static enthalpy $h_{\text{static},d}$ (at inlet discharge pressure $p_{\text{static},d}$) plus kinetic energy

$$\frac{V_d^2}{2}$$

A-5 SYSTEM BOUNDARIES

The preceding analysis can be applied only because the system boundaries were carefully defined to preclude any consideration of events, thermodynamic or otherwise, taking place within the compressor proper. Studies of events internal to the compressor section are not included within the scope of this Code.

NONMANDATORY APPENDIX B TYPE 2 PERFORMANCE TESTING OF BACK-TO-BACK COMPRESSORS

B-1 OBJECTIVE

A sample setup and test loop arrangement used for Type 2 back-to-back compressor performance testing is described in this Appendix.

The objectives of the setup and piping arrangement are to

(a) eliminate division wall leakage and heat transfer effects between sections

(b) determine division wall and seal balance leakage rates that shall be applied in the conversion of test results to specified conditions

B-2 FUNCTIONAL OPERATION OF TYPE 2 PERFORMANCE TEST

B-2.1 Compressor Section One

During performance testing of compressor section one, loop balance valve A, as shown in Figure B-2.1-1, should be fully open and loop balance valve B should be fully closed. This results in a minimum pressure differential across the division wall, eliminating any division wall leakage that may affect the discharge temperature reading of compressor section one. The discharge temperature of section two should be maintained as close as possible to the discharge temperature of section one. This can be achieved by adjusting the inlet temperature of section two, thus minimizing any heat transfer effects between sections. The seal balance line valve C should be fully closed. This eliminates the seal leakage from the inlet of section two back to the inlet of section one. This requires each end seal to be referenced independently, which may require two seal systems to be used.

B-2.2 Compressor Section Two

During the performance test of compressor section two, loop balance valve A should be closed, and loop balance valve B should be open. This will result in the suction pressure of section two being as close as possible to the discharge pressure of section one. This orientation mimics how the machine will be operated in the field, where section two will be at a higher pressure than section one. Therefore, the discharge pressure of section two will be greater than the discharge pressure of section one, causing the leakage flow to cross the division wall seal. Pressure and temperature at these locations are being measured while the leakage is allowed to return to the second section loop through the loop balance orifice.

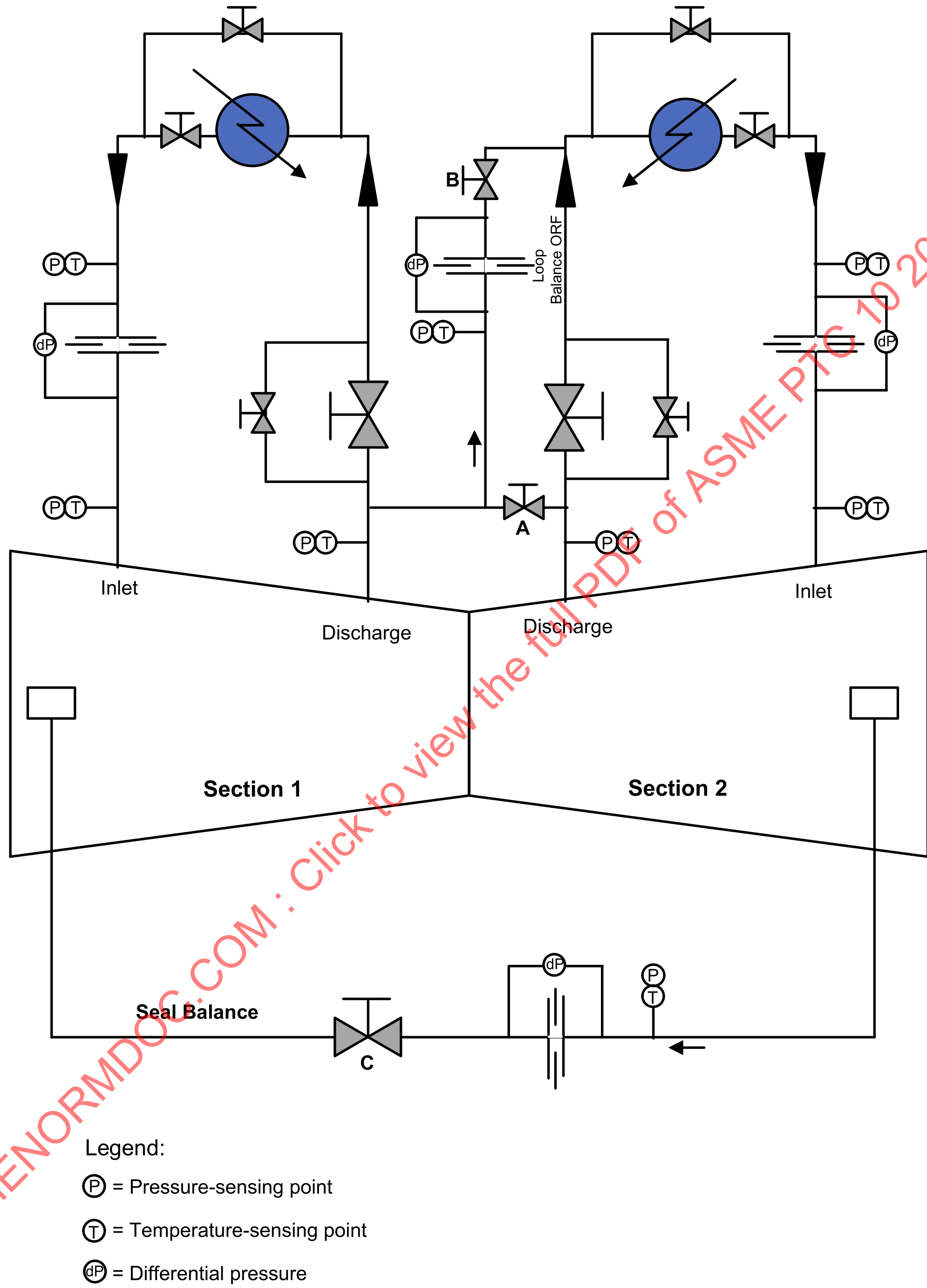
The difference between the discharge temperature of section one and the discharge temperature of section two should be maintained within 10 K by adjusting the inlet temperature of section one. This will minimize the heat transfer effect between sections on temperature readings.

Also, during the performance test of section two, seal balance valve C is fully open. This permits leakage to flow from the higher-pressure suction of section two to the lower-pressure suction of section one. This allows for the pressure balancing of the seal cavities and measurement of the leakage mass flow rate via the seal balance orifice. This leakage flow will then pass through the loop balance orifice as it returns to the loop of section two. Valve C is positioned downstream of the seal balance orifice.

B-2.3 Evaluation Considerations

The flow across the loop balance line orifice during the performance test of section two is the summation of flow that passed across the division wall and the seal balance line flow. Therefore, the division wall leakage mass flow rate is equal to the loop balance orifice mass flow rate minus the seal balance line orifice mass flow rate. Knowing these flow rates and the upstream and downstream conditions of the division wall seal and of the end seal, the effective seal leakage area for each seal may be calculated and used in the conversion of results to the specified conditions.

Figure B-2.1-1
Typical Back-to-Back Compressor Type 2 Test Setup



The upstream conditions of the division wall seal are the discharge conditions of section two. The downstream conditions of the division wall seal are the discharge conditions of section one for the corresponding data point.

The upstream conditions of the end seal are the inlet conditions of section two. The downstream conditions of the seal are the upstream conditions of the seal balance line orifice for the corresponding data point.

B-2.4 Calculation

Several different methods are available to calculate leakage mass flow across a restriction. One example method is illustrated by eq. (B-2-1):

$$\dot{m}_{sl} = C_{sl} A_{sl} p_h \sqrt{\frac{1 - \left(p_l / p_h\right)^2}{n_{sl} p_h v_h}} \tag{B-2-1}$$

where

A_{sl} = effective seal leakage area
C_{sl} = nondimensional seal type constant
$\dot{m}_{sl}$ = seal mass flow rate
n_{sl} = dimensionless constant representing seal type
p = pressure
v = specific volume calculated using pressure and temperature values

and the subscripts are defined as follows:

h = high pressure side (upstream of the seal)
l = low pressure side (downstream of the seal)
sl = related to seal leakage

The seal balance leakage mass flow is directly measured during the performance test via a flowmeter (orifice). The upstream density and pressure of the seal are the suction conditions of section two observed during the test. The downstream pressure is the upstream pressure at the orifice observed during the test. Knowing this data, the seal type constant, C_{sl}, and the seal type reference constant, n_{sl}, an effective seal leakage area, A_{sl}, may be calculated using eq. (B-2-1) for each test point. The average value of these as-tested effective seal leakage area values is then to be used in the evaluation at specified conditions.

For the division wall seal, the leakage flow is the difference between the measured loop balance flow and the measured seal balance flow. The upstream condition is the section two discharge condition, and the downstream pressure is the section one discharge pressure. The same evaluation procedure detailed earlier in this Appendix is used to establish an as-tested division wall effective seal leakage area. The average value of these as-tested effective seal leakage area values is then used in the evaluation at specified conditions.

The C_{sl} and n_{sl} values for a given seal design are usually defined by a party conducting the test. Tables B-2.4-1 and B-2.4-2 represent typical post-test evaluation forms.

A party conducting the test may choose a different method of leakage evaluation. The chosen method shall be cited in the test report.

Table B-2.4-1
Sample Post-Test Leakage Evaluation for Division Wall

Reading No.	Loop Balance Line Mass Flow Rate (Measured), $\dot{m}_{bl}$, kg/s	Seal Leakage Mass Flow Rate (Measured), $\dot{m}_{sb}$, kg/s	Division Wall Leakage Mass Flow Rate (Loop Balance Minus Seal Balance), $\dot{m}_{sl} = \dot{m}_{bl} - \dot{m}_{sb}$, kg/s	Upstream Pressure (Measured at Discharge of Section Two), p_h, kPa	Upstream Specific Volume (Measured at Discharge of Section Two), p_l, kPa	Upstream Specific Volume (Measured at Discharge of Section Two), v_h, m^3/kg	Effective Leakage Area, A_{sl}, m^2 [Note (1)]
1	0.184	0.094	0.090	871.2	660.7	0.1239	2.862 E−04
2	0.212	0.098	0.115	1 007.6	668.8	0.1095	2.785 E−04
3	0.222	0.094	0.127	1 098.8	667.3	0.1023	2.683 E−04
4	0.218	0.096	0.122	1 145.0	664.9	0.1000	2.430 E−04

NOTE: (1) Average effective leakage area to be used in calculations at specified conditions is 2.690 E−04.

Table B-2.4-2
Sample Post-Test Leakage Evaluation for Balance Seal

Data Point No.	Balance Seal Leakage Mass Flow Rate (Measured), $\dot{m}_{sb}$, kg/s	Upstream Pressure (Measured at Suction of Section Two), p_h, kPa	Downstream Pressure (Measured at Seal Orifice Upstream), p_l, kPa	Upstream Specific Volume (Measured at Suction of Section Two), v_h, m^3/kg	Effective Leakage Area, A_{sl}, m^2 [Note (1)]
1	0.094	621.5	408.2	0.1481	1.942 E−04
2	0.098	627.3	415.7	0.1469	2.001 E−04
3	0.094	627.0	415.1	0.1469	1.973 E−04
4	0.096	628.3	418.7	0.1468	1.976 E−04

NOTE: (1) Average effective leakage area to be used in calculations at specified conditions is 1.964 E−04.

NONMANDATORY APPENDIX C
SAMPLE CASE CALCULATIONS

C-1 INTRODUCTION

C-1.1 Purpose

This Appendix illustrates some of the required calculations and their sequence for a sample case for an ASME PTC 10 Type 2 performance test of a centrifugal compressor. It does not provide a complete test report as outlined in Section 6.

C-1.2 Type 2 Test

Subsection 1-4 describes two types of performance tests. Calculations for each type are similar, but additional restrictions are imposed on a Type 1 test. Extending the calculations shown in this Appendix to a Type 1 test is not included but is straightforward.

Type 2 tests of a centrifugal compressor must conform to the requirements shown in Tables 3-2.1-2 and 3-12.2-1 to adhere to the rules imposed by similitude. Additional engineering design and safety restraints that are not explicit ASME PTC 10 requirements, such as pressure/temperature limits of construction materials, test stand limits, and compressor critical speeds, should also be considered when creating a test procedure.

C-1.3 Thermodynamic Properties

Thermodynamic properties required for compressor performance evaluation shall be determined by use of an equation of state (EOS) that considers the potential for strongly varying property behavior of the compressed fluid, i.e., a real gas. A number of generalized and compositional equations of state and associated mixing rules are available for the many gases and mixtures of gases that may be encountered during the application of ASME PTC 10. Users of this Code should, to the extent practical, apply equations of state that have been validated for the gases, mixtures of gases, and range of conditions required for a particular application. Whenever possible, published validations of these equations of state may be referenced or, when not available, any proprietary validations should be summarized. Performance test result uncertainty due to EOS uncertainties should be described in the test report. Different implementations of a common EOS might include some variation in equation constants and thermodynamic property correlations, which may impact calculated property values. While equations of state are integral to performance calculations, they are beyond the purview of ASME Performance Test Codes.

In this Code, thermodynamic properties are based on absolute, total conditions, which are used as arguments in calls to an EOS software package unless otherwise noted (see Nonmandatory Appendix A).

It is important to maintain a consistent EOS in developing performance parameters for compressor design, compressor predicted performance, any Type 1 or Type 2 testing that may be completed, and evaluation and monitoring of the compressor during installed operation.

C-1.3.1 Historical Labels for Gas Behavior Models and Polytropic Work. Gas state thermodynamics began historically with the concept of a perfect gas model, then transitioned to an ideal gas model, and later transitioned to the real gas models available today. Table C-1.3.1-1 provides a high-level definition of these three gas behavior models. The most general is a real gas model as it encompasses perfect and ideal gas models and handles situations in which gas properties vary strongly with pressure and temperature. Fluids encountered in a compressor may not be adequately represented by the early concepts of perfect and ideal gas models. This Code requires the use of a real gas model.

Compressor analysis previously used the label “head” when referring to polytropic work developed by a compressor. In this context, the interchangeable use of the words “head” and “work” became ingrained within the industry. ASME PTC 10 now uses only the label “polytropic work.”

Table C-1.3.1-1
Historical Gas Models and Labels

Gas Model Label	Equation of State	Treatment of Compressibility	Treatment of Specific Heat
Perfect	$Pv = RT$	Constant = 1 implied	Constant
Ideal	$Pv = RT$	Constant = 1 implied	Function of temperature only
Real	$Pv = ZRT$	Function of pressure and temperature	Function of pressure and temperature

C-2 PERFORMANCE ANALYSIS ELEMENTS

The following actions must be completed to execute a Type 2 test, determine polytropic performance test results, and apply those results to obtain a compressor's site performance at specified conditions.

(a) Select and define an EOS to be used to determine fluid thermodynamic properties.
(b) Select and define the polytropic work computational method (see subsection 5-2).
(c) Gather required physical data regarding the compressor and its specified performance.
(d) Define a Type 2 test gas, inlet conditions, discharge conditions, test speed, mass flow rates, and test design performance according to similitude rules.
(e) Complete the test and record the necessary measured data.
(f) Process the measured data (see para. 5-4.1).
(g) Calculate compressor as-tested performance.
(h) Convert test results to nondimensional coefficients (see Table 5-6.1.2-1).
(i) Confirm the as-tested results conform to Type 2 test requirements.
(j) Apply Reynolds number corrections to as-tested nondimensional coefficients.
(k) Calculate converted test results in dimensional form by converting the corrected as-tested nondimensional coefficients to specified conditions (see Table 5-6.1.2-2).

NOTE: The uncertainty sample calculations are included in Nonmandatory Appendix H.

C-3 HIGH-PRESSURE NATURAL GAS COMPRESSOR SAMPLE CALCULATIONS

Four sets of performance calculations and/or parameters are presented in this subsection to show the sequential process from conceptual compressor design to Type 2 tested, as-built performance. To avoid confusion with various terms labeled "specified" in this subsection, and to help distinguish between the four sets of values, the following categories are established:

(a) Specified Performance. The first set of values represents an original equipment manufacturer's (OEM) predicted design performance based on customer-specified parameters and the OEM's hardware design. See para. 3-1.1, which defines specified conditions that do not change throughout the analysis process. This performance data set is typically provided to a customer by the OEM during the purchasing phase of a project. In this Code, the word *specified* and the subscript *sp* are used to identify input and output values belonging to this set.

(1) The Type 2 test is intended to provide results that can be used to demonstrate how well the as-built compressor compares to this specified performance data set.

(2) Although they are not an explicit part of an ASME PTC 10 test, the calculations to develop specified performance values are included in this Appendix (see paras. C-3.1 through C-3.3) for completeness and clarity. Some of these values are necessary for further calculations in subsequent sections in this Appendix.

(b) Test Design Performance. The second set of values has two parts, representing the screening of several potential Type 2 test gases in para. C-3.4 and the final refined Type 2 test design in para. C-3.5, using the selected test gas. Values from this set are the test target values established before the test and will be replaced by the actual test values from the set described in (c) after completion of a successful test.

(c) As-Tested Performance. The third set of values represents as-tested values, including operating conditions and performance results. Paragraphs C-3.6 through C-3.9 cover as-tested performance. In this Code, the words *as-tested* and the subscript *t* are used to identify input and output values belonging to this set.

(d) Converted Specified Performance. The fourth set of values deals with conversion of test results to final as-built performance at specified conditions and is covered in paras. C-3.10 and C-3.11. In this Code, the subscript csp is used to identify converted values, and the subscript *sp* is retained for the specified conditions from the set in (a) since those values have not changed.

Calculations for heat transfer to ambient surroundings are covered in para. C-3.12.

C-3.1 Analysis Method Decisions

C-3.1.1 Equation of State. The Type 2 performance analyses included in this Appendix use a real gas EOS package, REFPROP (version 10), from the National Institute of Standards and Technology. REFPROP implements multiple pure-substance, reference-quality, multiparameter equations of state for thermodynamic and transport properties of individual fluids. REFPROP then applies mixing rules to determine mixture properties (Lemmon et al., 2018). REFPROP is capable of addressing all thermodynamic and transport property requirements for the various gas compositions and conditions being used in this Appendix. This Appendix uses the International Steam Table definition of 1 Btu/lbm equals exactly 2 326 J/kg for the relationship between U.S. Customary and SI units regarding energy (see ASME PTC 2, Table 5.4).

Temperature values in most of the tables in this Appendix are shown in degrees Fahrenheit or degrees Celsius even though absolute values are used in the calculations. The chosen EOS package has options that handle this adjustment automatically.

C-3.1.2 Polytropic Work Calculation Methods. Three distinct polytropic work calculation methods are identified in this Code for the determination of compressor thermodynamic performance. These are discussed here in order of increasing complexity. Each of the three methods is applicable to both Type 2 and Type 1 testing and generally provides reduced uncertainty with increasing calculation complexity. The use of the more complex procedures should be considered when the test gas conditions fall in regions where gas properties vary strongly to maintain minimal calculation uncertainties. Although uncertainties are comparable for anticipated Type 2 conditions, the more significant gas property deviations encountered in some Type 1 tests benefit from the more complex calculation procedures.

The three different calculation methods may be summarized as follows:

(a) *Sandberg–Colby Endpoint Method.* This is the simplest of all the methods, requiring only values of specified thermodynamic properties at inlet conditions and discharge conditions. Although it is applicable to all types of testing, it also allows independent validation of the more complex methods within an acceptable error tolerance with a simple calculation that can be readily achieved with hand calculations using values of the identified properties obtainable from a number of thermodynamic property sources.

(b) *Huntington Method.* The Huntington method will normally provide an approximate order-of-magnitude reduction in uncertainty from the Sandberg–Colby endpoint method for Type 1 test conditions and requires some iteration to arrive at a solution. The Huntington method can be accommodated with a spreadsheet or numerical computer application with the addition of embedded thermodynamic properties.

(c) *Sandberg–Colby Multistep Method.* This is the most complex calculation method but asymptotically converges toward an exact solution through a numerical integration along a path of constant polytropic efficiency. The reduced uncertainty available from this method requires nested iteration loops and logic achievable within customized computer calculations. This method's superior accuracy allows it to serve as the baseline for alternate calculation method comparisons, and when gas properties vary in an extremely strong fashion this is the desired method.

Each of these methods relies on the use of total pressures and temperatures at inlet and discharge measurement locations. The use of static and/or measured values for pressures and temperatures instead of total values would lead to greater uncertainty and is not allowed by this Code.

Further information on these calculation methods can be found in Evans and Huble (2017b), Huntington (1985), Sandberg (2020), and Sandberg and Colby (2013).

Reported polytropic performance in this Appendix includes all three computational methods described in subsection 5-2 for comparison. The Sandberg–Colby single-step method is used to screen potential test gas compositions prior to determining a refined test performance using the Sandberg–Colby multistep method for the selected gas. Results from the Sandberg–Colby endpoint method are used for further processing to obtain site performance.

C-3.2 Analysis Inputs for Specified Performance of OEMs

The sample Type 2 test and performance analysis described in this subsection pertains to a high-pressure natural gas compressor. It is a single-stage, straight-through design with eight impellers and no sidestreams, similar to the sketch in Figure C-3.2-1.

Neither seal leakage nor balance piston flows are addressed in this sample case; the inlet, rotor, and discharge mass flows are assumed equal. Although this assumption eases some of the sample calculations, in many actual cases it can lead to excessive uncertainty. In reality, all parties should agree how leakage calculations will be addressed in the test plan. The heat balance method is used to determine power (see para. 5-4.7.2).

Table C-3.2-1 provides the specified gas composition, specified conditions, hardware description, and mechanical design limitations for the subject compressor. Both U.S. Customary and SI units are provided. The source for each input is also listed (OEM or user). These values are typically set by contractual agreements between the OEM and the user; they are beyond the scope of ASME PTC 10.

Figure C-3.2-2 illustrates a pressure–temperature phase diagram for the specified gas composition and the relation of suction conditions and discharge conditions to the two-phase fluid region. Figure C-3.2-2 shows that specified operating conditions are in a supercritical region and are well removed from the two-phase region, thus avoiding the possibility of liquid being ingested into the compressor during operation at or near specified inlet conditions. The analysis techniques included in ASME PTC 10 are not intended to be used for any fluid applications involving two-phase flow.

The term *specified conditions* is defined in para. 3-1.1 and includes the items highlighted in blue in Table C-3.2-1. These items do not change throughout the following performance analysis process. Inlet and discharge inside pipe diameters should correspond to pressure and temperature measurement locations and are referred to as nozzle locations.

Note that measured inlet pressure and discharge pressure are shown in gauge pressure units of pounds per square inch gauge (psig) [kilopascals gauge (kPa gauge)], and corresponding measured temperatures are shown in degrees Fahrenheit (degrees Celsius). See Nonmandatory Appendices A and G, Figures 5-4.4.1-1 and 5-4.4.2-1, and the following sections for conversion to absolute pressures and determination of static conditions and total conditions. In practice, users should note the types of instruments used for measurement and if the specified pressure and temperature values are presented as static, measured, or total, and should adjust the calculations accordingly.

Figure C-3.2-1
Typical Straight-Through Centrifugal Compressor Section

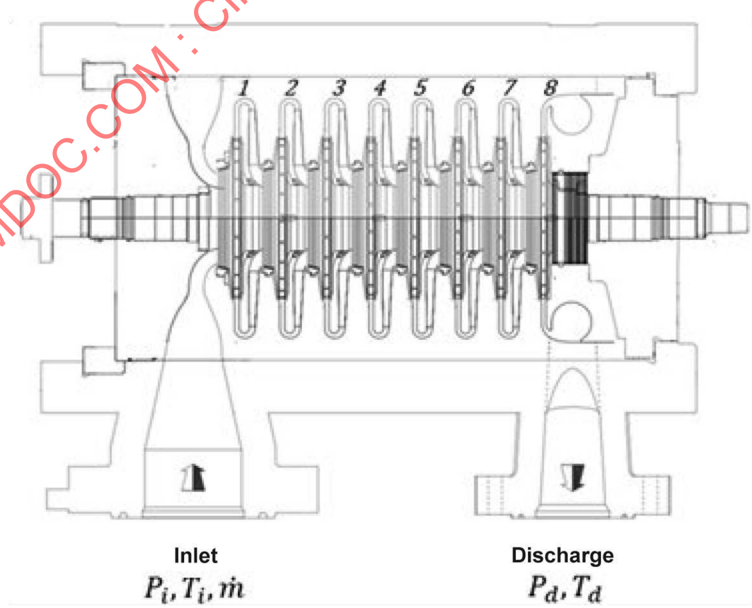
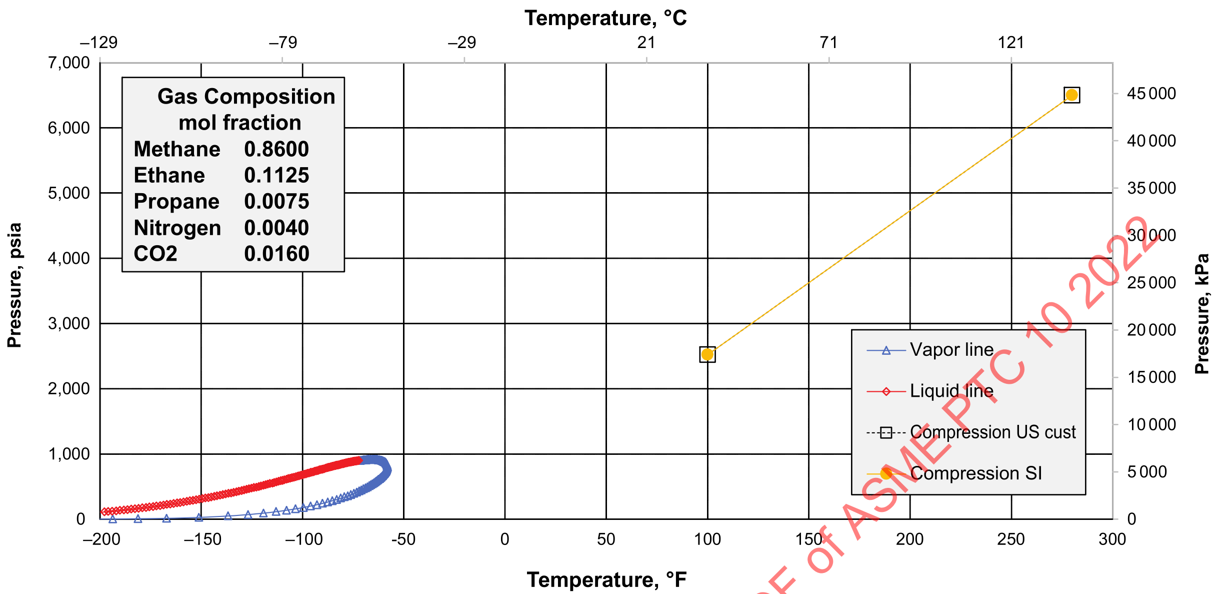


Table C-3.2-1
Specified Design for High-Pressure Natural Gas Compressor

	U.S. Customary Units		SI Units		Source
Standard Conditions					
Standard Pressure	psia	14.696	101.325	kPa	User
Standard Temperature	F	59.00	15.00	C	User
Operating Conditions					
Project Identifier		HP Natural Gas	HP Natural Gas		User
Condition Name		Specified	Specified		User
Barometric Pressure	psia	14.700	101.353	kPa	User
Ambient Temperature	F	65.00	18.33	C	User
Inlet Pressure measured	psig	2505.28	17273.3	kPag	User
Inlet Temperature measured	F	99.99	37.77	C	User
Inlet Temperature Probe Recovery Factor		0.65	0.65		OEM
Discharge Pressure measured	psig	6483.68	44703.4	kPag	User
Discharge Temperature measured	F	279.99	137.77	C	OEM
Discharge Temperature Probe Recovery Factor		0.65	0.65		OEM
Rotor Speed	rev / min	10680	178.0	1 / s	OEM
Inlet Mass Flow	lbm / min	3900.0	29.484	kg / s	User
Fluid Composition					
Methane	mole fraction	0.8600	0.8600	mole fraction	User
Ethane	mole fraction	0.1125	0.1125	mole fraction	User
Propane	mole fraction	0.0075	0.0075	mole fraction	User
Nitrogen	mole fraction	0.0040	0.0040	mole fraction	User
Carbon dioxide	mole fraction	0.0160	0.0160	mole fraction	User
Fluid Physical Properties		See Table C-3-3-1			OEM
Compressor Section Hardware Details					
Compressor Model / Section Label		HP Section	HP Section		OEM
Inlet Pipe Inside Diameter	inches	7.090	0.1801	m	OEM
Discharge Pipe Inside Diameter	inches	5.160	0.1311	m	OEM
Number of Impellers		8	8		OEM
1st Impeller Diameter	inches	13.636	0.3464	m	OEM
2nd Impeller Diameter	inches	13.636	0.3464	m	OEM
3rd Impeller Diameter	inches	13.636	0.3464	m	OEM
4th Impeller Diameter	inches	13.636	0.3464	m	OEM
5th Impeller Diameter	inches	13.636	0.3464	m	OEM
6th Impeller Diameter	inches	13.636	0.3464	m	OEM
7th Impeller Diameter	inches	13.636	0.3464	m	OEM
8th Impeller Diameter	inches	13.636	0.3464	m	OEM
1st Impeller Tip Width	inches	0.34	0.00864	m	OEM
Flow Path Surface Roughness	inches	0.000125	3.1750E-06	m	OEM
Compressor Design Limits					
Maximum Continuous Speed	rev / min	11500	191.67	1 / s	OEM
1st Critical Speed	rev / min	5400	90.00	1 / s	OEM
2nd Critical Speed	rev / min	13700	228.33	1 / s	OEM
Maximum Allowable Working Pressure	psia	8300	57226	kPa	OEM
Maximum Allowable Temperature	F	380	193	C	OEM
Minimum Allowable Temperature	F	-20	-29	C	OEM
Maximum Allowable Nozzle Flow Velocity	ft / s	100	30.5	m / s	OEM
Maximum Available Driver Power	hp	15000	11185	kW	OEM
Bearing Mechanical Loss and Casing Section Heat Loss Inputs					
Lube Oil Flow Rate (all bearings)	gal / min	75.0	0.004732	m^3 / s	OEM
Lube Oil Supply Temperature	F	120.0	48.89	C	OEM
Lube Oil Return Temperature	F	130.0	54.44	C	OEM
Lube Oil Specific Heat, Cp	BTU / (lbm R)	0.46	1.93	kJ / (kg K)	OEM
Lube Oil Density	lbm / ft^3	53.50	857.0	kg / m^3	OEM
Casing Section Surface Area	ft^2	62.00	5.76	m^2	OEM
Casing Section Surface Temperature	F	223.00	106.11	C	OEM
Casing Section Surface Emissivity		0.90	0.90		OEM
Ambient Temperature Near Casing Section	F	100.00	37.78	C	OEM

GENERAL NOTE: Blue highlight denotes specified conditions as per para. 3-1.1.

Figure C-3.2-2
Phase Diagram for Specified Gas Composition



C-3.3 Specified Performance

Performance parameters and calculations associated with specified operating conditions are shown in Table C-3.3-1. Fluid properties are calculated with the selected EOS.

The compressor designer should ensure that the chosen EOS is valid for the specified performance conditions and the test design performance conditions.

Measured pressures and temperatures at inlet and discharge in Table C-3.3-1 are related to static conditions and total conditions using the rigorous method described in Figure 5-4.4.1-1. Note that static pressures and measured pressures are equal. The calculations to develop these values are shown in Tables C-3.3-2 (Table C-3.3-2M) and C-3.3-3 (Table C-3.3-3M) for both the rigorous Mach number and alternative Mach number methods. Only small differences are seen between static conditions and total conditions due to the low inlet and discharge nozzle velocities shown in Table C-3.3-1. Nozzle and piping inside diameters were chosen to provide low fluid velocities to reduce pressure drop for the fairly dense fluid conditions.

Inlet and discharge static function values in Table C-3.3-1 are based on static absolute pressures and temperatures. Fluid velocity refers to an average velocity at the inlet and discharge measurement locations for pressures and temperatures (see subsection 2-4).

Inlet and discharge total function values in Table C-3.3-1 are based on total absolute pressures and temperatures.

Note that the derived ratios for pressure, temperature, and volume flow are based on total conditions.

Specified performance results are shown in Table C-3.3-1 for all three numerical polytropic work computational methods described in subsection 5-2. Method 3 should have the smallest value of maximum expected uncertainty according to subsection 7-4. Results from the Sandberg–Colby multistep method are used for further processing. Also listed are the convergence values for temperature and efficiency that meet the required criteria for methods 2 and 3.

Method 1 shows a higher value for polytropic work than method 3, while method 2 shows a lower value than method 3.

Heat transfer losses are shown in Table C-3.3-1. See para. C-3.12 for background regarding determination of heat transfer coefficients and heat loss calculations.

Nondimensional coefficients for method 3 are shown in Table C-3.3-1. See Table 5-6.1.2-1 for definitions of the nondimensional coefficients. Nondimensional coefficients derived from test results as shown in Table C-3.8-1 will be compared to the values shown in Table C-3.3-1 and shall meet Type 2 test limits contained in Table 3-2.1-2.

The rigorous and alternative Mach number methods described in Figures 5-4.4.1-1 and 5-4.4.2-1 were used to determine static conditions and total conditions for inlet and discharge pressures and inlet and discharge temperatures using the iterative methods shown. Five iterations are shown, but most cases should converge with fewer iterations. Tables C-3.3-2 (Table C-3.3-2M) and C-3.3-3 (Table C-3.3-3M) show calculations for the specified conditions. Similar calculations were performed for the Type 2 test design and as-tested performance data sets, but only the final results are included in Tables C-3.5-2 and C-3.7-2, respectively.

Table C-3.3-1
Specified Performance Calculations for High-Pressure Natural Gas Compressor

	U.S. Customary Units		SI Units	
Standard Conditions and Constants				
Standard Pressure	psia	14.70	101.325	kPa
Standard Temperature	F	59.00	15.00	C
Standard Density Air	lbm / ft^3	0.07651	1.2256	kg / m^3
Universal Gas Constant	ft lbf / (lbmol R)	1545.35	8.314462	kJ / (kmol K)
Gc Conversion Factor (from PTC 2)	ft lbm / (lbf s^2)	32.174	NA	
Mechanical Equivalent of Heat (from PTC 2)	ft lbf / BTU	778.169	NA	
Compressor Mechanical Details				
1st Impeller Tip Speed	f t / s	635.44	193.68	m / s
2nd Impeller Tip Speed	f t / s	635.44	193.68	m / s
3rd Impeller Tip Speed	f t / s	635.44	193.68	m / s
4th Impeller Tip Speed	f t / s	635.44	193.68	m / s
5th Impeller Tip Speed	f t / s	635.44	193.68	m / s
6th Impeller Tip Speed	f t / s	635.44	193.68	m / s
7th Impeller Tip Speed	f t / s	635.44	193.68	m / s
8th Impeller Tip Speed	f t / s	635.44	193.68	m / s
Sum Impeller Tip Speeds Squared	ft^2 / s^2	3230288	300104	m^2 / s^2
Fluid Details				
Mol Weight	lbm / lbmol	18.3265	18.3265	kg / kmol
R specific gas constant	ft lbf / (lbm R)	84.3232	0.4537	kJ / (kg K)
Density, 1 SCF	lbm / ft^3 (lbm / SCF)	0.0485	0.7771	kg / Sm^3
Volume, 1 SCF	ft^3 / lbm (SCF / lbm)	20.614	1.2869	Sm^3 / kg
Standard Mass Flow	MMSCFD	115.8	136590	Sm^3 / hr
Specific Gravity (Std Cond)		0.634	0.634	
Critical Pressure	psia	899.7	6203.1	kPa
Critical Temperature	F	-71.7	-57.6	C
	U.S. Customary Units		SI Units	
Inlet Conditions (static, measured, total)				
Temperature static	F	99.97	37.76	C
Temperature measured	F	99.99	37.77	C
Temperature Total	F	100.00	37.78	C
Pressure static absolute	psia	2519.98	17374.7	kPa
Pressure Total absolute	psia	2520.60	17378.9	kPa
Discharge Conditions (static, measured, total)				
Temperature static	F	279.97	137.76	C
Temperature measured	F	279.99	137.77	C
Temperature Total	F	280.00	137.78	C
Pressure static absolute	psia	6498.4	44804.7	kPa
Pressure Total absolute	psia	6500.0	44815.9	kPa
	U.S. Customary Units		SI Units	
Compressor Inlet Static Functions				
Density	lbm / ft^3	9.7814	156.682	kg / m^3
Specific Volume	ft^3 / lbm	0.1022	0.0064	m^3 / kg
Phase		Supercritical	Supercritical	
Dew Point	F	NA	NA	
Superheat	F	NA	NA	C
Actual Volume Flow	ACFM	398.7	0.1882	m^3 / s
Fluid Velocity	ft / s	24.24	7.388	m / s
Specific Heat, Cp	BTU / (lbm R)	0.8150	3.4123	kJ / (kg K)
Specific Heat, Cv	BTU / (lbm R)	0.4293	1.7974	kJ / (kg K)
Compressibility Factor		0.7861	0.7861	
Dynamic Viscosity	lbm / (ft s)	1.27E-05	1.89E-05	kg / (m s)
Velocity of Sound	ft / s	1527	465.3	m / s
Isentropic Exponent		1.9527	1.9527	
Nozzle Mach Number		0.0159	0.0159	
Compressor Discharge Static Functions				
Density	lbm / ft^3	13.3109	213.221	kg / m^3
Specific Volume	ft^3 / lbm	0.0751	0.0047	m^3 / kg
Actual Volume Flow	ACFM	293.0	0.1383	m^3 / s
Fluid Velocity	ft / s	33.63	10.25	m / s
Specific Heat, Cp	BTU / (lbm R)	0.7433	3.1120	kJ / (kg K)
Specific Heat, Cv	BTU / (lbm R)	0.4975	2.0829	kJ / (kg K)
Compressibility Factor		1.1272	1.1272	
Dynamic Viscosity	lbm / (ft s)	1.83E-05	2.73E-05	kg / (m s)
Velocity of Sound	ft / s	2310	704.1	m / s
Nozzle Mach Number		0.0146	0.0146	
	U.S. Customary Units		SI Units	
Compressor Inlet Total Functions				
Total Density	lbm / ft^3	9.7826	156.702	kg / m^3
Total Specific Volume	ft^3 / lbm	0.1022	0.0064	m^3 / kg
Total Volume Flow	ft^3 / min	398.7	0.1882	m^3 / s
Total Fluid Velocity	ft / s	24.23	7.387	m / s
Total Specific Enthalpy	BTU / lbm	300.1	698.0	kJ / kg
Total Specific Entropy	BTU / lbm R	0.8167	3.4192	kJ / (kg K)
Total Compressibility Factor		0.7862	0.7862	
Total Specific Heat, Cp	BTU / (lbm R)	0.8155	3.4121	kJ / (kg K)
Total Specific Heat, Cv	BTU / (lbm R)	0.4296	1.7974	kJ / (kg K)
Total Dynamic Viscosity	lbm / (ft s)	1.27E-05	1.89E-05	kg / (m s)
Total Velocity of Sound	ft / s	1527	465.4	m / s
Nozzle Mach Number		0.0159	0.0159	
Machine Mach Number		0.416	0.416	
Machine Reynolds Number		1.38E+07	1.38E+07	
Compressor Discharge Total Functions				
Total Density	lbm / ft^3	13.3123	213.243	kg / m^3
Total Specific Volume	ft^3 / lbm	0.0751	0.0047	m^3 / kg
Total Volume Flow	ft^3 / min	293.0	0.1383	m^3 / s
Total Fluid Velocity	ft / s	33.62	10.248	m / s
Total Specific Enthalpy	BTU / lbm	406.2	944.8	kJ / kg
Total Specific Entropy	BTU / (lbm R)	0.8831	3.6972	kJ / (kg K)
Total Compressibility Factor		1.1273	1.1273	
Total Specific Heat, Cp	BTU / (lbm R)	0.7433	3.1120	kJ / (kg K)
Total Specific Heat, Cv	BTU / (lbm R)	0.4975	2.0830	kJ / (kg K)
Total Dynamic Viscosity	lbm / ft s	1.83E-05	2.73E+01	kg / (m s)
Total Velocity of Sound	ft / s	2310	704.2	m / s
Nozzle Mach Number		0.0146	0.0146	
	U.S. Customary Units		SI Units	
Derived Compressor Total Functions				
Pressure Ratio		2.579	2.579	
Temperature Ratio		1.322	1.322	
Volume Ratio		1.361	1.361	
Specified Performance				
Polytropic Work Sandberg-Colby - Method 1	ft lbf / lbm	49011.58	146.499	kJ / kg
Polytropic Efficiency Method 1 (1 Step)	%	59.353	59.353	%
Polytropic Work Huntington 3 Point - Method 2	ft lbf / lbm	48985.09	146.420	kJ / kg
Polytropic Efficiency Method 2 (6 Steps)	%	59.281	59.281	%
Midpoint Temperature Convergence ≤ 1E-06		3.71E-15		
Polytropic Work Sandberg-Colby Multi-step - Method 3	ft lbf / lbm	48999.34	146.462	kJ / kg
Polytropic Efficiency Method 3 (20 Steps)	%	59.299	59.299	%
Discharge Temperature Convergence ≤ 1E-05		1.97E-08	1.11E-08	
Efficiency Convergence ≤ 1E-05		2.78E-06	2.70E-06	
	U.S. Customary Units		SI Units	
Bearing and Casing Section Heat Transfer Losses				
Bearing Mechanical Losses	hp	58.2	43.4	kW
Convective Heat Transfer Coefficient	BTU / (hr ft^2 R)	0.87	4.9	W / (m^2 K)
Section Convective Heat Transfer	BTU / hr	6603	1933	W
Section Radiative Heat Transfer	BTU / hr	11388.80	3334.4	W
Section Boundary Total Heat Loss	hp	7.07	5.27	kW
	U.S. Customary Units		SI Units	
Compressor Work and Power				
Gas Specific Work	ft lbf / lbm	82576	246.83	kJ / kg
Gas Power (Heat Balance Method)	hp	9766	7283	kW
Shaft Power (Heat Balance Method)	hp	9824	7326	kW
Specified Non-Dimensional Coefficients Based upon Sandberg-Colby Multi-step - Method 3				
Specific Volume Ratio		1.361	1.361	
Polytropic Efficiency		0.5930	0.5930	
Polytropic Work Coefficient		0.4880	0.4880	
Flow Coefficient		0.0103	0.0103	
Work Input Coefficient		0.8225	0.8225	
Total Work Input Coefficient		0.8231	0.8231	
Machine Mach Number		0.416	0.416	
Machine Reynolds Number		1.38E+07	1.38E+07	

Table C-3.3-2
Calculation of Total Conditions for Specified Performance: Inlet — U.S. Customary Units

HP Natural Gas Specified **Inlet** **U.S. Customary Units**

Input Values

Mole Weight	lbm / lbmol	18.3265
Temperature measured	F	99.989
Recovery Factor		0.65
Pressure measured static	psig static	2505.28
Barometric Pressure	psia	14.70
Mass Flow	lbm/min	3900
Nozzle Diameter	inches	7.09
Nozzle Area	ft^2	0.2742
Pressure measured absolute	psia	2519.98
Phase		Supercritical
Universal Gas Constant	BTU/ (lb mol R)	1.986
Specific Gas Constant	BTU / (lbm R)	0.10837
Gc Conversion Factor	ft lbm / (lbf s^2)	32.174
Mechanical Equivalent of Heat	ft lbf / BTU	778.169

Results

Temperature		
Static	99.970	F
Measured	99.989	F
Total	100.000	F
Pressure		
Measured static gauge	2505.28	psig
Measured static absolute	2519.98	psia
Total absolute	2520.60	psia

Rigorous Method Calculated Values

	Units	Initial Estimate	1st Iteration	2nd Iteration	3rd Iteration	4th iteration	5th Iteration
Assumed Temperature static	F	99.99	99.97968	99.97479	99.97223	99.97089	99.97019
Assumed Entropy at probe	BTU / (lbm R)	0.817240	0.817226	0.817219	0.817215	0.817213	0.817212
Density	lbm / ft^3	9.780655	9.780998	9.781177	9.781271	9.781320	9.781346
Volume Flow	ft^3 / min	398.7	398.7	398.7	398.7	398.7	398.7
Nozzle Velocity	ft / s	24.24	24.24	24.24	24.24	24.24	24.24
Acoustic Velocity	ft / s	1526.68	1526.67	1526.67	1526.67	1526.67	1526.67
Mach Number		0.0159	0.0159	0.0159	0.0159	0.0159	0.0159
Kinetic Energy	BTU / lbm	0.011734	0.011733	0.011733	0.011732	0.011732	0.011732
Enthalpy measured	BTU / lbm	300.2774	300.2734	300.2713	300.2702	300.2696	300.2694
Enthalpy static	BTU / lbm	300.2698	300.2658	300.2637	300.2626	300.2620	300.2617
Temperature static new	F	99.97968	99.97479	99.97223	99.97089	99.97019	99.96982
Enthalpy total	BTU / lbm	300.2815	300.2775	300.2754	300.2743	300.2738	300.2735
Entropy static	BTU / (lbm R)	0.817226	0.817219	0.817215	0.817213	0.817212	0.817212
Temperature static difference	F	0.00935	0.00489	0.00256	0.00134	0.00070	0.00037
Total Pressure	psia	2520.6001	2520.6000	2520.6000	2520.6000	2520.6000	2520.6000
Total Temperature	F	100.0099	100.0050	100.0024	100.0011	100.0004	100.0000

Alternative Mach Number Method Calculated Values

	Units	Initial Estimate	1st Iteration	2nd Iteration	3rd Iteration	4th iteration	5th Iteration
Assumed Temperature static	F	99.99	99.96942	99.96942	99.96942	99.96942	99.96942
Compressibility		0.786176	0.786145	0.786145	0.786145	0.786145	0.786145
Specific Heat at Constant Pressure	BTU / (lbm R)	0.81552	0.81555	0.81555	0.81555	0.81555	0.81555
Specific Heat at Constant Volume	BTU / (lbm R)	0.42958	0.42958	0.42958	0.42958	0.42958	0.42958
Acoustic Velocity	ft / s	1526.68	1526.67	1526.67	1526.67	1526.67	1526.67
Density	lbm / ft^3	9.780655	9.781374	9.781374	9.781374	9.781374	9.781374
Isentropic Exponent		1.95252	1.95265	1.95265	1.95265	1.95265	1.95265
Compressibility Function X		1.09800	1.09813	1.09813	1.09813	1.09813	1.09813
Volume Flow	ft^3 / min	398.7	398.7	398.7	398.7	398.7	398.7
Nozzle Velocity	ft / s	24.24	24.24	24.24	24.24	24.24	24.24
Mach Number		0.0159	0.0159	0.0159	0.0159	0.0159	0.0159
Theta Bracket Value		1.000120	1.000120	1.000120	1.000120	1.000120	1.000120
Theta Exponent Value		0.449277	0.449235	0.449235	0.449235	0.449235	0.449235
Theta		1.000035	1.000035	1.000035	1.000035	1.000035	1.000035
Temperature static New	F	99.96942	99.96942	99.96942	99.96942	99.96942	99.96942
Temperature static difference	F	0.01962	0.00000	0.00000	0.00000	0.00000	0.00000
Total Pressure	psia	2520.6005	2520.6004	2520.6004	2520.6004	2520.6004	2520.6004
Total Temperature	F	99.9996	99.9996	99.9996	99.9996	99.9996	99.9996

Table C-3.3-2M
Calculation of Total Conditions for Specified Performance: Inlet — SI Units

HP Natural Gas Specified **Inlet** **SI Units**

Input Values

Mole Weight	kg / kmol	18.3265
Temperature measured	C	37.772
Recovery Factor		0.65
Pressure measured static	kPag static	17273.30
Barometric Pressure	kPa	101.35
Mass Flow	kg / s	29.484
Nozzle Diameter	m	0.1801
Nozzle Area	m^2	0.0255
Pressure measured absolute	kPa	17374.65
Phase		Supercritical
Universal Gas Constant	kJ / (kmol K)	8.314
Specific Gas Constant	kJ / (kg K)	0.45369

Results

Temperature		
Static	37.761	C
Measured	37.772	C
Total	37.778	C
Pressure		
Measured static gauge	17273.30	kPag
Measured static absolute	17374.65	kPa
Total absolute	17378.93	kPa

Rigorous Method Calculated Values

	Units	Initial Estimate	1st Iteration	2nd Iteration	3rd Iteration	4th iteration	5th Iteration
Assumed Temperature static	C	37.77	37.76649	37.76377	37.76234	37.76160	37.76121
Assumed Entropy at probe	kJ / (kmol K)	3.4193	3.419274	3.419244	3.419229	3.419220	3.419216
Density	kg / m^3	156.67	156.68	156.68	156.68	156.68	156.68
Volume Flow	m^3 / s	0.18819	0.18818	0.18818	0.18818	0.18817	0.18817
Nozzle Velocity	m / s	7.39	7.39	7.39	7.39	7.39	7.39
Acoustic Velocity	m / s	464.21	464.21	464.21	464.21	464.21	464.21
Mach Number		0.0159	0.0159	0.0159	0.0159	0.0159	0.0159
Kinetic Energy	kJ / kg	0.027293	0.027291	0.027290	0.027290	0.027289	0.027289
Enthalpy measured	kJ / kg	697.9781	697.9688	697.9640	697.9614	697.9601	697.9594
Enthalpy static	kJ / kg	697.9604	697.9511	697.9462	697.9437	697.9424	697.9417
Temperature static new	C	37.76649	37.76377	37.76234	37.76160	37.76121	37.76100
Enthalpy total	kJ / kg	697.9877	697.9784	697.9735	697.9710	697.9697	697.9690
Entropy static	kJ / (kmol K)	3.419274	3.419244	3.419229	3.419220	3.419216	3.419214
Temperature static difference	C	0.00520	0.00272	0.00142	0.00075	0.00039	0.00020
Total Pressure	kPa	17378.9278	17378.9276	17378.9275	17378.9274	17378.9274	17378.9273
Total Temperature	C	37.7833	37.7805	37.7791	37.7784	37.7780	37.7778

Alternative Mach Number Method Calculated Values

	Units	Initial Estimate	1st Iteration	2nd Iteration	3rd Iteration	4th iteration	5th Iteration
Assumed Temperature static	C	37.77	37.76078	37.76078	37.76078	37.76078	37.76078
Compressibility		0.786176	0.786145	0.786145	0.786145	0.786145	0.786145
Specific Heat at Constant Pressure	kJ / (kg K)	3.41212	3.41226	3.41226	3.41226	3.41226	3.41226
Specific Heat at Constant Volume	kJ / (kg K)	1.79738	1.79737	1.79737	1.79737	1.79737	1.79737
Acoustic Velocity	m / s	465.33	465.33	465.33	465.33	465.33	465.33
Density	kg / m^3	156.671052	156.682583	156.682582	156.682582	156.682582	156.682582
Isentropic Exponent		1.95262	1.95265	1.95265	1.95265	1.95265	1.95265
Compressibility Function X		1.09800	1.09813	1.09813	1.09813	1.09813	1.09813
Volume Flow	m^3 / s	0.18819	0.18817	0.18817	0.18817	0.18817	0.18817
Nozzle Velocity	m / s	7.39	7.39	7.39	7.39	7.39	7.39
Mach Number		0.0159	0.0159	0.0159	0.0159	0.0159	0.0159
Theta Bracket Value		1.000120	1.000120	1.000120	1.000120	1.000120	1.000120
Theta Exponent Value		0.449550	0.449508	0.449508	0.449508	0.449508	0.449508
Theta		1.000035	1.000035	1.000035	1.000035	1.000035	1.000035
Temperature static New	C	37.76078	37.76078	37.76078	37.76078	37.76078	37.76078
Temperature static difference	C	0.01091	0.00000	0.00000	0.00000	0.00000	0.00000
Total Pressure	kPa	17378.9277	17378.9273	17378.9273	17378.9273	17378.9273	17378.9273
Total Temperature	C	37.7776	37.7776	37.7776	37.7776	37.7776	37.7776

Table C-3.3-3
Calculation of Total Conditions for Specified Performance: Discharge — U.S. Customary Units

HP Natural Gas Specified **Discharge** **U.S. Customary Units**

Input Values

Mole Weight	lbm / lbmol	18.3265
Temperature measured	F	279.988
Recovery Factor		0.65
Pressure measured static	psig static	6483.68
Barometric Pressure	psia	14.70
Mass Flow	lbm / min	3900
Nozzle Diameter	inches	5.16
Nozzle Area	ft^2	0.1452
Pressure measured absolute	psia	6498.38
Phase		Supercritical
Universal Gas Constant	BTU/ (lb mol R)	1.986
Specific Gas Constant	BTU / (lbm R)	0.10837
Gc Conversion Factor	ft lbm / (lbf s^2)	32.174
Mechanical Equivalent of Heat	ft lbf / BTU	778.169

Results

Temperature		
Static	279.966	F
Measured	279.988	F
Total	280.000	F
Pressure		
Measured static gauge	6483.68	psig
Measured static absolute	6498.38	psia
Total absolute	6500.00	psia

Rigorous Method Calculated Values

	Units	Initial Estimate	1st Iteration	2nd Iteration	3rd Iteration	4th iteration	5th Iteration
Assumed Temperature static	F	279.99	279.96827	279.96602	279.96576	279.96573	279.96573
Assumed Entropy at probe	BTU / (lbm R)	0.883671	0.883651	0.883649	0.883648	0.883648	0.883648
Density	lbm / ft^3	13.310469	13.310870	13.310916	13.310921	13.310922	13.310922
Volume Flow	ft^3 / min	293.0	293.0	293.0	293.0	293.0	293.0
Nozzle Velocity	ft / s	33.63	33.63	33.63	33.63	33.63	33.63
Acoustic Velocity	ft / s	2310.03	2310.04	2310.04	2310.04	2310.04	2310.04
Mach Number		0.0146	0.0146	0.0146	0.0146	0.0146	0.0146
Kinetic Energy	BTU / lbm	0.022583	0.022581	0.022581	0.022581	0.022581	0.022581
Enthalpy measured	BTU / lbm	406.4546	406.4529	406.4527	406.4527	406.4527	406.4527
Enthalpy static	BTU / lbm	406.4399	406.4383	406.4381	406.4380	406.4380	406.4380
Temperature static new	F	279.96827	279.96602	279.96576	279.96573	279.96573	279.96573
Enthalpy total	BTU / lbm	406.4625	406.4608	406.4606	406.4606	406.4606	406.4606
Entropy static	BTU / (lbm R)	0.883651	0.883649	0.883648	0.883648	0.883648	0.883648
Temperature static difference	F	0.01974	0.00225	0.00026	0.00003	0.00000	0.00000
Total Pressure	psia	6500.0001	6500.0000	6500.0000	6500.0000	6500.0000	6500.0000
Total Temperature	F	280.0025	280.0003	280.0000	280.0000	280.0000	280.0000

Alternative Mach Number Method Calculated Values

	Units	Initial Estimate	1st Iteration	2nd Iteration	3rd Iteration	4th iteration	5th Iteration
Assumed Temperature static	F	279.99	279.96573	279.96573	279.96573	279.96573	279.96573
Compressibility		1.127183	1.127178	1.127178	1.127178	1.127178	1.127178
Specific Heat at Constant Pressure	BTU / (lbm R)	0.74379	0.74379	0.74379	0.74379	0.74379	0.74379
Specific Heat at Constant Volume	BTU / (lbm R)	0.49784	0.49783	0.49783	0.49783	0.49783	0.49783
Acoustic Velocity	ft / s	2310.03	2310.04	2310.04	2310.04	2310.04	2310.04
Density	lbm / ft^3	13.310469	13.310922	13.310922	13.310922	13.310922	13.310922
Isentropic Exponent		2.35914	2.35925	2.35925	2.35925	2.35925	2.35925
Compressibility Function X		0.12887	0.12887	0.12887	0.12887	0.12887	0.12887
Volume Flow	ft^3 / min	293.0	293.0	293.0	293.0	293.0	293.0
Nozzle Velocity	ft / s	33.63	33.63	33.63	33.63	33.63	33.63
Mach Number		0.0146	0.0146	0.0146	0.0146	0.0146	0.0146
Theta Bracket Value		1.000144	1.000144	1.000144	1.000144	1.000144	1.000144
Theta Exponent Value		0.321795	0.321784	0.321784	0.321784	0.321784	0.321784
Theta		1.000030	1.000030	1.000030	1.000030	1.000030	1.000030
Temperature static New	F	279.96573	279.96573	279.96573	279.96573	279.96573	279.96573
Temperature static difference	F	0.02228	0.00000	0.00000	0.00000	0.00000	0.00000
Total Pressure	psia	6500.0011	6500.0011	6500.0011	6500.0011	6500.0011	6500.0011
Total Temperature	F	280.0000	280.0000	280.0000	280.0000	280.0000	280.0000

Table C-3.3-3M
Calculation of Total Conditions for Specified Performance: Discharge — SI Units

HP Natural Gas Specified **Discharge** **SI Units**

Input Values

Mole Weight	kg / kmol	18.3265
Temperature measured	C	137.771
Recovery Factor		0.650
Pressure measured static	kPag static	44703.38
Barometric Pressure	kPa	101.35
Mass Flow	kg / s	29.484
Nozzle Diameter	m	131.06
Nozzle Area	m^2	0.0135
Pressure measured absolute	kPa	44804.73
Phase		Supercritical
Universal Gas Constant	kJ / (kmol K)	8.314
Specific Gas Constant	kJ / (kg K)	0.45369

Results

Temperature		
Static	137.759	C
Measured	137.771	C
Total	137.778	C
Pressure		
Measured static gauge	44703.38	kPag
Measured static absolute	44804.73	kPa
Total absolute	44815.93	kPa

Rigorous Method Calculated Values

	Units	Initial Estimate	1st Iteration	2nd Iteration	3rd Iteration	4th iteration	5th Iteration
Assumed Temperature static	C	137.77	137.76014	137.75889	137.75875	137.75873	137.75873
Assumed Entropy at probe	kJ / (kmol K)	3.6973	3.697195	3.697186	3.697185	3.697184	3.697184
Density	kg / m^3	213.21	213.22	213.22	213.22	213.22	213.22
Volume Flow	m^3 / s	0.13828	0.13828	0.13828	0.13828	0.13828	0.13828
Nozzle Velocity	m / s	10.25	10.25	10.25	10.25	10.25	10.25
Acoustic Velocity	m / s	703.30	703.31	703.31	703.31	703.31	703.31
Mach Number		0.0146	0.0146	0.0146	0.0146	0.0146	0.0146
Kinetic Energy	kJ / kg	0.052528	0.052524	0.052524	0.052524	0.052524	0.052524
Enthalpy measured	kJ / kg	944.7812	944.7773	944.7768	944.7768	944.7768	944.7768
Enthalpy static	kJ / kg	944.7470	944.7431	944.7427	944.7426	944.7426	944.7426
Temperature static new	C	137.76014	137.75889	137.75875	137.75873	137.75873	137.75873
Enthalpy total	kJ / kg	944.7995	944.7957	944.7952	944.7952	944.7952	944.7952
Entropy static	kJ / (kmol K)	3.69720	3.69719	3.69718	3.69718	3.69718	3.69718
Temperature static difference	C	0.01097	0.00125	0.00014	0.00002	0.00000	0.00000
Total Pressure	kPa	44815.9287	44815.9281	44815.9280	44815.9280	44815.9280	44815.9280
Total Temperature	C	137.7792	137.7779	137.7778	137.7778	137.7778	137.7778

Alternative Mach Number Method Calculated Values

	Units	Initial Estimate	1st Iteration	2nd Iteration	3rd Iteration	4th iteration	5th Iteration
Assumed Temperature static	C	137.77	137.75873	137.75873	137.75873	137.75873	137.75873
Compressibility		1.127183	1.127178	1.127178	1.127178	1.127178	1.127178
Specific Heat at Constant Pressure	kJ / (kg K)	3.11204	3.11201	3.11201	3.11201	3.11201	3.11201
Specific Heat at Constant Volume	kJ / (kg K)	2.08296	2.08293	2.08293	2.08293	2.08293	2.08293
Acoustic Velocity	m / s	704.10	704.10	704.10	704.10	704.10	704.10
Density	kg / m^3	213.213257	213.220511	213.220511	213.220511	213.220511	213.220511
Isentropic Exponent		2.35914	2.35925	2.35925	2.35925	2.35925	2.35925
Compressibility Function X		0.12887	0.12888	0.12888	0.12888	0.12888	0.12888
Volume Flow	m^3 / s	0.13828	0.13828	0.13828	0.13828	0.13828	0.13828
Nozzle Velocity	m / s	10.25	10.25	10.25	10.25	10.25	10.25
Mach Number		0.0146	0.0146	0.0146	0.0146	0.0146	0.0146
Theta Bracket Value		1.000144	1.000144	1.000144	1.000144	1.000144	1.000144
Theta Exponent Value		0.321990	0.321979	0.321979	0.321979	0.321979	0.321979
Theta		1.000030	1.000030	1.000030	1.000030	1.000030	1.000030
Temperature static New	C	137.75873	137.75873	137.75873	137.75873	137.75873	137.75873
Temperature static difference	C	0.01238	0.00000	0.00000	0.00000	0.00000	0.00000
Total Pressure	kPa	44815.9284	44815.9280	44815.9280	44815.9280	44815.9280	44815.9280
Total Temperature	C	137.7778	137.7778	137.7778	137.7778	137.7778	137.7778

C-3.4 Test Gas Selection Screening and Test Design

(a) Test Conditions Versus Specified Conditions

(1) General. Operating conditions for a Type 2 test must conform to test stand physical limits, which can be lower than specified conditions, especially for pressure, temperature, speed, and power. The high-pressure natural gas compressor specified inlet pressure of 17 375 kPa (2,520 psia) would require a very expensive test stand arrangement to duplicate site conditions. A much lower suction pressure can be used with a different test gas to obtain equivalent performance by applying methods of similitude.

(2) Equality of Coefficients

(-a) An acceptably designed Type 2 test must satisfy equality of test versus specified nondimensional coefficients within the limits shown in Table 3-2.1-2.

(-1) specific volume ratio: $r_{v,t} = r_{v,sp} \pm 5\%$

(-2) flow coefficient: $\phi_t = \phi_{sp} \pm 4\%$

(-3) machine Mach number: between upper and lower limits shown in Figure 3-2.1-1

(-4) machine Reynolds number: between upper and lower limits shown in Figure 3-2.1-3

(-b) Equality of the following two nondimensional coefficients can also be used in the test design process as follows:

(-1) polytropic efficiency: $\eta_t = \eta_{sp}$

(-2) polytropic work coefficient: $\mu_{p,t} = \mu_{p,sp}$

(b) Selection of Test Gas

(1) General. A test gas composition must be selected to begin the design of the Type 2 test. This is usually an inert gas or a mixture of inert gases. For this high-pressure natural gas sample case, four fluids were screened to find a suitable test composition: carbon dioxide, nitrogen, a 50/50 mol% mixture of carbon dioxide and nitrogen, and refrigerant R134a. Once a test gas is selected as outlined in (c), refined calculations will be performed to determine target values for test design performance.

(2) Required Variables. For this sample calculation, it is assumed that all compressor hardware-related dimensions are identical between specified conditions and test conditions. The following six variables need to be determined for each of the four potential test gases:

(-a) inlet total pressure

(-b) inlet total temperature

(-c) discharge total pressure

(-d) discharge total temperature

(-e) compressor inlet mass flow

(-f) compressor rotating speed

(3) Procedure. The following steps can be used to determine acceptable values for the six variables in (2).

Step 1. Choose inlet conditions to accommodate easily achievable values within normal test stand capabilities and safety. Inlet pressure should be above atmospheric pressure to ensure no atmospheric air leakage into the system. Inlet temperature should be well within the capabilities of any inlet cooler when accounting for ambient temperatures and expected heat load.

Step 2. Once inlet conditions are set, adjust the test discharge conditions to achieve equal nondimensional values for specific volume ratio and polytropic efficiency. The discharge total pressure and temperature can be found simultaneously using a classical nonlinear optimization algorithm such as can be found in common mathematical software packages. Judicious initial guesses for the discharge condition values could speed up the solution process.

Step 3. Check the resulting discharge conditions against compressor and test stand limits. If any discharge pressure or temperature limits are exceeded, the inlet conditions must be revised or the gas composition changed before proceeding.

Step 4. Set the compressor test speed by equating the specified and test polytropic work coefficients and solving for test speed.

Step 5. Determine the mass flow rate that will satisfy specified and test flow coefficient equality. Perform machine Reynolds number and machine Mach number checks for specified and test conditions to confirm satisfactory design of a Type 2 test according to Figures 3-2.1-1 and 3-2.1-3.

Step 6. Reexamine compressor and test stand mechanical limits in light of the proposed test design parameters.

Step 7. If any required test criteria are not satisfied, modify inlet conditions and repeat the process or investigate using another test gas composition.

(c) Selected Gas: Carbon Dioxide. Table C-3.4-1 shows a successfully designed Type 2 test for the high-pressure natural gas compressor that will use carbon dioxide for the test gas. The table lists not only potential test conditions but also the specified performance values that must be matched within the allowable limits imposed by Table 3-2.1-2. The specified conditions are repeated from Table C-3.3-1 in the left column of values. Calculations for the Type 2 test design are shown in the middle column of values. The gas composition and test design input adjustable variables are highlighted in this column.

The right column shows the ratio of test/specified values for the nondimensional performance parameters. Each is seen to be equal to 1.00 within a reasonable tolerance, which signifies a well-designed Type 2 test.

At the bottom of the table, the resulting test design values are compared to appropriate minimum and maximum acceptable values for performance and mechanical aspects. All proposed test values are acceptable, however, a refined analysis of test conditions and test design performance with carbon dioxide is included in subsection C-3.5.

(d) Comparison of Test Gas Compositions. Table C-3.4-2 provides a comparison of the four Type 2 test gas compositions that were investigated to be used for the high-pressure natural gas case.

(1) Nitrogen was rejected as a test gas because the test Reynolds number was below the acceptable range.

(2) Carbon dioxide was accepted as the test gas.

(3) The mix of 50% carbon dioxide and 50% nitrogen (mol%) was rejected because the required operating speed was too close to the compressor's first critical speed according to rotor-dynamic restraints that are beyond the purview of ASME PTC 10.

(4) Refrigerant R134a was rejected because the test Reynolds number was below the acceptable range. Also, the test inlet-to-discharge temperature rise was less than 50°F and the power consumed during the test was very low. These two items could lead to increased uncertainty.

(e) Parasitic Losses. Parasitic losses are not considered in this preliminary test design but are included in the refined test design shown in Table C-3.5-1. Total pressures and temperatures must be used in these calculations.

Table C-3.4-1
Type 2 Preliminary Test Design

HP Natural Gas Case		U.S. Customary Units Specified	U.S Customary Units Test Design	Test / Specified
Potential Test Gas			CO2	
1st Impeller Diameter	inches	13.636	13.636	
1st Impeller Tip Width	inches	0.34	0.34	
Number of Impellers		8	8	
Inlet Pressure Total Absolute	psia	2520.60	300.00	
Inlet Total Temperature	F	100.00	100.00	
Inlet Total Specific Volume	ft^3 / lbm	0.10222	0.41160	
Inlet Total Enthalpy	BTU / lbm	300.073	214.420	
Inlet Total Entropy	BTU / (lbm R)	0.81667	0.51625	
Inlet Total Viscosity	lbm / ft s	1.27E-05	1.06E-05	
Inlet Speed of Sound	ft / s	1526.83	852.66	
Inlet Gas Phase		Supercritical	Superheated gas	
Inlet Gas Dew Point	F	NA	-1.12	
Inlet Gas Superheat	F	NA	101.124	
Discharge Pressure Total Absolute	psia	6500.00	487.76	
Discharge Total Temperature	F	280.00	201.59	
Discharge Total Specific Volume	ft^3 / lbm	0.07512	0.30247	
Discharge Total Enthalpy	BTU / lbm	406.189	234.888	
Discharge Total Entropy		0.88306	0.52988	
Specific Volume Ratio	vd / vi	1.36082	1.36082	1.00000
Pressure Ratio		2.57875	1.62585	
Total Enthalpy Change (Specific Gas Work)	BTU / lbm	106.116	20.469	
Total Entropy Change		0.06639	0.01363	
Average Inlet & Discharge Temp	R	649.670	610.463	
Polytropic Work	ft lb f /lbm	49011.58	9453.81	
Polytropic Efficiency		0.59353	0.59353	1.00000
Compressor Rotating Speed	rev / min	10680.0	4691.2	
Inlet Mass Flow	lbm / min	3900.0	425.4	
Inlet Volume Flow	ft^3 /min	398.67	175.11	
Inlet Nozzle Velocity		24.24	20.10	
Tip Speed 1st Impeller	ft / s	635.44	279.12	
SUM of Tip Speeds ^2	ft^2 / s^2	3230288	623249	
Gas Power (Neglecting Heat Transfer)	hp	9759.0	205.3	
Polytropic Work Coefficient		0.48804	0.48804	1.00000
Flow Coefficient		0.01031	0.01031	1.00000
Work Input Coefficient		0.82247	0.82226	0.99970
Machine Mach Number		0.4162	0.3273	
Machine Reynolds Number		1.38E+07	1.81E+06	

		Minimum Acceptable Value		Test Design Value		Maximum Acceptable Value
Volume Ratio Allowable Range	%	95.00	OK	100.00	OK	105.00
Flow Coefficient Allowable Range	%	96.00	OK	100.00	OK	104.00
Machine Reynolds Number Minimum Value		9.00E+04	OK	1.81E+06		NA
Reynolds Number Allowable Range		1.38E+06	OK	1.81E+06	OK	1.38E+09
Mach Number Allowable Range		0.26	OK	0.33	OK	0.60
Maximum Continuous Speed	rev / min	NA		4691	OK	11500
1st Critical Speed Avoidance Range	rev / min	4860	OK	4691	OK	5940
2nd Critical Speed Avoidance Range	rev / min	12330	OK	4691	OK	15070
Maximum Allowable Working Pressure	psia	NA		487.76	OK	8300.00
Working Temperature Allowable Range	F	-20.00	OK	100 to 202	OK	380.00
Maximum Allowable Nozzle Flow Velocity	ft / s	NA		20.10	OK	100.00
Minimum Allowable Inlet Fluid Superheat	F	5.00	OK	101.12		NA
Maximum Available Driver Power	hp	NA		1065	OK	2500

Table C-3.4-2
Screening Criteria Comparison for Potential Test Gas Compositions

HP Natural Gas Case

		U.S. Customary Units Test Design	U.S. Customary Units Test Design	U.S. Customary Units Test Design N2+CO2;50/50	U.S. Customary Units Test Design
Potential Test Gas		N2	CO2	N2;CO2|.5;.5	R134a
1st Impeller Diameter	inches	13.636	13.636	13.636	13.636
1st Impeller Tip Width	inches	0.34	0.34	0.34	0.34
Number of Impellers		8	8	8	8
Inlet Pressure Total Absolute	psia	300.00	300.00	300.00	100.00
Inlet Total Temperature	F	100.00	100.00	100.00	100.00
Inlet Total Specific Volume	ft^3 / lbm	0.71386	0.41160	0.53511	0.51076
Inlet Total Enthalpy	BTU / lbm	136.995	214.420	185.355	182.595
Inlet Total Entropy	BTU / (lbm R)	1.42629	0.51625	0.90982	0.41903
Inlet Total Viscosity	lbm / ft s	1.25E-05	1.06E-05	1.18E-05	8.23E-06
Inlet Speed of Sound	ft / s	1191.75	852.66	1001.08	491.56
Inlet Gas Phase		Superheated gas	Superheated gas	Superheated gas	Superheated gas
Inlet Gas Dew Point	F	-250.45	-1.12	-45.32	79.16
Inlet Gas Superheat	F	350.450	101.124	145.321	20.841
Discharge Pressure Total Absolute	psia	555.53	487.76	513.73	140.38
Discharge Total Temperature	F	291.15	201.59	234.42	132.08
Discharge Total Specific Volume	ft^3 / lbm	0.52458	0.30247	0.39323	0.37534
Discharge Total Enthalpy	BTU / lbm	184.938	234.888	215.573	188.084
Discharge Total Entropy	BTU / (lbm R)	1.45603	0.52988	0.92941	0.42291
Specific Volume Ratio	vd / vi	1.36082	1.36082	1.36082	1.36082
Pressure Ratio		1.85176	1.62585	1.71242	1.40384
Total Enthalpy Change	BTU / lbm	47.943	20.469	30.218	5.489
Total Entropy Change	BTU / (lbm R)	0.02974	0.01363	0.01959	0.00388
Average Inlet & Discharge Temp	R	655.247	610.463	626.882	575.711
Polytropic Work	ft lb f /lbm	22143.15	9453.81	13956.78	2535.01
Polytropic Efficiency		0.59353	0.59353	0.59353	0.59353
Compressor Rotating Speed	rev / min	7179.5	4691.2	5699.9	2429.2
Inlet Mass Flow	lbm / min	375.4	425.4	397.6	177.5
Inlet Volume Flow	ft^3 /min	268.00	175.11	212.77	90.68
Inlet Nozzle Velocity		30.76	20.10	24.42	10.41
Tip Speed 1st Impeller	ft / s	427.17	279.12	339.13	144.53
SUM of Tip Speeds ^2	ft^2 / s^2	1459790	623249	920098	167121
Gas Power (Neglecting Heat Transfer)	hp	424.4	205.3	283.3	23.0
Polytropic Work Coefficient		0.48804	0.48804	0.48804	0.48804
Flow Coefficient		0.01031	0.01031	0.01031	0.01031
Work Input Coefficient		0.82226	0.82226	0.82227	0.82226
Machine Mach Number		0.3584	0.3273	0.3388	0.2940
Machine Reynolds Number		1.35E+06	1.81E+06	1.53E+06	9.74E+05

			N2 Test Design Value			CO2 Test Design Value			N2;CO2|.5;.5 Test Design Value			R134a Test Design Value	
Volume Ratio Allowable Range	%	OK	100.00	OK	OK	100.00	OK	OK	100.00	OK	OK	100.00	OK
Flow Coefficient Allowable Range	%	OK	100.00	OK	OK	100.00	OK	OK	100.00	OK	OK	100.00	OK
Reynolds Number Allowable Range		NOT OK	1.35E+06	OK	OK	1.81E+06	OK	OK	1.53E+06	OK	NOT OK	9.74E+05	OK
Mach Number Allowable Range		OK	0.36	OK	OK	0.33	OK	OK	0.34	OK	OK	0.29	OK
Maximum Continuous Speed	rev / min		7180	OK		4691	OK		5700	OK		2429	OK
1st Critical Speed Avoidance Range	rev / min	OK	7180	OK	OK	4691	OK	NOT OK	5700	NOT OK	OK	2429	OK
2nd Critical Speed Avoidance Range	rev / min	OK	7180	OK	OK	4691	OK	OK	5700	OK	OK	2429	OK
Maximum Allowable Working Pressure	psia		555.53	OK		487.76	OK		513.73	OK		140.38	OK
Working Temperature Allowable Range	F	OK	100 to 291	OK	OK	100 to 202	OK	OK	100 to 234	OK	OK	100 to 132	OK
Maximum Allowable Nozzle Flow Velocity	ft / s		30.76	OK		20.10	OK		24.42	OK		10.41	OK
Minimum Allowable Inlet Fluid Superheat	F	OK	350.45		OK	101.12		OK	145.32		OK	20.84	
Maximum Available Driver Power	hp		424	OK		205	OK		283	OK		23	OK

C-3.5 Test Design Performance With Carbon Dioxide Test Gas

The Type 2 test design parameters are shown in Table C-3.5-1, and the test fluid's phase diagram is shown in Figure C-3.5-1. Test conditions do not encroach on the two-phase region for the carbon dioxide test gas. Table C-3.5-2 shows the calculations for the compressor's predicted performance when tested.

Table C-3.5-1
Test Design for High-Pressure Natural Gas Compressor Using Carbon Dioxide

	U.S. Customary Units		SI Units	
Standard Conditions				
Standard Pressure	psia	14.696	101.325	kPa
Standard Temperature	F	59.00	15.00	C
Operating Conditions				
Project Identifier		HP Natural Gas	HP Natural Gas	
Condition Name		CO2 Test Design	CO2 Test Design	
Barometric Pressure	psia	14.700	101.353	kPa
Ambient Temperature	F	65.00	18.33	C
Inlet Pressure measured	psig	285.28	1966.9	kPag
Inlet Temperature measured	F	100.00	37.78	C
Inlet Temperature Probe Recovery Factor		0.65	0.65	
Discharge Pressure measured	psig	472.98	3261.1	kPag
Discharge Temperature measured	F	201.58	94.21	C
Discharge Temperature Probe Recovery Factor		0.65	0.65	
Rotor Speed	rev / min	4691	78.12	1 / s
Inlet Mass Flow	lbm / min	425.4	3.214	kg / s
Fluid Composition				
Carbon Dioxide	mole fraction	1.0000	1.0000	mole fraction
Compressor Section Hardware Details				
Compressor Model / Section Label		HP Section	HP Section	
Inlet Pipe Inside Diameter	inches	7.090	0.18	m
Discharge Pipe Inside Diameter	inches	5.160	0.13	m
Number of Impellers		8	8	
1st Impeller Diameter	inches	13.636	0.3464	m
2nd Impeller Diameter	inches	13.636	0.3464	m
3rd Impeller Diameter	inches	13.636	0.3464	m
4th Impeller Diameter	inches	13.636	0.3464	m
5th Impeller Diameter	inches	13.636	0.3464	m
6th Impeller Diameter	inches	13.636	0.3464	m
7th Impeller Diameter	inches	13.636	0.3464	m
8th Impeller Diameter	inches	13.636	0.3464	m
1st Impeller Tip Width	inches	0.34	0.00864	m
Flow Path Surface Roughness	inches	0.000125	3.175E-06	m
Compressor Design Limits				
Maximum Continuous Speed	rev / min	11500	191.67	1 / s
1st Critical Speed	rev / min	5400	90.00	1 / s
2nd Critical Speed	rev / min	13700	228.33	1 / s
Maximum Allowable Working Pressure	psia	8300	57226	kPa
Maximum Allowable Temperature	F	380	193	C
Minimum Allowable Temperature	F	-20	-29	C
Maximum Allowable Nozzle Flow Velocity	ft / s	100	30.5	m / s
Maximum Available Driver Power	hp	1200	895	kW
Bearing Mechanical Loss and Casing Section Heat Loss Inputs				
Lube Oil Flow Rate (all bearings)	gal / min	75.0	0.00473	m^3 / s
Lube Oil Supply Temperature	F	120.0	48.89	C
Lube Oil Return Temperature	F	127.0	52.78	C
Lube Oil Specific Heat, Cp	BTU / (lbm R)	0.46	1.93	kJ / (kg K)
Lube Oil Density	lbm / ft^3	53.50	857.0	kg / m^3
Casing Section Surface Area	ft^2	62.00	5.76	m^2
Casing Section Surface Temperature	F	180.00	82.22	C
Casing Section Surface Emissivity		0.90	0.90	
Ambient Temperature Near Casing Section	F	90.00	32.22	C

Figure C-3.5-1
Phase Diagram for Test Gas Composition

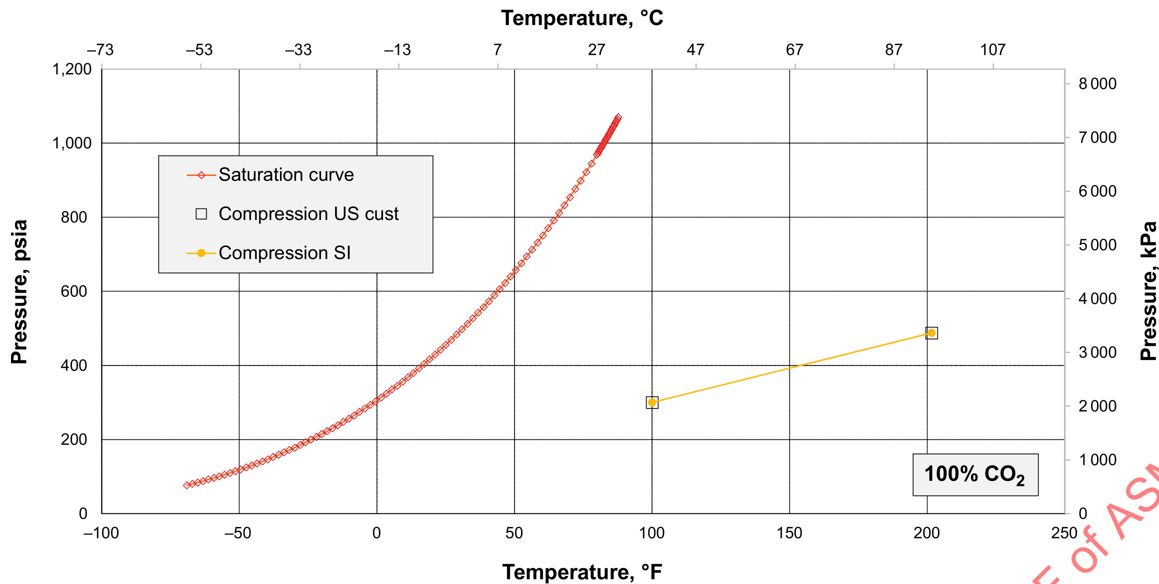


Table C-3.5-2
Test Design Performance Calculations for High-Pressure Natural Gas Compressor Using Carbon Dioxide

Table C-3.5-2
Test Design Performance Calculations for High-Pressure Natural Gas Compressor Using Carbon Dioxide

	U.S. Customary Units		SI Units	
Standard Conditions and Constants				
Standard Pressure	psia	14.70	101.325	kPa
Standard Temperature	F	59.00	15.00	C
Standard Density Air	lbm / ft^3	0.07651	1.2256	kg / m^3
Universal Gas Constant	ft lbf / (lbmol R)	1545.35	8.314462	kJ / (kmol K)
Gc Conversion Factor (from PTC 2)	ft lbm / (lbf s^2)	32.174	NA	
Mechanical Equivalent of Heat (from PTC 2)	ft lbf / BTU	778.169	NA	
Compressor Mechanical Details				
1st Impeller Tip Speed	f t / s	279.11	85.07	m / s
2nd Impeller Tip Speed	f t / s	279.11	85.07	m / s
3rd Impeller Tip Speed	f t / s	279.11	85.07	m / s
4th Impeller Tip Speed	f t / s	279.11	85.07	m / s
5th Impeller Tip Speed	f t / s	279.11	85.07	m / s
6th Impeller Tip Speed	f t / s	279.11	85.07	m / s
7th Impeller Tip Speed	f t / s	279.11	85.07	m / s
8th Impeller Tip Speed	f t / s	279.11	85.07	m / s
Sum Impeller Tip Speeds Squared	ft^2 / s^2	623203	57897	m^2 / s^2
Fluid Details				
Mol Weight	lbm / lbmol	44.0098	44.0098	kg / kmol
R specific gas constant	ft lbf / (lbm R)	35.1138	0.1889	kJ / (kg K)
Density, 1 SCF	lbm / ft^3 (lbm / SCF)	0.1169	1.8719	kg / Sm^3
Volume, 1 SCF	ft^3 / lbm (SCF / lbm)	8.558	0.5342	Sm^3 / kg
Standard Mass Flow	MMSCFD	5.2	6181	Sm^3 / hr
Specific Gravity (Std Cond)		1.527	1.527	
Critical Pressure	psia	1070.0	7377.3	kPa
Critical Temperature	F	87.8	31.0	C
Inlet Conditions (static, measured, total)				
Temperature static	F	99.99	37.77	C
Temperature measured	F	100.00	37.78	C
Temperature Total	F	100.00	37.78	C
Pressure static absolute	psia	299.98	2068.3	kPa
Pressure Total absolute	psia	300.01	2068.5	kPa
Discharge Conditions (static, measured, total)				
Temperature static	F	201.56	94.20	C
Temperature measured	F	201.58	94.21	C
Temperature Total	F	201.59	94.22	C
Pressure static absolute	psia	487.68	3362.4	kPa
Pressure Total absolute	psia	487.76	3363.0	kPa

	U.S. Customary Units		SI Units	
Compressor Inlet Static Functions				
Density	lbm / ft^3	2.4294	38.916	kg / m^3
Specific Volume	ft^3 / lbm	0.4116	0.0257	m^3 / kg
Phase		Superheated gas	Superheated gas	
Dew Point	F	-1.1276	-18.4042	
Superheat	F	101.11	56.17	C
Actual Volume Flow	ACFM	175.1	4.96	m^3 / min
Fluid Velocity	ft / s	10.64	3.244	m / s
Specific Heat, Cp	BTU / (lbm R)	0.2416	1.0113	kJ / (kg K)
Specific Heat, Cv	BTU / (lbm R)	0.1708	0.7152	kJ / (kg K)
Compressibility Factor		0.9048	0.9048	
Dynamic Viscosity	lbm / (ft s)	1.06E-05	1.58E-05	Pa s
Velocity of Sound	ft / s	853	259.9	m / s
Isentropic Exponent		1.2708	1.2708	
Nozzle Mach Number		0.0125	0.0125	
Compressor Discharge Static Functions				
Density	lbm / ft^3	3.3057	52.953	kg / m^3
Specific Volume	ft^3 / lbm	0.3025	0.0189	m^3 / kg
Actual Volume Flow	ACFM	128.7	3.64	m^3 / min
Fluid Velocity	ft / s	14.77	4.50	m / s
Specific Heat, Cp	BTU / (lbm R)	0.2504	1.0484	kJ / (kg K)
Specific Heat, Cv	BTU / (lbm R)	0.1812	0.7585	kJ / (kg K)
Compressibility Factor		0.9149	0.9149	
Dynamic Viscosity	lbm / (ft s)	1.26E-05	1.88E-05	Pa s
Velocity of Sound	ft / s	928	283.0	m / s
Nozzle Mach Number		0.0159	0.0159	
Compressor Inlet Total Functions				
Total Density	lbm / ft^3	2.4296	38.919	kg / m^3
Total Specific Volume	ft^3 / lbm	0.4116	0.0257	m^3 / kg
Total Volume Flow	ft^3 / min	175.1	4.96	m^3 / min
Total Fluid Velocity	ft / s	10.64	3.244	m / s
Total Specific Enthalpy	BTU / lbm	214.4	498.7	kJ / kg
Total Specific Entropy	BTU / (lbm R)	0.5163	2.1614	kJ / (kg K)
Total Compressibility Factor		0.9048	0.9048	
Total Specific Heat, Cp	BTU / (lbm R)	0.2417	1.0114	kJ / (kg K)
Total Specific Heat, Cv	BTU / (lbm R)	0.1709	0.7152	kJ / (kg K)
Total Dynamic Viscosity	lbm / (ft s)	1.06E-05	1.58E-05	Pa s
Total Velocity of Sound	ft / s	853	259.9	m / s
Nozzle Mach Number		0.0125	0.0125	
Machine Mach Number		0.327	0.327	
Machine Reynolds Number		1.81E+06	1.81E+06	
Compressor Discharge Total Functions				
Total Density	lbm / ft^3	3.3062	52.960	kg / m^3
Total Specific Volume	ft^3 / lbm	0.3025	0.0189	m^3 / kg
Total Volume Flow	ft^3 / min	128.6	3.64	m^3 / min
Total Fluid Velocity	ft / s	14.76	4.498	m / s
Total Specific Enthalpy	BTU / lbm	234.9	546.4	kJ / kg
Total Specific Entropy	BTU / (lbm R)	0.5299	2.2185	kJ / (kg K)
Total Compressibility Factor		0.9149	0.9149	
Total Specific Heat, Cp	BTU / (lbm R)	0.2504	1.0484	kJ / (kg K)
Total Specific Heat, Cv	BTU / (lbm R)	0.1812	0.7585	kJ / (kg K)
Total Dynamic Viscosity	lbm / (ft s)	1.26E-05	1.88E+01	Pa s
Total Velocity of Sound	ft / s	928	283.0	m / s
Nozzle Mach Number		0.0159	0.0159	
Derived Compressor Total Functions				
Pressure Ratio		1.626	1.626	
Temperature Ratio		1.182	1.182	
Volume Ratio		1.361	1.361	

	U.S. Customary Units		SI Units	
Test Design Performance				
Polytropic Work Sandberg-Colby Method 1	ft lbf / lbm	9453.24	28.256	kJ / kg
Polytropic Efficiency Method 1 (1 Step)	%	59.347	59.347	%
Polytropic Work Huntington Method 2	ft lbf / lbm	9467.28	28.298	kJ / kg
Polytropic Efficiency Method 2 (3 points, 2 steps)	%	59.396	59.396	%
Midpoint Temperature Convergence ≤ 1E-06		0.00E+00		
Polytropic Work Sandberg-Colby Multi-step Method 3	ft lbf / lbm	9469.88	28.306	kJ / kg
Polytropic Efficiency Method 3 (20 Steps)	%	59.412	59.412	%
Discharge Temperature Convergence ≤ 1E-05		1.55E-06	2.15E-06	
Efficiency Convergence ≤ 1E-05		3.47E-06	4.04E-06	
Bearing and Casing Section Heat Transfer Losses				
Bearing Mechanical Losses	hp	40.7	30.4	kW
Convective Heat Transfer Coefficient	BTU / (hr ft^2 R)	0.81	4.6	KW / (m^2 K)
Section Convective Heat Transfer	BTU / hr	4495	1316	W
Section Radiative Heat Transfer	BTU / hr	7282	2132	W
Section Boundary Total Heat Loss	hp	4.63	3.45	kW
Compressor Work and Power				
Specific Gas Work	ft lbf / lbm	15929	47.61	kJ / kg
Gas Power (Heat Balance Method)	hp	210	156	kW
Shaft Power (Heat Balance Method)	hp	251	187	kW
Test Design Non-Dimensional Coefficients Based upon Sandberg-Colby Multi-step Method 3				
Specific Volume Ratio		1.361	1.361	
Polytropic Efficiency		0.5941	0.5941	
Polytropic Work Coefficient		0.4889	0.4889	
Flow Coefficient		0.0103	0.0103	
Work Input Coefficient		0.8223	0.8223	
Total Work Input Coefficient		0.8409	0.8409	
Machine Mach Number		0.327	0.327	
Machine Reynolds Number		1.81E+06	1.81E+06	

C-3.6 Measured Test Data and Processing

Data collected and recorded in the course of compressor testing was processed according to the procedure outlined in para. 5-4.1 prior to calculating performance results. It is assumed that successful end-to-end signal calibrations were performed prior to test initiation. Raw data observations for each performance input variable were recorded from each single- or multiple-measuring instrument (probe) and at least three sequential readings of all instruments were required. If three or more instruments are used for a single variable such as inlet static pressure, the individual observations shall be investigated for outliers (see ASME PTC 19.1, Nonmandatory Appendix A). All readings must be investigated for excessive fluctuations (see Table 3-12.2-1 and para. 5-4.3.3).

Paragraph C-3.6.1 contains detailed samples of the processing sequence for inlet measured static gauge pressure and inlet measured temperature according to the procedure outlined in para. 5-4.1. Paragraph C-3.6.2 shows the measured and processed test point data that has been prepared for use in test performance calculations. These values can be compared to the test design values in Table C-3.5-2.

C-3.6.1 Test Inlet Pressure and Temperature Processing. Table C-3.6.1-1 shows the necessary information for test inlet pressure and temperature processing. The inlet pressure sample calculation is described in para. C-3.6.1.1 and the inlet temperature sample calculation is described in para. C-3.6.1.2.

C-3.6.1.1 Inlet Pressure Calculation Sample. The test stand's data acquisition system has performed any necessary calibrations and corrections as required by para. 5-4.3.1 such that the values in Table C-3.6.1.1-1 represent the corrected raw data for five readings collected sequentially over a 5-min period after stable operation was achieved according to para. 3-10.3. Note that the observations for reading 5 have a much larger range than the other readings.

Barometric pressure has been added to the gauge pressure observations to obtain absolute pressures as shown in Table C-3.6.1.1-2. Absolute values must be used to test for outliers and fluctuations.

The calculations shown in Table C-3.6.1.1-3 are based on the modified Thompson Tau technique discussed in ASME PTC 19.1, Nonmandatory Appendix A, which is intended to identify observations that are more than two standard deviations away from the mean. Values for τ can be found in ASME PTC 19.1.

Raw pressure data observations excluding outliers for five readings are shown in Table C-3.6.1.1-4. Verification of fluctuation compliance for the five readings is shown in Table C-3.6.1.1-5.

Actual fluctuation values for each reading are calculated according to para. 5-4.3.3 and are expressed as percentages. Permissible fluctuation percentages are listed in Table 3-12.2-1. If a reading's actual fluctuation is less than the permissible fluctuation, the reading is acceptable. In Table C-3.6.1.1-5, readings 1 through 4 are acceptable and reading 5 is not acceptable. Reading 5 is excluded from consideration to determine a test point value for compressor inlet absolute test pressure.

The remaining readings (1 through 4) are averaged to determine the test point value of 2 069.26 kPa (300.12 psia) as shown in Table C-3.6.1.1-6. If outliers and fluctuations had not been addressed, the test point value would have been 2 069.60 kPa (300.71 psia), introducing an error of 0.2% for this example.

The compressor inlet absolute test pressure value must be converted to total conditions in a similar manner as shown in Table C-3.3-1 (see also Nonmandatory Appendices A and G).

Table C-3.6.1-1
Necessary Information

Variable Being Processed	Number of Probes	Number of Readings	Barometric Pressure, in. Hg (psia)
Inlet static gauge pressure	4	5	29.726 (14.70)
Inlet measured temperature	4	5	29.726 (14.70)

Table C-3.6.1.1-1
Recorded Raw Data Observations for Compressor Inlet Static Pressure for a Test Point

	Measured p_i Gauge Pressure Test Observations, psig				
Probe Number	Reading 1	Reading 2	Reading 3	Reading 4	Reading 5
1	285.29	285.48	285.52	285.28	284.83
2	285.66	286.09	285.47	285.39	285.82
3	285.47	285.58	285.39	285.31	290.82
4	285.35	285.36	285.50	285.27	291.39

Table C-3.6.1.1-2
Raw Data Observations Converted to Absolute Values

Probe Number	p_i Absolute Pressure Test Observations, psig				
	Reading 1	Reading 2	Reading 3	Reading 4	Reading 5
1	299.9	300.18	300.22	299.98	299.53
2	300.36	300.79	300.17	300.09	300.52
3	300.17	300.28	300.09	300.01	305.52
4	300.05	300.06	300.20	299.97	306.09

Table C-3.6.1.1-3
Test for and Remove Outliers on a per-Reading Basis

		Reading 1	Reading 2	Reading 3	Reading 4	Reading 5
Average of All Observations (per Reading)	psia	300.14	300.33	300.17	300.01	302.92
S_x Standard Deviation (per Reading)	psia	0.1625	0.3205	0.0572	0.0537	3.3695
τ from Table A-2.2-1 of PTC 19.1		1.425	1.425	1.425	1.425	1.425
τ^*S_x (per Reading)		0.2316	0.4567	0.0814	0.0766	4.8016
δ 1 = ABS VAL(Observation 1 - Average)		0.1533	0.1458	0.0500	0.0300	3.3850
δ 2 = ABS VAL(Observation 2 - Average)		0.2167	0.4608	0.0000	0.0767	2.3950
δ 3 = ABS VAL(Observation 3 - Average)		0.0267	0.0458	0.0800	0.0033	2.6050
δ 4 = ABS VAL(Observation 4 - Average)		0.0900	0.2692	0.0300	0.0433	3.1750
δ 1 - τ^*S_x		-0.0783	-0.3109	-0.0314	-0.0466	-1.4166
δ 2 - τ^*S_x		-0.0150	0.0041	-0.0814	0.0001	-2.4066
δ 3 - τ^*S_x		-0.2050	-0.4109	-0.0014	-0.0733	-2.1966
δ 4 - τ^*S_x		-0.1416	-0.1876	-0.0514	-0.0333	-1.6266
Observation 1 Check for Outliers		OK	OK	OK	OK	OK
Observation 2 Check for Outliers		OK	Outlier	OK	Outlier	OK
Observation 3 Check for Outliers		OK	OK	OK	OK	OK
Observation 4 Check for Outliers		OK	OK	OK	OK	OK

Legend:

S_x = standard deviation for a reading

δ = the absolute difference between an individual observation and the average for a reading

τ = a function of the number of probes being used to measure a single variable

GENERAL NOTES:

(a) A positive or zero value for each sum ($\delta - \tau S_x$) indicates the observation is an outlier. Outliers are marked with a yellow highlight for reading 2, probe 2 and reading 4, probe 2. These observations are eliminated from the respective readings before further processing. Any rejected observations shall be clearly stated in the test report.

(b) If only one or two probes are used for a variable, it is not possible to identify outliers, and data processing should skip forward to verification of compliance with allowable fluctuations on a per-reading basis.

Table C-3.6.1.1-4
Raw Data Observations Excluding Outliers

Observation With Outliers Excluded	Reading 1, psia	Reading 2, psia	Reading 3, psia	Reading 4, psia	Reading 5, psia
1	299.99	300.18	300.22	299.98	299.53
2	300.36	Outlier	300.17	Outlier	300.52
3	300.17	300.28	300.09	300.01	305.52
4	300.05	300.06	300.20	299.97	306.09

Table C-3.6.1.1-5
Verification of Fluctuation Compliance

Reading	Averages Excluding Outliers, psia	Actual Reading Fluctuation, %	Permissible Reading Fluctuation, %	Acceptable Readings, psia
1	300.14	0.12	2.00	300.14
2	300.18	0.07	2.00	300.18
3	300.17	0.04	2.00	300.17
4	299.99	0.01	2.00	299.99
5	302.92	2.17	2.00	0.00

Table C-3.6.1.1-6
Determination of Test Point Value, p_i, Excluding Outliers and Rejected Readings Due to Fluctuations

Absolute Pressure, psia (kPa)	Static Pressure, psig (kPa)
300.12 (2 069.26)	285.42 (1 967.90)

C-3.6.1.2 Inlet Temperature Calculation Sample. The data acquisition system has performed any necessary calibrations and corrections as required by para. 5-4.3.1 such that the values in Table C-3.6.1.2-1 represent the corrected raw data for five readings collected sequentially over a 5-min period after stable operation was achieved according to para. 3-10.3.

A value of 273.15°C (459.67°F) has been added to the measured temperature observations to obtain absolute temperatures in kelvin (degrees Rankine) as shown in Table C-3.6.1.2-2. Absolute values must be used to test for outliers and fluctuations.

The calculations shown in Table C-3.6.1.2-3 are based on the modified Thompson Tau technique discussed in ASME PTC 19.1, Nonmandatory Appendix A, which is intended to identify observations that are more than two standard deviations away from the mean. Values for τ can be found in ASME PTC 19.1.

Raw temperature data observations excluding outliers for five readings are shown in Table C-3.6.1.2-4. Verification of fluctuation compliance for the five readings is shown in Table C-3.6.1.2-5.

Actual fluctuation values for each reading are calculated according to para. 5-4.3.3 and are expressed as percentages. Permissible fluctuation percentages are listed in Table 3-12.2-1. If a reading's actual fluctuation is less than the permissible fluctuation, the reading is acceptable. In Table C-3.6.1.2-5, readings 1 and 3 through 5 are acceptable and reading 2 is not acceptable. Reading 2 is excluded from consideration to determine a test point value for compressor inlet measured test temperature.

The remaining readings (1 and 3 through 5) are averaged to determine the test point value of 310.93 K (559.68°R) as shown in Table C-3.6.1.2-6. If outliers and fluctuations had not been addressed, the test point value would have been 310.93 K (559.67°R).

The compressor inlet absolute temperature value must now be converted to total conditions in a similar manner as shown in Table C-3.3-1 (see also Nonmandatory Appendices A and G).

Table C-3.6.1.2-1
Recorded Raw Data Observations for Compressor Inlet Measured Temperature for a Test Point

	Inlet Measured Temperature T_i Test Data, °F				
Probe Number	Reading 1	Reading 2	Reading 3	Reading 4	Reading 5
1	100.85	100.95	100.85	100.35	99.85
2	99.20	99.65	99.85	99.85	99.85
3	99.35	99.85	100.15	99.85	100.25
4	100.15	98.85	99.55	99.85	100.85

Table C-3.6.1.2-2
Converted Raw Data Observations to Absolute Values

	Inlet Absolute Temperature T_i Test Data, °R				
Probe Number	Reading 1	Reading 2	Reading 3	Reading 4	Reading 5
1	560.52	560.62	560.52	560.02	559.52
2	558.87	559.32	559.52	559.52	559.52
3	559.02	559.52	559.82	559.52	559.92
4	559.82	558.52	559.22	559.52	560.52

Table C-3.6.1.2-3
Test for and Remove Outliers on a per-Reading Basis

		Reading 1	Reading 2	Reading 3	Reading 4	Reading 5
Average of All Observations (per Reading)	R	559.56	559.50	559.77	559.65	559.87
S_x Standard Deviation (per Reading)	R	0.7641	0.8643	0.5560	0.2496	0.4719
τ from Table A-2.2-1 of PTC 19.1		1.425	1.425	1.425	1.425	1.425
τ^*S_x (per Reading)		1.0889	1.2316	0.7922	0.3557	0.6724
δ 1 = ABS VAL(Observation 1 - Average)		0.9611	1.1233	0.7489	0.3744	0.3495
δ 2 = ABS VAL(Observation 2 - Average)		0.6865	0.1747	0.2496	0.1248	0.3495
δ 3 = ABS VAL(Observation 3 - Average)		0.5367	0.0250	0.0499	0.1248	0.0499
δ 4 = ABS VAL(Observation 4 - Average)		0.2621	0.9736	0.5492	0.1248	0.6490
δ 1 - τ^*S_x		-0.1278	-0.1082	-0.0433	0.0187	-0.3229
δ 2 - τ^*S_x		-0.4024	-1.0568	-0.5426	-0.2309	-0.3229
δ 3 - τ^*S_x		-0.5522	-1.2066	-0.7423	-0.2309	-0.6225
δ 4 - τ^*S_x		-0.8268	-0.2580	-0.2430	-0.2309	-0.0234
Observation 1 Check for Outliers		OK	OK	OK	Outlier	OK
Observation 2 Check for Outliers		OK	OK	OK	OK	OK
Observation 3 Check for Outliers		OK	OK	OK	OK	OK
Observation 4 Check for Outliers		OK	OK	OK	OK	OK

Legend:

S_x = the standard deviation for a reading

δ = the absolute difference between an individual observation and the average for a reading

τ = a function of the number of probes being used to measure a single variable

GENERAL NOTES:

(a) A positive or zero value for each sum ($\delta - \tau S_x$) indicates the observation is an outlier. An outlier is marked with a yellow highlight for reading 4, probe 1. This observation is eliminated from the respective reading before further processing. Any rejected observations shall be clearly stated in the test report.

(b) If only one or two probes are used for a variable, it is not possible to identify outliers, and data processing should skip forward to verification of compliance with allowable fluctuations on a per-reading basis.

Table C-3.6.1.2-4
Raw Data Observations Excluding Outliers

Observation With Outliers Excluded	Reading 1, °R	Reading 2, °R	Reading 3, °R	Reading 4, °R	Reading 5, °R
1	560.52	560.62	560.52	Outlier	559.52
2	558.87	559.32	559.52	559.52	559.52
3	559.02	559.52	559.82	559.52	559.92
4	559.82	558.52	559.22	559.52	560.52

Table C-3.6.1.2-5
Verification of Fluctuation Compliance

Reading	Averages Excluding Outliers, °R	Actual Reading Fluctuation, %	Permissible Reading Fluctuation, %	Acceptable Readings, °R
1	559.56	0.29	0.30	559.56
2	559.50	0.37	0.30	0.00
3	559.77	0.23	0.30	559.77
4	559.52	0.00	0.30	559.52
5	559.87	0.18	0.30	559.87

Table C-3.6.1.2-6
Determination of Test Point Value, T_i, Excluding Outliers and Rejected Readings Due to Fluctuations

Absolute Measured Temperature, °R (K)	Measured Temperature, °F (°C)
559.68 (310.93)	100.01 (37.78)

C-3.6.2 Test Data Results Prepared for As-Tested Performance Calculations. Table C-3.6.2-1 lists the measured, processed, and recorded performance input for the Type 2 test point when the compressor was operated using carbon dioxide. Compressor hardware dimensions are the same as the specified design.

Mass flows for any seal leakage or balance piston flows have not been included due to the initial assumption that the inlet, rotor, and discharge mass flows are equal.

Comparison of the as-tested data point values to the test design values shows the compressor speed was slightly lower and the mass flow was slightly higher than predicted.

Table C-3.6.2-1
Type 2 As-Tested Data Point

	U.S. Customary Units		SI Units	
Type 2 Test Operating Conditions				
Condition Name		CO2 As Tested	CO2 As Tested	
Barometric Pressure	psia	14.70	101.35	kPa
Ambient Temperature	F	65.00	18.33	C
Inlet Pressure measured	psig	285.28	1966.94	kPag
Inlet Temperature measured	F	100.01	37.78	C
Discharge Pressure measured	psig	474.00	3268.11	kPag
Discharge Temperature measured	F	203.10	95.06	C
Rotor Speed	rev / min	4676	77.93	1 / s
Inlet Mass Flow	lbm / min	435	3.289	kg / s
Fluid Composition				
Carbon Dioxide	mole fraction	1.0000	1.0000	mole fraction
Type 2 Test Bearing Mechanical and Casing Section Heat Loss Inputs				
Lube Oil Flow Rate (all bearings)	gal / min	70.0	0.265	m^3 / s
Lube Oil Supply Temperature	F	118.0	47.78	C
Lube Oil Return Temperature	F	126.0	52.22	C
Lube Oil Specific Heat Cp	BTU / (lbm R)	0.46	1.93	kJ / (kg K)
Lube Oil Density	lbm / ft^3	53.5	857.0	kg / m^3
Casing Section Surface Area	ft^2	62.0	5.76	m^2
Casing Section Surface Temperature	F	184.0	84.44	C
Casing Section Surface Emissivity		0.90	0.90	
Ambient Temperature Near Casing Section	F	103.0	39.44	C

C-3.7 Type 2 As-Tested Performance

Based on the test data listed in Table C-3.6.2-1, the compressor test performance was calculated in the same manner as shown in Table C-3.5-2. A comparison of test design and as-tested inlet and discharge conditions is shown in Table C-3.7-1.

As-tested inlet conditions are very close to test design. However, as-tested discharge conditions show higher temperatures and pressures than test design. Table C-3.7-2 illustrates a comparison of test design and as-tested performance.

Minor differences between test design and as-tested values are seen for polytropic work and efficiency; these differences are due to the slightly elevated discharge conditions mentioned earlier. Somewhat larger percentage differences are seen for parasitic losses and power; however, these are reasonable values and do not invalidate the test.

The as-tested section heat loss is 1.6% of the shaft power, which meets the requirement of para. 4-15.3(d) of less than 5%.

Table C-3.7-1
Comparison of Test Design and As-Tested Inlet and Discharge Conditions

	Test Design		As Tested		Difference
	U.S. Customary Units		U.S. Customary Units		
Inlet Conditions (static, measured, total)					
Temperature static	deg F	99.99	100.00	deg F	0.015
Temperature measured	deg F	100.00	100.01	deg F	0.015
Temperature Total	deg F	100.00	100.01	deg F	0.015
Pressure static absolute	psia	299.98	299.98	psia	0.000
Pressure Total absolute	psia	300.01	300.01	psia	0.001
Discharge Conditions (static, measured, total)					
Temperature static	deg F	201.56	203.08	deg F	1.519
Temperature measured	deg F	201.58	203.10	deg F	1.520
Temperature Total	deg F	201.59	203.11	deg F	1.520
Pressure static absolute	psia	487.68	488.70	psia	1.020
Pressure Total absolute	psia	487.76	488.78	psia	1.024
	SI Units		SI Units		
Inlet Conditions (static, measured, total)					
Temperature static	deg C	37.77	37.78	deg C	0.008
Temperature measured	deg C	37.78	37.78	deg C	0.008
Temperature Total	deg C	37.78	37.79	deg C	0.008
Pressure static absolute	kPa	2068.3	2068.3	kPa	0.000
Pressure Total absolute	kPa	2068.5	2068.5	kPa	0.010
Discharge Conditions (static, measured, total)					
Temperature static	deg C	94.20	95.05	deg C	0.844
Temperature measured	deg C	94.21	95.06	deg C	0.844
Temperature Total	deg C	94.22	95.06	deg C	0.845
Pressure static absolute	kPa	3362.4	3369.5	kPa	7.033
Pressure Total absolute	kPa	3363.0	3370.0	kPa	7.058

Table C-3.7-2
Comparison of Test Design and As-Tested Performance

	Test Design Performance		As Tested Performance		Difference
	U.S. Customary Units		U.S. Customary Units		%
Test Design Performance					
Polytropic Work Sandberg-Colby Method 1	ft lbf / lbm	9453.24	9507.96	ft lbf / lbm	0.58
Polytropic Efficiency Method 1 (1 Step)	%	59.347	58.669	%	-0.68
Polytropic Work Huntington Method 2	ft lbf / lbm	9467.28	9522.62	ft lbf / lbm	0.58
Polytropic Efficiency Method 2 (3 points, 2 steps)	%	59.396	58.720	%	-0.68
Midpoint Temperature Convergence ≤ 1E-06		0.00E+00	3.73E-16		
Polytropic Work Sandberg-Colby Multi-step Method 3	ft lbf / lbm	9469.88	9525.37	ft lbf / lbm	0.59
Polytropic Efficiency Method 3 (20 Steps)	%	59.412	58.737	%	-0.68
Discharge Temperature Convergence ≤ 1E-05		1.55E-06	2.04E-06		
Efficiency Convergence ≤ 1E-05		3.47E-06	4.50E-06		
Bearing and Casing Section Heat Transfer Losses					
Bearing Mechanical Losses	hp	40.7	43.4	hp	6.67
Convective Heat Transfer Coefficient	BTU / (hr ft^2 R)	0.81	0.77	BTU / (hr ft^2 R)	-4.22
Section Convective Heat Transfer	BTU / hr	4495	3875	BTU / hr	-13.79
Section Radiative Heat Transfer	BTU / hr	7282	6831	BTU / hr	-6.20
Section Boundary Total Heat Loss	hp	4.63	4.21	hp	-9.10
Compressor Work and Power					
Specific Gas Work	ft lbf / lbm	15929	16206.24	ft lbf / lbm	1.74
Gas Power (Heat Balance Method)	hp	210	218	hp	3.82
Shaft Power (Heat Balance Method)	hp	251	261	hp	4.28

	SI Units		SI Units		%
Polytropic Work Sandberg-Colby Method 1	kJ / kg	28.256	28.420	kJ / kg	0.58
Polytropic Efficiency Method 1 (1 Step)	%	59.347	58.669	%	-0.68
Polytropic Work Huntington Method 2	kJ / kg	28.298	28.464	kJ / kg	0.58
Polytropic Efficiency Method 2 (3 points, 2 steps)	%	59.396	58.720	%	-0.68
Midpoint Temperature Convergence ≤ 1E-06					
Polytropic Work Sandberg-Colby Multi-step Method 3	kJ / kg	28.306	28.472	kJ / kg	0.59
Polytropic Efficiency Method 3 (20 Steps)	%	59.412	58.737	%	-0.68
Discharge Temperature Convergence ≤ 1E-05		2.15E-06	1.99E-06		
Efficiency Convergence ≤ 1E-05		4.04E-06	3.70E-06		
Bearing and Casing Heat Transfer Losses					
Bearing Mechanical Losses	kW	30.4	32.4	kW	6.67
Convective Heat Transfer Coefficient	KW / (m^2 K)	4.6	4.4	KW / (m^2 K)	-4.22
Section Convective Heat Transfer	W	1316	1135	W	-13.79
Section Radiative Heat Transfer	W	2132	2000	W	-6.20
Section Boundary Total Heat Loss	kW	3.45	3.13	kW	-9.10
Compressor Work and Power					
Gas Specific Work	kJ / kg	47.61	48.44	kJ / kg	1.74
Gas Power (Heat Balance Method)	kW	156	162	kW	3.82
Shaft Power (Heat Balance Method)	kW	187	195	kW	4.28

C-3.8 As-Tested Nondimensional Performance

The nondimensional Type 2 as-tested results are shown in Table C-3.8-1. The coefficients are based on the equations in Table 5-6.1.2-1.

Table C-3.8-1
As-Tested Nondimensional Results

As Tested Non-Dimensional Coefficients Based upon Sandberg-Colby Multi-step Method 3	U.S. Customary Units	SI Units
Specific Volume Ratio	1.360	1.360
Flow Coefficient	0.0106	0.0106
Work Input Coefficient	0.8421	0.8421
Polytropic Work Coefficient	0.4949	0.4949
Polytropic Efficiency	0.5874	0.5874
Total Work Input Coefficient	0.8586	0.8586
Machine Mach Number	0.326	0.326
Machine Reynolds Number	1.80E+06	1.80E+06

C-3.9 Type 2 Test Confirmation

The Type 2 test results must meet the criteria listed in Table 3-2.1-2 to ensure the compressor test results conform to this Code's requirements and can be used to determine site performance at specified conditions. Table C-3.9-1 lists the specified and as-tested values for the four operating parameters. All four are within acceptable deviation ranges. Thus, a successful Type 2 test is confirmed.

Figures C-3.9-1 through C-3.9-3 provide graphical confirmation of the Type 2 test results. Figure C-3.9-1 shows the tested and specified flow coefficient ratios versus the tested and specified ratios of specific volume ratios. An acceptance region is shown inside the blue box. The red dot is the actual data from the specified conditions and as-tested results. The graph shows the as-tested flow coefficient was approximately 3% high and the as-tested volume ratio was about 0.1% low compared to specified values. This Code allows deviations within the region encompassed by the blue box since small departures from true model similitude are considered acceptable if other Type 2 test criteria are met. Figures C-3.9-2 and C-3.9-3 show the as-tested machine Mach number and machine Reynolds number superimposed on copies of Figures 3-2.1-1 and 3-2.1-3, respectively. In both cases, the as-tested values are seen to appropriately lie between the lower and upper limit lines.

Table C-3.9-1
Specified and As-Tested Operating Parameters

Parameter	Criteria From Table 3-2.1-2	Specified Values	As Tested Results: Minimum Acceptable	As Tested Results: Tested Values	As Tested Results: Maximum Acceptable	Acceptable?
Specific Volume Ratio	Specified +/- 5%	1.361	1.293	1.360	1.429	Yes
Flow Coefficient	Specified +/-4%	0.0103	0.0099	0.0106	0.0107	Yes
Machine Mach Number	See Figure 3-2-1	0.416	0.256	0.326	0.598	Yes
Machine Reynolds Number	See Figure 3-2-3	1.38E+07	1.38E+06	1.80E+06	1.38E+09	Yes

Figure C-3.9-1
As-Tested Flow Coefficient and Volume Ratio

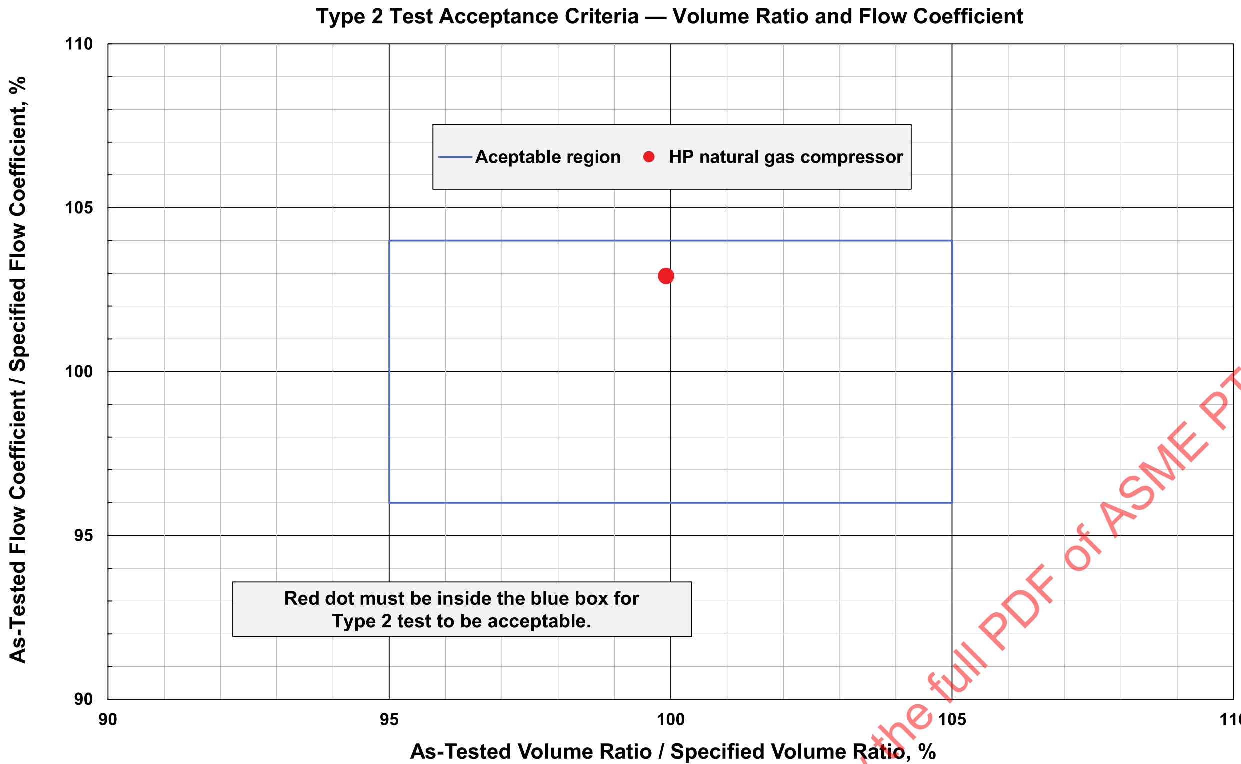


Figure C-3.9-2
As-Tested Machine Mach Number

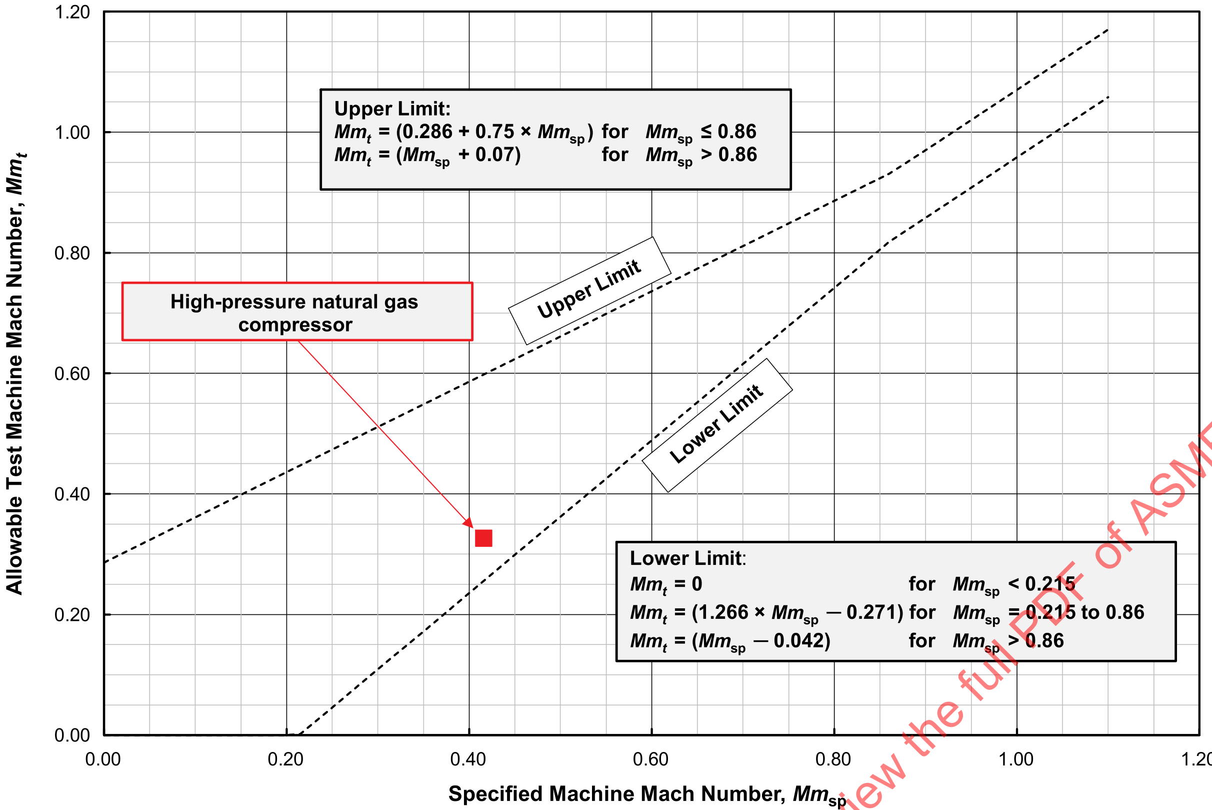
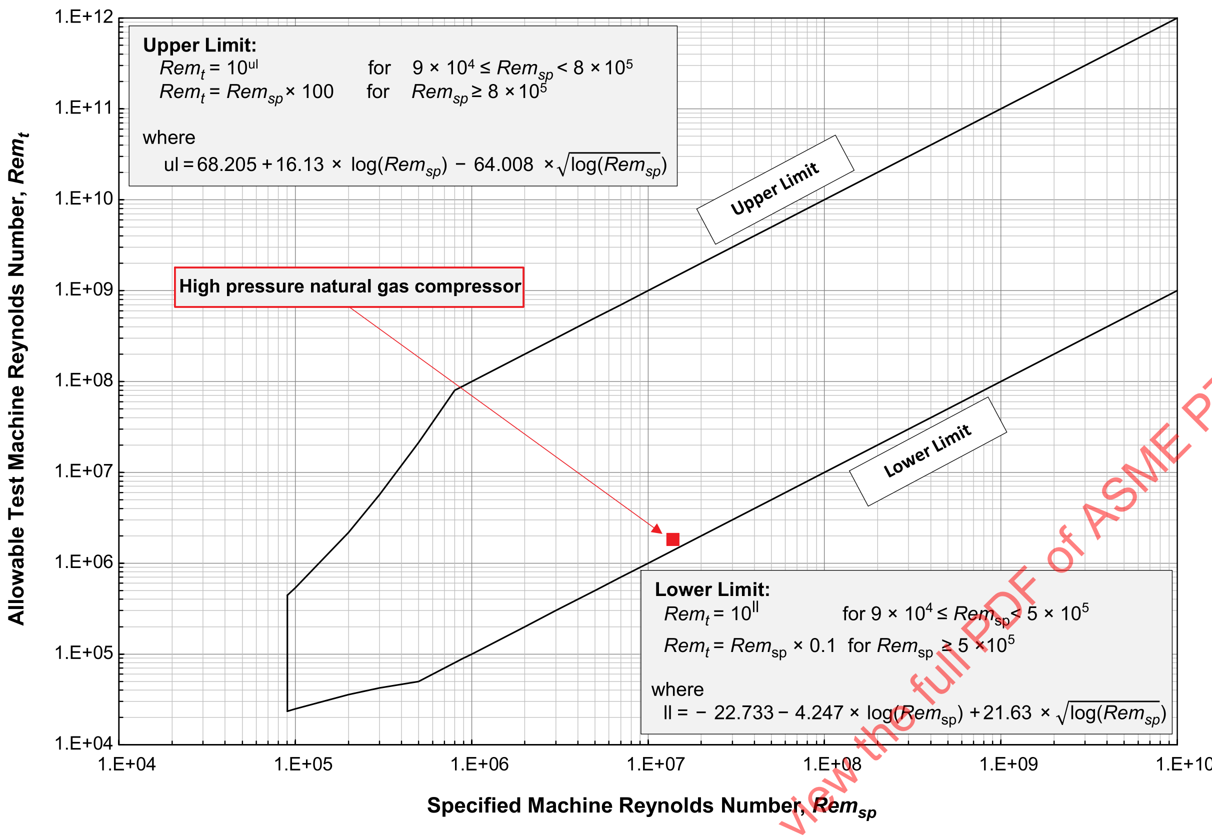


Figure C-3.9-3
As-Tested Machine Reynolds Number



C-3.10 Type 2 Test Reynolds Number Corrections

C-3.10.1 As-Tested Reynolds Number Corrections. Testing at a Reynolds number different from the specified value requires corrections to be applied to as-tested values for the following nondimensional coefficients:

(a) flow coefficient
(b) work input coefficient
(c) polytropic work coefficient
(d) polytropic efficiency
(e) total work input coefficient

Equations for these corrections are listed in Figure 5-6.3.2-1.

This Code uses the machine Reynolds number. In general, if the test machine Reynolds number is less than the specified machine Reynolds number, the flow coefficient, polytropic work coefficient, and polytropic efficiency are increased by these corrections and the work input and total work input coefficients are decreased by the correction factors. The correction method used is based on Strub et al. (1987) and Nonmandatory Appendix F. The correction method uses an analogy to pipe friction factors that are impacted by flow path surface relative roughness and Reynolds number. Three values of friction factors are derived from equations that represent a Moody diagram for pipe friction. Solution of the equations shown in Figure 5-6.3.2-1 requires an iterative direct substitution method to determine two of the correction factors. Table C-3.10.1-1 summarizes the results for the high-pressure natural gas compressor sample case. The corrected values represent expected coefficient values at the site rated conditions. Note that the correction factors for work input and total work input coefficients are equal.

C-3.10.2 Perfect Test Trends for Reynolds Number Corrections. Figures C-3.10.2-1 through C-3.10.2-4 illustrate "perfect test" trends for machine Reynolds number correction factors. The variables investigated were relative roughness, the ratio of test machine Reynolds number to specified machine Reynolds number, and impeller tip width. The trends shown and discussed here pertain to the sample high-pressure natural gas compressor used in this Appendix and may not represent trends found in other machines.

Data for the figures was generated assuming a so-called perfect test in which the specified and as-tested nondimensional performance coefficients were set equal except for the machine Reynolds numbers (*Rem*). This removes any test design and/or testing errors so that Reynolds number correction trends can be independently investigated. The specified *Rem* was kept constant while the test *Rem* was varied to provide a machine Reynolds number ratio covering allowable range limits. Three values of relative roughness and two values of impeller flow passage tip width were used.

For the coefficients in Figures C-3.10.2-1 through C-3.10.2-3, as flow passage surface roughness is increased, the effect of Reynolds number corrections decreases at constant impeller tip width. As impeller tip width is increased, the effect of Reynolds number corrections increases at constant roughness. In other words, for this pipe friction analogy model, the need for Reynolds number corrections appears to be more prevalent in larger compressors with smoother flow passages when tested at low machine Reynolds numbers.

Table C-3.10.1-1
As-Tested Reynolds Number Corrections

Given Data		At Infinite Rem	At Specified Rem	At As Tested Rem	Correction Factor	Corrected Value
Impeller Tip Width (b2)	inches		0.34	0.34		
Roughness (Ra)	inches		0.000125	0.000125		
Relative Roughness (Ra/b2)			0.0003676	0.0003676		
Friction Factor	λ	0.015597389	0.015646998	0.015962388		
Flow Coefficient			0.0103	0.0106	1.0024	0.0106
Work Input Coefficient			0.8225	0.8421	0.9952	0.8380
Polytropic Work Coefficient			0.4880	0.4949	1.0049	0.4973
Polytropic Efficiency			0.5930	0.5874	1.0098	0.5931
Total Work Input Coefficient			0.8231	0.8586	0.9952	0.8544
Machine Reynolds Number			1.38E+07	1.80E+06		

Figure C-3.10.2-1
Reynolds Number Correction Trend — Polytropic Efficiency — Perfect Test

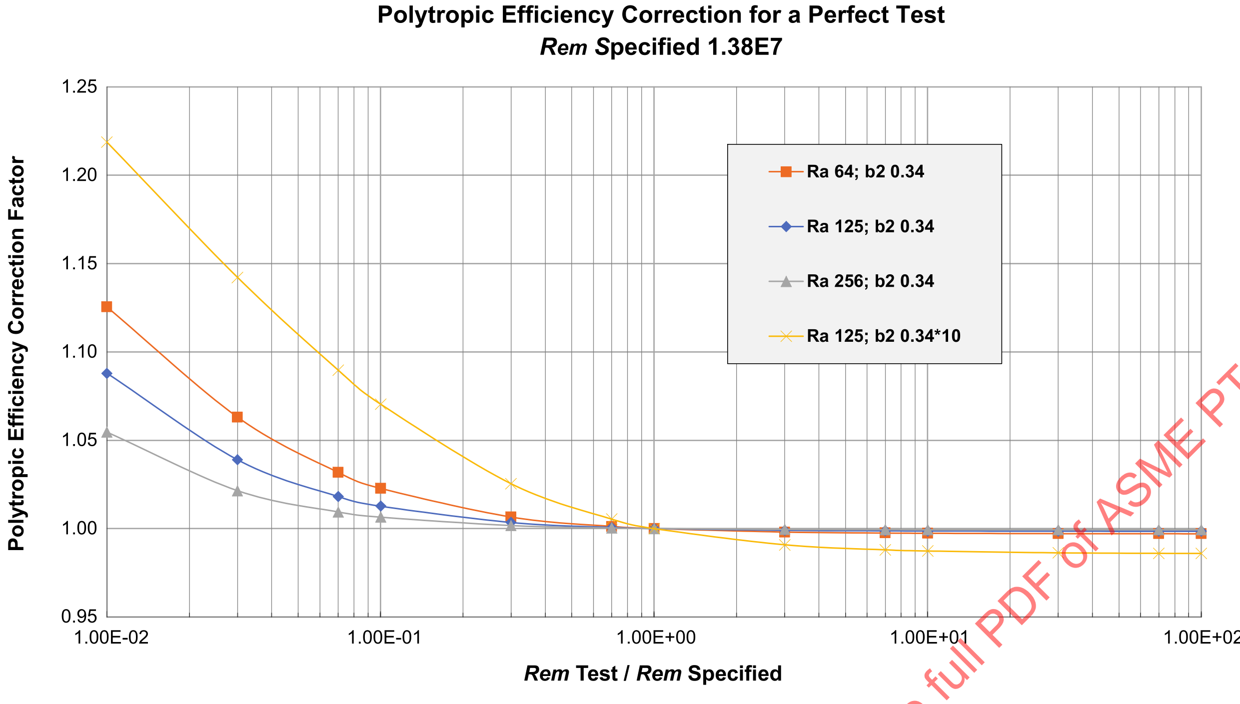


Figure C-3.10.2-2
Reynolds Number Correction Trend — Polytropic Work Coefficient — Perfect Test

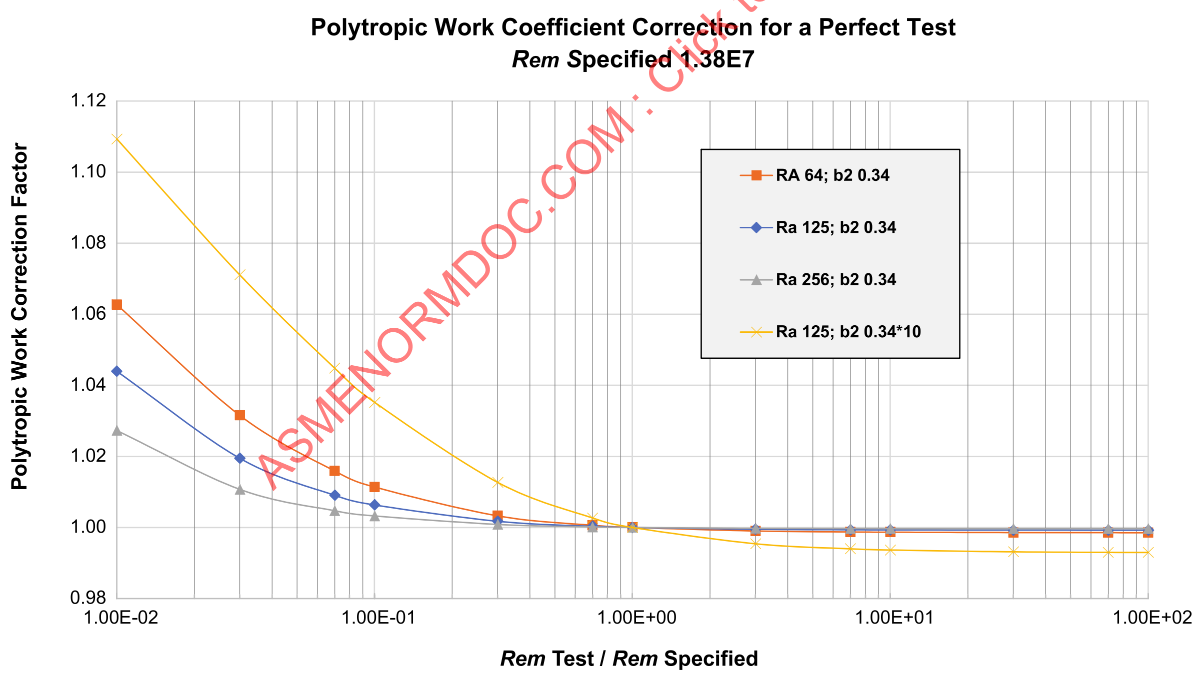


Figure C-3.10.2-3
Reynolds Number Correction Trend — Flow Coefficient — Perfect Test

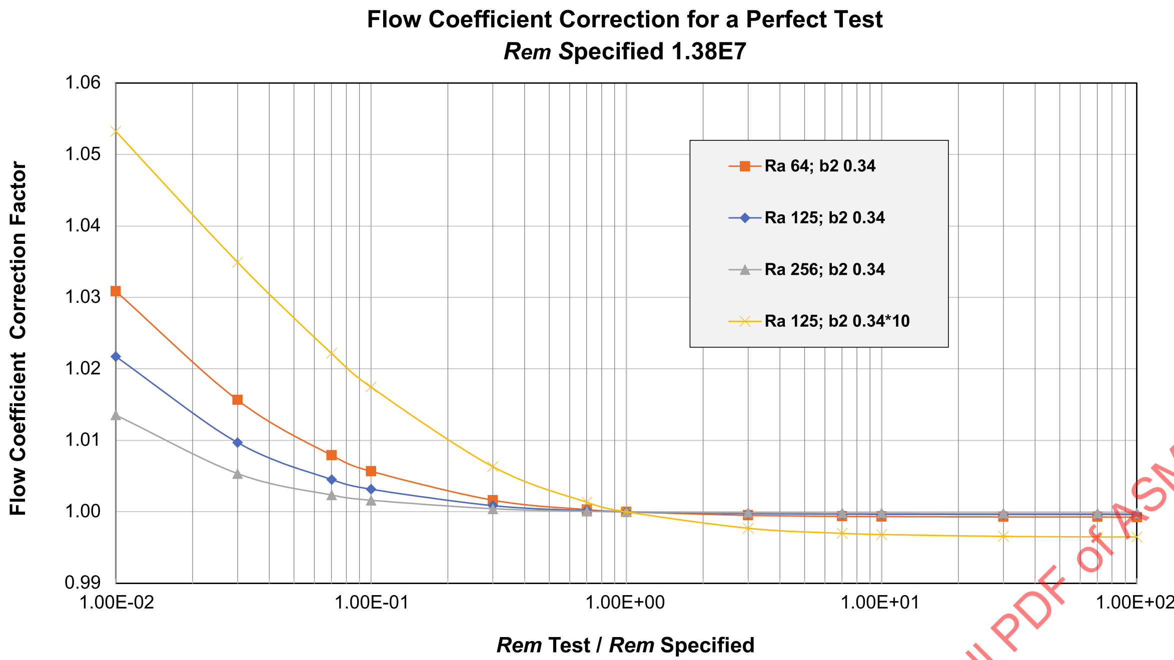
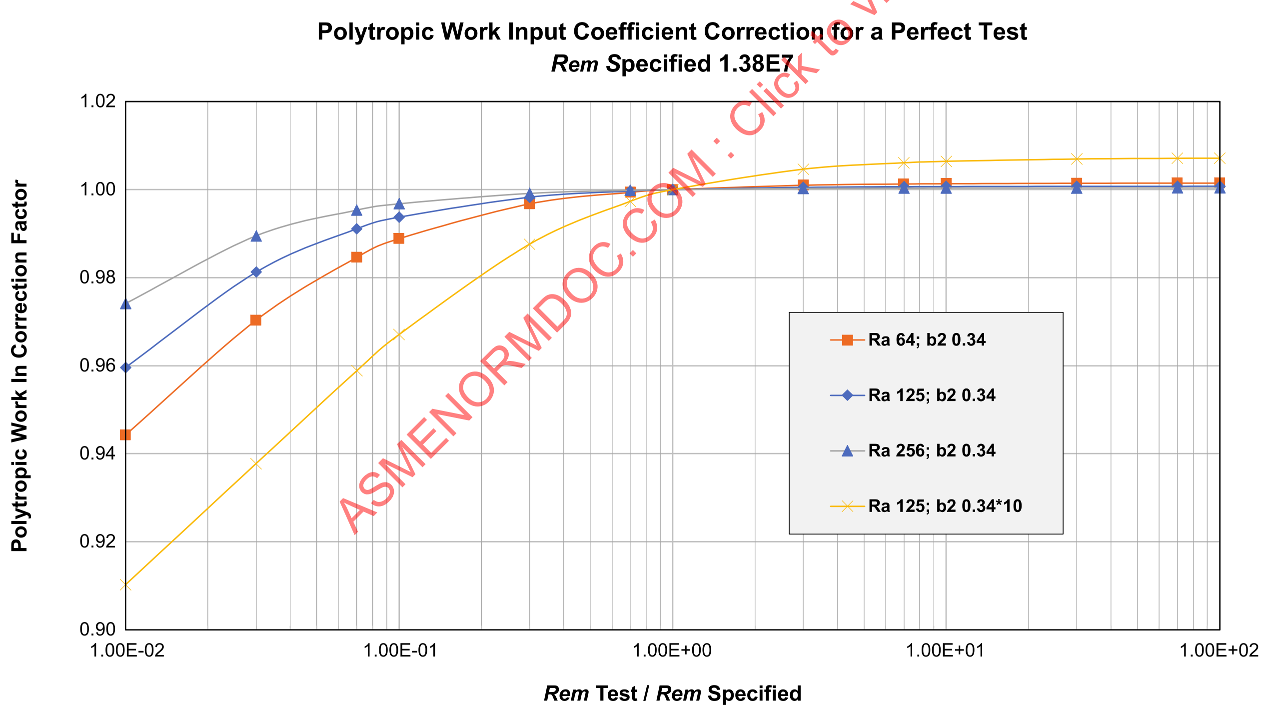


Figure C-3.10.2-4
Reynolds Number Correction Trend — Work Input and Total Work Input Coefficients — Perfect Test



C-3.11 Conversion of As-Tested Performance to Specified Conditions

The corrected as-tested nondimensional coefficients from Table C-3.10.1-1 are used to determine the converted specified performance according to Table 5-6.1.2-2. The lower section of Table C-3.11-1 shows the results based on the inputs listed in the upper section of the table. The values shown represent the final as-built performance when installed at the site in terms of total pressures and temperatures.

Determination of discharge values for total entropy, total pressure, and total temperature must be determined by iterative techniques. Mechanical and heat transfer losses were assumed to be equal to the design values for this sample calculation but could be recalculated based on mechanical testing or vendor or user experience.

In this sample case, the delivered mass flow rate is equal to the rotor mass flow due to the initial decision to ignore leakage flows. In actual practice, the leakage flows should be included. If leakage had been considered, it would also impact the values for gas power and shaft power.

These results represent specified site inlet conditions without any overarching system controller activity. Had a second measured test data point been recorded at a lower "bracketing" capacity, performance results at the specified capacity could be interpolated according to para. 5-6.1.2, Step 2.

For specified inlet conditions and rotor speed, the converted specified mass flow rate is approximately 3% high. For given design input parameters, design discharge parameter deviations can result from design and manufacturing inaccuracies. For variable-speed or variable guide vane compressors, in practice, this deviation could trigger a controller activity on a selected target variable to operate at the exact value of the desired parameter such as discharge pressure or mass flow. For any guarantee considerations, which are beyond the scope of this Code, the controller activity could be taken into account.

Table C-3.11-2 illustrates a comparison of the specified design performance with the converted specified performance. Gas composition, inlet pressure, inlet temperature, and compressor speed are held identical. Discharge conditions and mass flow are based on the as-tested converted results. The compressor is 2.77% high in flow, 2.23% high in total pressure rise, 0.46% high in total temperature rise, and 6.61% high in shaft power. This illustrates that even though the Type 2 test was shown to accurately represent the compressor's capabilities in Table C-3.9-1, the resulting site operation may not meet all of the desired design parameters. System controller adjustments mentioned earlier in para. C-3.11 could be used to change variables such as speed or inlet guide vane position to mitigate the impact of the flow and discharge pressure deviation values.

If the OEM had run an internal pretest prior to a scheduled customer witness test, the speed and flow discrepancies could have been discovered. It would then have been possible to adjust the test design speed and predicted test performance prior to a witness test, which could result in a better match.

Comparison of polytropic efficiency values shows a difference of −0.06% between specified and converted specified performance values. Without the Reynolds number correction discussed in para. C-3.10, the difference would be −0.63%.