ANSI/CAGI BL 300-2022

American National Standard

Performance Test Code for Electric Driven Low Pressure Air Compressor Packages

Compressed Air and Gas Institute

Sponsor:

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Compressed Air & Gas Institute

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Compressed Air and Gas Institute (CAGI)

Foreword (This foreword is included for information only and is not part of ANSI/CAGI/BL 300 - Performance Test Code for Electric Driven Low Pressure Air Compressor Packages).

This document was developed by the Compressed Air & Gas Institute Blower Section. It allows for the comparability for all kinds of low-pressure compressors (blowers) as defined by existing ISO 1217 and ISO 5389 standards.

The standard provides a uniform method of testing all types of low-pressure compressor packages. It also allows for side-by-side in-field performance comparisons. Readers are encouraged to review Appendix E which provides examples of standardized performance data sheets to help facilitate comparison of different type of low-pressure compressor packages and performance guarantee tolerances.

The Compressed Air & Gas Institute recognizes the need to periodically review and update this standard. Suggestions for improvement should be forwarded to the Compressed Air & Gas Institute, 1300 Sumner Ave., Cleveland, OH 44115; E-mail address: cagi@cagi.org.

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Performance Test Code for Electric Driven Low Pressure Air Compressor Packages

1 Introduction

This document was developed in response to a recognized need to provide a simplified, wire to air performance test methodology to:

a) Measure true package performance of low pressure air compressors (blowers)
b) Correct as tested performance to reference or contract specified conditions.

This document addresses the need for both positive displacement and dynamic compression types and can be used for verification of serial products performance data sheets or customer specified operating conditions.

Dynamic compression involves gas drawn between the blades of a rapidly rotating impeller and accelerates to a high velocity. The velocity energy is converted to pressure energy via a diffuser and into a volute or collector. Dynamic compressors (blowers) are of a radial flow design, with the following typical examples: single-stage centrifugal compressors, multi-stage centrifugal compressors without intercooling, and high speed "turbo" compressors.

Positive displacement low pressure compressors (blowers) work on the principle of trapping a volume of air and reducing its volume, internally or externally. Two basic types are typical, as follows:

- Rotary Screw Positive Displacement Compressor (Blower): Air is drawn into a compression chamber formed by intermeshing rotors. As the rotors turn, the cavity between the rotors becomes smaller, reducing the volume of the trapped air. When the pressure has reached the designed built in pressure ratio, the rotors uncover the outlet port and the air is discharged into the customer's piping.
- Rotary Lobe Positive Displacement Compressor (Blower): Air is drawn into the case and is trapped between the rotor and the case wall. These pockets are progressively moved to the outlet port. At the outlet, some air from the piping comes back into the compressor, compressing the air.

For positive displacement compressors (blowers) without internal compression (isochoric system such as the Rotary Lobe Roots- type), it is possible to describe their behavior with the necessary accuracy; however, for positive displacement compressors with internal compression (e.g. screw type compressors/ blowers), the situation is more complex. For these types of machines, part of the compression will be internal (comparable with isentropic processes) and part will be external (isochoric process). These processes must be addressed, under both test and guarantee conditions. This document includes the method that takes both kinds of processes under consideration and weights them according to the given conditions. This method is valid for single stage positive displacement compressors (blowers) with a known value of the volume ratio (defining the degree of internal compression) and without any liquid injection into the process air stream.

2 Scope

This document specifies the performance test method of electrically driven low-pressure compressor packages, where the compression is performed by positive displacement or dynamic compression. Low-pressure air compressor packages are often referred to as "blowers".

"Low Pressure" is defined in section 4.2 further in the text.

Low-pressure compressors with and without means of controlling flow are covered. The means of controlling flow may be electrical (e.g. with a variable frequency drive) or mechanical or both.

This document is not applicable to:

- Low-pressure positive displacement compressors with a liquid in the compression element (such as liquid ring pumps and liquid injected low-pressure compressor of screw type).
- Multi-stage low-pressure compressors with intercooling between stages of compression

This document applies to low-pressure compressors meeting all the limits defined in section 4.2.

3 References

The following referenced documents are relevant for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

- ASME PTC 10-1997, Performance Test Code on Compressors and Exhausters
    - Equivalent ISO 5389:2005, Turbo compressors – Performance test code
- ASME PTC 9-1970, Displacement Compressors, Vacuum Pumps and Blowers
- ISO 1217:2009, Displacement compressors – Acceptance tests
- EN 60051:1999, Accuracy classes for measuring instruments
- EN 60688:2002, Technical basics for measurements
- ASME MFC-3M-2004, Measurement of Fluid Flow in Pipes Using Orifice, Nozzle, and Venturi
    - Equivalent ISO 5167-1:2003, Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full – Part 1: General principles and requirements
- ISO 80000:2009 Quantities and units – Part 1: General
- ASME MFC-7M:2006, Measurement of Gas Flow by Means of Critical Flow Venturi Nozzles
    - Equivalent ISO 9300:2005, Measurement of gas flow by means of critical flow Venturi nozzles

In addition, all ISO norms feasible for this work can be used.

4 Definitions

4.1 Symbols and units

The following symbols are to be used unless otherwise defined in the text. These might slightly differ from existing standards like ISO 5389, ISO 1217 or ISO 80000.

		Units	
Latin letters Symbol	**Meaning**	**US Customary**	**SI**
c	sonic velocity	ft/sec	m/s
cp , cv	specific heat capacity	Btu/lb·ºR	J/(kg·ºK)
D	Outer impeller diameter of the first impeller	ft	m
e	specific energy	kW/(ft^3/min)	kW/(m^3/sec)
h	specific enthalpy	Btu/lb	J/kg
Ma	Mach number	—	—
M	molar mass	lbm/lbmole	kg/kgmole
m	Mass	lb	kg
q_m	mass flow	lb/s	kg/s
n	speed of rotation	1/s	1/s
P	power	kW	kW
p	pressure	lbf/in^2	bar
R	specific gas constant	ft·lbf/lbmole·ºR	J/(kg·ºK)
Rmol	molar gas constant	ft·lbf/(mol·ºR)	J/(mol·ºK)
Re	Reynolds number	—	—
s	specific entropy	Btu/(lb·ºR)	J/(kg·ºK)
T	thermodynamic temperature	ºR	ºK
t	temperature	°F	°C
u	tip speed	ft/s	m/s
v	specific volume	ft^3/lb	m^3/kg
v_i	Internal volume ratio		
V	Volume	ft^3	m^3
q_v	Volume flow	ft^3/min	m^3/min
X_n	ratio of reduced speeds of rotation	—	—
x	mass ratio of water vapour to dry gas	lb/lb	kg/kg
y	specific compression work	ft·lbf/lbm	J/kg
Δ	difference	—	—
η	efficiency	—	—
ϑ	ratio of (RZ1 T1) values	—	—
κ	ratio of specific heat capacities	—	—
π	pressure ratio	—	—

ρ	density	lb/ft^3	kg/m^3
ϕ	ratio of volume flow ratios	—	—
φ	flow coefficient	—	—
φ_{rel}	relative humidity		
ψ	reference process work		
σ	standard deviation		

Subscripts

Index	Meaning
1	inlet (suction side)
2	outlet (discharge side)
air	dry air
amb	ambient (air, temperature)
co	converted to guarantee conditions
comb	combined
cool	coolant
d	dynamic
driver	compressor driver
dry	dry
g	guarantee or reference conditions
i	Internal or intermediate
isoc	isochoric
ideal	according to an ideal thermodynamic process
out	output
pack	compressor package
Pr	reference or standard process
red	reduced speed
ref	reference value
rel	relative
s	isentropic
sat	saturated
st	static
te	test result
test 1	first test in 2-speed testing
test 2	second test in 2-speed testing
tol	permissible deviation
tot	total
u	tip or peripheral
vap	vapour, steam
wet	moist

4.2 Low pressure

The test code applies to compressors by the following limits:

$$7 \text{ psia} \leq p_1 \leq 16 \text{ psia}$$
$$0.483 \text{ bar} \leq p_1 \leq 1.103 \text{ bar}$$

and

$$1.5 \text{ psi} \leq (p_2 - p_1) \leq 30 \text{ psi}$$
$$.103 \text{ bar} \leq (p_2 - p_1) \leq 2.068 \text{ bar}$$

and

$$1.1 \leq \left({}^{p_2}/_{p_1}\right) \leq 3.5$$

4.3 Package

The package shall comprise all components that are necessary for the long-term functioning of the compressor under guarantee conditions and are needed to fulfil the object of the guarantee and the preconditions of the guarantee:

- Low-pressure compressor with drive system,
- variable frequency drive (as applicable),
- cooling / lubrication system,
- inlet filter,
- inlet valve / guide vanes (as applicable),
- bearing power supply (as applicable),
- fully piped and wired internally,
- ancillary and auxiliary items of equipment and all power devices that affect power consumption (as applicable).

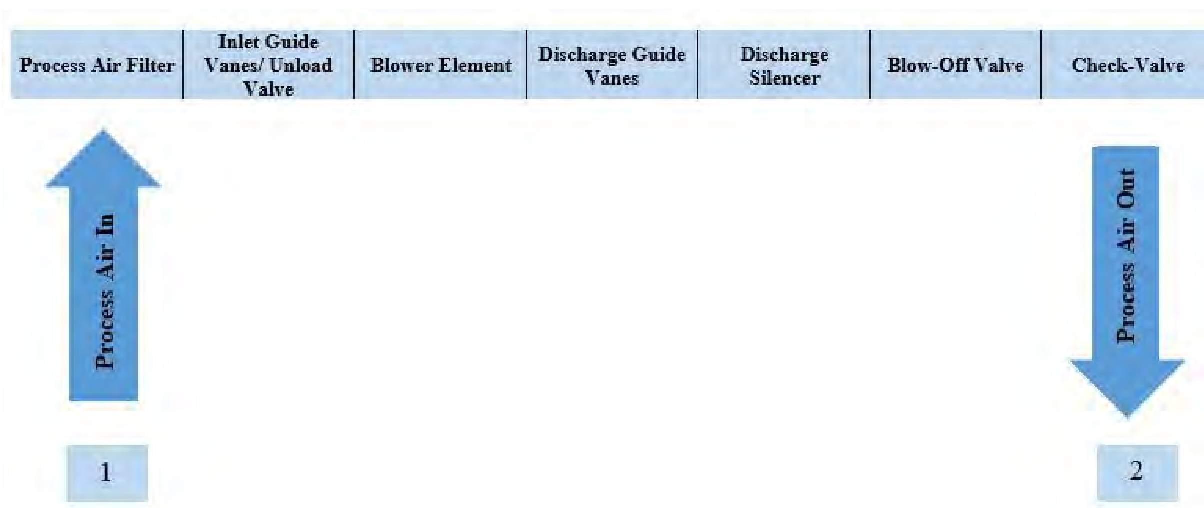


Figure 1: Components and Ancillaries

4.4 Definitions regarding performance

4.4.1 Relative humidity

The relative humidity can be expressed as follows:

$$\varphi_{rel} = \frac{p_{vap}}{p_{vap,sat}}$$

4.4.2 Water vapor content

The water vapour content related to a dry mass of air is:

$$x = 0.622 \cdot \frac{\varphi_{rel} \cdot p_{vap,sat}}{p - \varphi_{rel} \cdot p_{vap,sat}}$$

4.4.3 Isentropic efficiency

Isentropic efficiency is a ratio that indicates how the real energy consumption of an air compressor compares to that for an idealized compression process. The value is expressed as a percentage, the higher the number indicates that the machine is more efficient at converting electrical energy into compressed air potential energy. Isentropic compression is considered where the system is frictionless, and there is no transfer of heat. Such idealized processes are useful as a model and basis for real compression processes. Isentropic efficiency can be calculated and measured independently of compression technology. Efficiency is not measured directly but is derived from power consumption, pressure ratio and delivered capacity.

4.4.4 Isentropic exponent

For dry air close to 14.7 psia pressure, the approximation for the isentropic exponent is:

$$\kappa_{dry} \approx 1.4$$

The isentropic exponent κ for humid air is then determined as follows:

$$\kappa_{wet} = \kappa_{dry} \cdot (1 - 0.11 \cdot x)$$

4.4.5 Gas constant

Determining the gas constant R of humid air can be done as follows:

$$R_{wet} = R_{air} \cdot \left(\frac{1}{1 - \frac{\varphi_{rel} \cdot p_{vap,sat}}{p} \cdot 0.378} \right)$$

or

$$R_{wet} = R_{air} \cdot \left(1 + \frac{x}{x+1} \cdot 0.608\right)$$

4.4.6 Reference process compression work

4.4.6.1 Specific isentropic compression work for dynamic machine

Within limits (table E1, ISO5389) the specific isentropic compression work is defined as:

$$y_s = \frac{\kappa}{\kappa-1} \cdot R_1 \cdot T_1 \cdot \left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{\kappa-1}{\kappa}\right)} - 1\right]$$

4.4.6.2 Specific isochoric compression work

The specific isochoric compression work is defined as

$$y_{isoc} = \frac{1}{\rho_1} \cdot [p_2 - p_1]$$

4.4.6.3 Specific combined isentropic and isochoric compression work for positive displacement machine

The specific combined isentropic and isochoric compression work is defined as

$$y_{comb} = R_1 \cdot T_1 \cdot \left[\frac{{p_2}/{p_1}}{v_i} + \frac{\kappa}{\kappa-1}\left(\frac{1}{\kappa} \cdot {v_i}^{\kappa-1} - 1\right)\right]$$

If the internal volume ratio v_i is equal to 1 then the specific combined isentropic and isochoric compression work is equal to the specific isochoric compression work.

If the internal volume ratio v_i is equal to $\left({p_2}/{p_1}\right)^{1/\kappa}$ then the specific combined isentropic and isochoric compression work is equal to the specific isentropic compression work.

4.4.7 Internal volume ratio

The internal volume ratio v_i of a volumetric compressor is defined as the ratio of the enclosed volume at moment of closure of the inlet port to the enclosed volume at the moment of opening of the outlet port.

4.4.8 Inlet volume flow rate

The inlet volume flow rate considers the gas condition at the process air inlet as defined in Figure 1.

The inlet volume flow rate is defined as the delivered mass flow rate q_{m2} divided by the total density at the compressor package inlet. The delivered mass flow rate shall be measured downstream of the process air discharge in order to exclude all leakage losses.

$$q_{v1} = \frac{q_{m2}}{\rho_1}$$

4.4.9 Specific energy

The specific energy is the absolute work required to compress a volume of gas from the pressure (and temperature) at package inlet, to the package discharge pressure, while accounting for changes in the enthalpy and kinetic energy of the gas during the compression process, including all possible mechanical and electrical losses.

$$e = \frac{P}{q_{v1}}$$

4.4.10 Rotor tip speed

The tip speed results from the rotor outer diameter and speed of rotation.

$$u = \pi \cdot D \cdot n$$

4.4.11 Flow coefficient

The flow coefficient is a flow velocity formed from the inlet volume flow and an impeller cross-section area, rendered dimensionless by the tip speed of the rotor.

$$\varphi = \frac{q_{v1}/60}{\frac{\pi}{4} \cdot D^2 \cdot u}$$

4.4.12 Work coefficient

The work coefficient of the reference process specific work (for the entire package) is rendered dimensionless by the kinetic energy of tip speed u.

$$\Psi = \frac{y_s \cdot g}{u^2}$$

Where G is gravitational constant of 32.17 ft/s^2

4.4.13 Machine Mach number

Calculation of the machine Mach number can be done as follows:

$$Ma = \frac{u}{c_1} = \frac{u}{\sqrt{g.\kappa_1 \cdot R \cdot T_1}}$$

Where g is the gravitational constant of 32.17 ft/sec^2

4.4.14 Process air inlet point definition (subscript 1)

The process air inlet point (index "$_1$") is defined as being upstream of any technically required component.

In the case in which a technically required component is not physically present during the test the impact of the component on performance shall be accounted for.

4.4.15 Process air outlet point definition (subscript 2)

The process air discharge point (index "$_2$") is defined as being downstream of any technically required component.

In the case in which a technically required component is not physically present during the test the impact of the component on performance shall be accounted for.

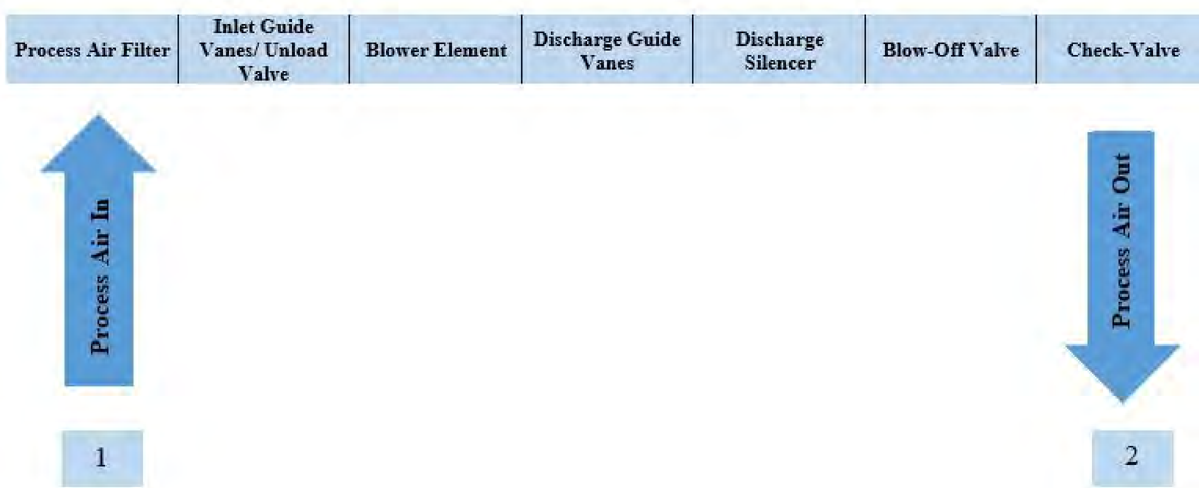


Figure 2: Process Air Inlet & Outlet

4.4.16 Inlet conditions

Inlet conditions for the guarantee are:

Table 1: Inlet Conditions

Default inlet condition	Value (Option 1)	Value (Option 2)
Inlet air pressure	14.7 psia	1 bar

Inlet air temperature	68ºF	20°C
Inlet relative humidity	36%	0%
Temperature of the coolants at package inlet	68ºF	20°C

4.4.17 Client specified inlet conditions

The client may specify the site conditions (other than the default inlet conditions) to which the performance test data is to be corrected to.

4.4.18 Definition: Steady state

Steady state is achieved when the difference between inlet and outlet temperature is within 2ºF (1.11ºC) and the speed variation is held to within 0.5% for at least three minutes.

4.4.19 Free air delivery (FAD)

Actual volume flow rate of air, compressed and delivered at the package discharge point, referred to conditions of total temperature, total pressure, and composition (e.g., humidity) prevailing at the ambient inlet of the compressor.

5 Guarantee and Measurement

5.1 Preconditions of the guarantee

Preconditions are the conditions the compressor (blower) will be externally exposed to in use and which shall be specified in the contract of supply (meaning "contract", "data sheet", "agreement" or similar) or default preconditions shall be applied according to this guideline (or some other applicable instruction).

For testing to be possible at least the following shall be specified:

Table 2: Preconditions of the Guarantee

Preconditions of the guarantee
Air inlet pressure*
Air inlet temperature*
Air inlet humidity*
Coolant inlet temperature*
Coolant flow
Supply voltage
Supply frequency

*These can be taken from the default conditions defined in Table 1 above.

5.2 Object of the guarantee

The object of guarantee is the set of values to be guaranteed within the defined preconditions:

1. Inlet volume flow rate
2. The discharge pressure at the outlet of the package.
3. The total Specific Energy of the package for the delivered flow at the guaranteed discharge pressure.

5.3 Low Pressure Compressor (Blower) to be tested

The compressor (blower) configuration to be tested shall include all components required to fulfill all the preconditions.

As a general rule, the configuration of the unit under test shall be identical to the configuration of the unit to be delivered.

A package checklist, such as given in Appendix A, shall be completed by the manufacturer and shall be part of each compressor test report. The checklist shall be used to ensure that the tested package matches the specified one.

The checklist shall indicate which components are included, excluded, or not applicable for normal functioning at guarantee conditions or accounted for. If any required components are not installed in the test configuration, the correction calculations for these components shall be shown in conjunction with the checklist.

Ancillaries required for the sustainable operation of the low-pressure compressor package, excluding stand-by ancillaries, are to be in operation.

5.4 Compressor (blower) specifications to be provided prior to testing

The compressor (blower) is tested against a specified outlet pressure (at the outlet of the package).

In addition to the preconditions (section 5.1) and the package check-list (Appendix A), the description of the compressor (blower) to be tested shall contain specific data for the performance calculation. This includes:

- The compressor (blower) rotational speed.
- The internal volume ratio for positive displacement compressors (blower).
- The variable geometry settings if applicable for the compressor (blower).

6 Measuring equipment, methods and accuracy

6.1 General

The equipment and methods given in this document are not intended to restrict the use of other equipment and methods with the same or better accuracy. Where an International Standard relating to a particular measurement or type of instrument exists, any measurements carried out or instruments used shall be in accordance with such an International Standard.

All inspection, measuring, test equipment and devices that can affect the test shall be calibrated and adjusted at prescribed intervals, or prior to use, against certified equipment having a known valid relationship to nationally recognized standards. The use of data acquisition systems shall be allowed, and test logs may be print outs resulting from the system.

No measurement uncertainty tolerances are to be taken into account in corrections or comparisons or acceptance. For guarantee acceptance, as tested results are treated as measured to comparison to Table 4 without additional uncertainty tolerances applied.

Figure 3 below is inserted on a typical measurement installation.

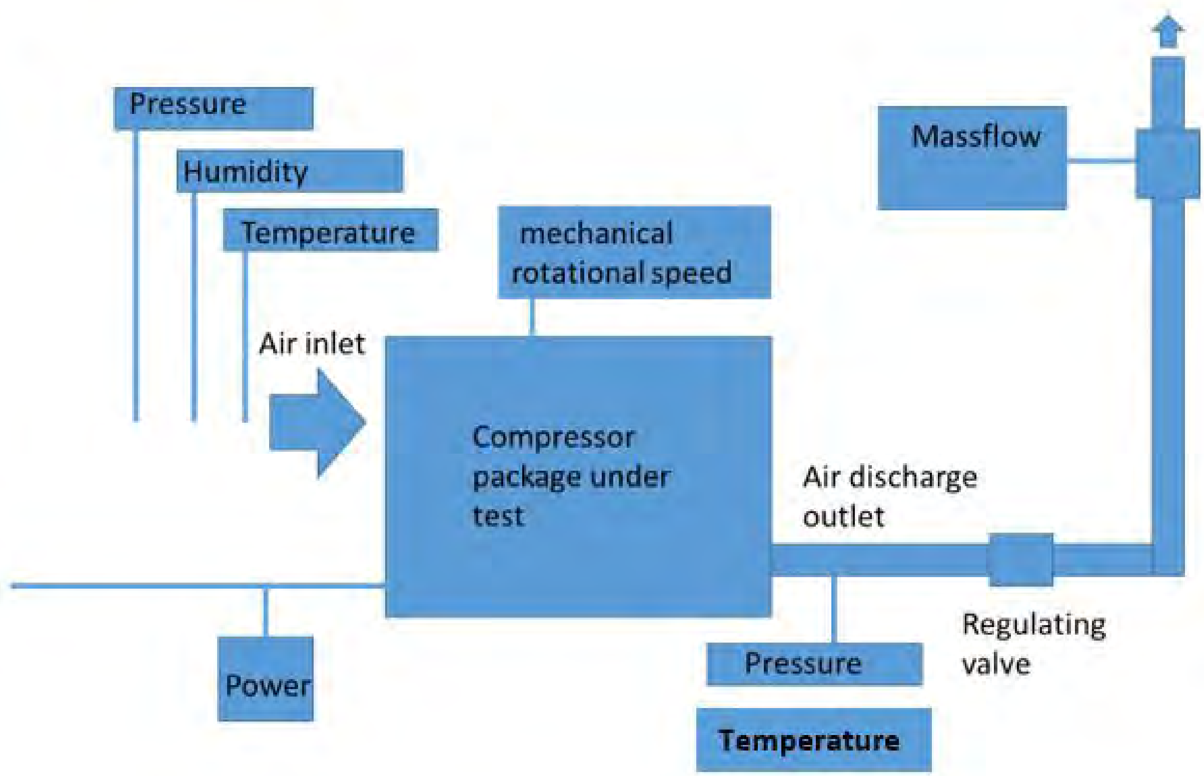


Figure 3: Overview of Typical Measurement Installation

6.2 Measurement of pressure

6.2.1 General

Pressure taps in the pipe or receiver shall be normal to, and flush with, the inside wall. A minimum of two static or total pressure-measuring instruments shall be utilized for each measurement location spaced at 180° intervals around the pipe circumference, and 90° to temperature instrumentation.

NOTE For low pressures or high flow velocities, minor irregularities such as burrs can lead to serious error.

Connecting piping shall be leak-free, as short as possible, of sufficient diameter and arranged to avoid blockage by dirt or condensed liquid. For measurement of liquid pressure or pressure of liquid-gas mixtures, the instrument shall be mounted at the same height as the measuring point and the connecting piping shall be arranged so that the height of liquid columns in the piping exerts no influence. Otherwise, account shall be taken of the difference in height.

Instruments shall be mounted so that they are not susceptible to disturbing vibrations.

The measuring instrument (analogue or digital) shall have an accuracy of ±1 % at the measure value.

The pressure measurement shall be a total measurement, or static measurement corrected to total conditions.

For definitions of static, dynamic and total measurements refer to ISO 5389:2005 Section 5.2. and 5.3.

6.2.2 Atmospheric pressure

The absolute atmospheric pressure shall be measured with a barometer having an accuracy better than ± 0.15%.

6.2.3 Pressure measurement for ambient inlet

The compressor package inlet pressure (p_1) is the atmospheric pressure measured by a barometer near the compressor package where the velocity is zero.

6.2.4 Pressure measurement for piped inlet

The pressure is the total pressure (p_1) measured at the process air inlet point. The pressure shall be measured at a location at least one pipe diameter upstream of the inlet.

6.2.5 Pressure measurement for piped outlet.

The pressure is the total pressure (p_2) measured at the process air outlet point. The pressure shall be measured at a location at least two pipe diameters downstream of the outlet.

6.3 Measurement of temperature

6.3.1 General

Temperature shall be measured by certified or calibrated instruments such as thermometers, thermos-electrical instruments, resistance thermometers or thermistors having an accuracy of ± 1°F inserted into the pipe or into thermowells.

A minimum of two temperature measuring instruments shall be used for each measurement location. For measurements made on piping these shall be spaced at 180° intervals around the pipe circumference.

Thermometer wells shall be as thin, and their diameters as small, as is practical, with their outside surface substantially free from corrosion or oxide. The well shall be partially filled with a suitable liquid.

The thermometers or the wells shall extend into the pipe to a distance of 4 inches or one third the diameter of the pipe, whichever is less.

When taking readings, the thermometer shall not be lifted out of the medium measured nor out of the thermowell when one is used.

Precautions shall be taken to ensure that the:

- Immediate vicinity of the insertion point and the projecting parts of the connection are well insulated so that the thermowell is virtually at the same temperature as the medium being observed;
- Sensor of any temperature measuring device or thermowell is well swept by the medium (the sensor or thermowell shall point against the gas stream; in extreme cases a position perpendicular to the gas stream may be used);
- Thermowell does not disturb the normal flow.

6.3.2 Temperature measurement – Ambient

The package ambient temperature is the atmospheric temperature measured at the package in the plane of the intake system.

6.3.3 Temperature measurement for piped inlet

The inlet temperature is the total temperature (T_1) measured at the process air inlet point. The temperature instrumentation shall be located at half of one pipe diameter upstream of the inlet.

6.3.4 Temperature measurement for piped outlet

The outlet temperature is the total temperature (T_2) measured at the process air outlet point. The temperature instrumentation shall be located one pipe diameter downstream of the outlet and 90° relatively rotated to the pressure measurement.

6.4 Measurement of humidity

If the gas contains moisture, the humidity shall be checked during the test. The humidity shall be measured at the standard inlet point with an instrument have an accuracy of ±2.5% or better.

6.5 Measurement of rotational frequency

Shaft speed shall be determined by using methods that have an accuracy of ± 0.2% or better.

6.6 Measurement of flow rate

The delivered flow rate is the net mass flow rate through the process connection at the package outlet. All seal losses and side streams not delivered to the process piping connection of the package shall be excluded from the delivered mass flow rate evaluation.

Measuring devices mentioned in ISO 5167 shall be deemed acceptable.

Overall uncertainty of measured value shall be ± 1.5% or better.

6.7 Measurement of power and energy

6.7.1 General

Electric power of the machine shall be referred to the electrical input terminals. Factors influencing the measurement, such as voltage drop in supply cables or measurement systems, shall be taken into account.

The two-wattmeter method or some other method with similar accuracy shall be used.

Current and voltage transformers shall be chosen to operate as near to their rated load as possible so that their ratio error will be minimized.

Electrical measurement equipment must be capable of measuring true root mean square (RMS) current, true RMS voltage, and real power up to the 40^{th} harmonic of fundamental supply source frequency.

Overall uncertainty of the measured value shall be ± 0.75% or better.

The power supply must:

- Maintain the voltage greater than or equal to 95% and less than or equal to 110% of the rated value of the motor,
- Maintain the voltage unbalance of the power supply within ±3% of rated values of the motor.

For inverter applications, the wire power-measuring instrument shall be capable of handling the distorted voltage and current waveforms and phase relationship of the power factor caused by the harmonics and EMI as a result of inverters high-speed switching mode. Wire power shall be measured by a precision power analyzer with high accuracy, broad bandwidth, fast sampling rate and high-speed data update.

6.8 Calibration of instruments

Calibration records of the instruments shall be available prior to the test.

Recalibration after the test shall be carried out for those instruments of primary importance that are liable to variation in their calibration because of use during the test.

Any change in the instrument calibrations, which will create a variation exceeding the class of accuracy of the instruments, may be cause for rejecting the test.

7 Test

7.1 General test process

In order to compare the performance data of low-pressure compressors (blowers) with different technologies, it is necessary to test these compressors (blowers) under the same similarity conditions and with the same methods by applying the same principles and process steps. The ideal reference process in this methodology is different for different types of compressors. Positive displacement compressors (blowers) with or without internal compression use the combined isentropic and isochoric process. Dynamic compressors (blowers) of any kind use the isentropic process.

The test shall be carried out at an appropriate test field under prevailing conditions. No changes to geometry are allowed between test and specified conditions. For the set-up, the compressor is connected to the test loop. Correct cooling conditions have to be assured. The compressor shall run for warm-up against rated outlet pressure until steady state conditions are reached and the temperature remains constant at the inlet and outlet of the flow measuring device. The compressor (blower) package shall operate at the steady state condition for the duration of data collection for each test point.

In Figure 4, the general process of testing is shown in a schematic.

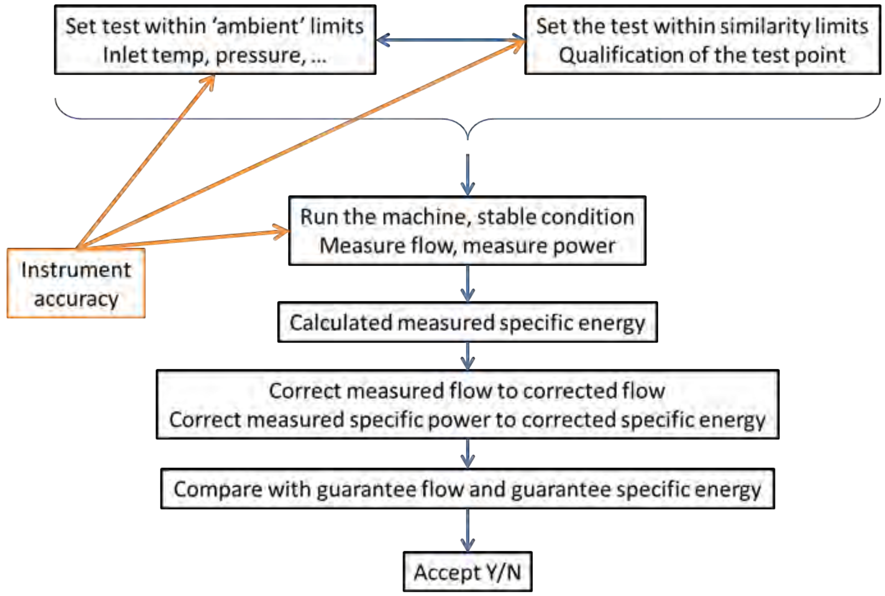


Figure 4: Overview of Test Process

This document describes a method to test the performance of a single operating point. For variable flow compressors (blowers) this method can be repeated at several flow rates to establish the performance over the operating range of the compressor at the specified outlet pressure / speed / mechanical vane setting. For the purpose of general data sheets, or in case there is no specific agreement with the client on how to test variable flow compressors, 5 flow rates shall be tested. The minimum and maximum flow rates that can be achieved continuously under guarantee conditions shall be specified and tested by the manufacturer. Furthermore, three additional flow rates evenly spread within the total flow rate range of the compressor (blowers) shall be specified and tested. For each test point, the adequate test loads shall be determined in the same way as for a single working point.

7.2 Allowed deviation of rotational speed between test and guarantee

The supplier shall inform about the expected rotational speed for the specified point while operating at guarantee conditions prior to conducting the test.

The allowed deviation for rotational speed is ±3%.

For variable speed dynamic compressors (blowers), it is possible that the proper setup of the compressor in order to keep the deviation of the dimensionless numbers within the stated limits, results in a rotational speed that deviates more than ±3% from the expected rotational speed. In cases in which the rotational speed deviates more than ±3% such cases a second test is to be conducted to verify the efficiency at the guarantee speed (see definition of two speeds testing in section 7.10.

7.3 Allowed deviation of ambient conditions

7.3.1 Requirements on test facility

The ambient air density shall not deviate by more than +/- 10% compared to the density at guarantee conditions.

7.3.2 Requirements for data to be published

The ambient conditions shall not deviate compared to the indicated and/or guarantee conditions by more than:

Inlet temperature: ± 27°F
Inlet pressure: ±10%

7.3.3 Customer specified site conditions

There is no limit of the deviations between the test conditions and the site specified conditions.

7.4 Allowed deviation of preconditions

Below are the allowed deviations of the precondition data, if applicable:

- liquid coolant temperature: ± 18°F
- mass flow of liquid coolant: ±10%

7.5 Allowed deviation of compressor Mach number

It is essential to keep the dimensionless numbers as similar as possible between the guarantee conditions and the test conditions.

For dynamic compressors the deviation of compressor Mach number shall be between -5% and +5%.

For positive displacement compressors there is no restriction on the deviation of compressor Mach number.

7.6 Selection of test flow

7.6.1 Selection of flow setting

7.6.1.1 Fixed flow, positive displacement and dynamic compressors

For compressors with no possibility to adjust the flow, the flow will result from the actual speed at which the compressor is running (e.g. constant speed compressors with no flow-adjusting device).

7.6.1.2 Variable flow, positive displacement compressors

The specified volume flow shall be matched within the given tolerance by adjusting the compressor (by speed or by positive displacement per revolution, if adjustable).

7.6.1.3 Variable speed, dynamic compressors

The supplier shall inform about the reference rotational speed and settings of variable geometry for the specified operating point at the guarantee conditions in advance of test.

The speed setting for the test will result from keeping the dimensionless numbers constant as follows:

Rotational speed is determined from the Mach numbers at specified and test conditions keeping Mach number constant.

The flow will result from the rotational speed and the outlet pressure setting as explained below.

7.6.1.4 Fixed speed, variable flow, dynamic compressors

The supplier shall inform about the reference rotational speed and settings of variable geometry for the specified operating point at the guarantee conditions in advance of the test. Rotational speed is determined by the speed of the drive. Test setting for flow will result from the outlet pressure setting as explained below.

7.7 Allowed deviation of flow and work coefficient

7.7.1 Allowed deviations to be checked for test validity (ensuring similarity)

7.7.1.1 For dynamic compressors (blowers)

It is essential to keep the dimensionless numbers as close as possible between the guarantee conditions and the test conditions.

Deviation of work coefficient: - 2%, +2%
Deviation of flow coefficient: -2%, +2%
Deviation of Mach number: -5%, +5%

7.7.1.2 For positive displacement compressors (blowers)

For the compressor (blower) under test it is essential to keep the following numbers as close as possible between the guarantee conditions and the test conditions.

Deviation of work $_{ycomb}$: - 2%, +2%
Deviation of flow coefficient: -2%, +2%

Remark: For typical positive displacement compressors (blowers) with fixed geometry, where the flow is linear to speed, this latter condition will be fulfilled.

7.8 Selection of test pressure

7.8.1 For positive displacement compressors (blowers) with or without internal compression

The outlet pressure for positive displacement compressors (blowers) with internal compression (internal volume ratio $v_i > 1$) or without internal compression (internal volume ratio $v_i = 1$) shall be set such that the specific combined compression work in the prevailing test conditions matches the one in the guarantee conditions.

$$y_{comb,te} = y_{comb,g}$$

The required outlet pressure can then be calculated as follows:

$$p_{2,te} = \left[\frac{y_{comb,g}}{R_{te} \cdot T_{1,te}} - \frac{\kappa_{te}}{\kappa_{te} - 1}\left(\frac{1}{\kappa} \cdot {v_i}^{\kappa-1} - 1\right)\right] \cdot \; v_i \cdot p_{1,te}$$

7.8.2 For dynamic compressors (blowers)

The outlet pressure for dynamic compressors shall be set in a way such that the specific isentropic compression work in the prevailing test conditions matches the one in the guarantee conditions multiplied with the square of the speed ratio, in order to achieve similarity.

$$y_{s,te} = y_{s,g} \cdot \left(\frac{u_{te}}{u_g}\right)^2$$

The required outlet pressure can then be calculated as follows:

$$p_{2,te} = p_{1,te} \cdot \left[1 + \left(\frac{\kappa_{te} - 1}{\kappa_{te}}\right)\left(\frac{y_{s,g} \cdot \left(\frac{u_{te}}{u_g}\right)^2}{\left(R_{te} \cdot T_{1,te}\right)}\right)\right]^{\left(\frac{\kappa_{te}}{\kappa_{te}-1}\right)}$$

7.9 Fluctuations on the specific test readings during test at steady state

Readings are to be taken at steady state which is defined as the state in which the difference between inlet and outlet temperatures is within 2ºF for a period of three minutes or more.

Before readings are taken, the compressor shall be run long enough to ensure that steady-state conditions are reached so that no systematic changes occur in the instrument readings during the test.

For each flow point, enough readings shall be taken to indicate that steady-state conditions have been reached. The number of readings and the intervals shall be chosen to demonstrate repeatability.

For individual readings, the limits on fluctuations in table 3 below apply:

Table 3: Permissible Fluctuations of Test Readings

Measurement (symbol)	Fluctuation
Inlet pressure ($p_{1,te}$)	1 %
Inlet temperature ($T_{1,te}$)	2ºF
Outlet temperature ($T_{2,te}$)	2ºF
Temperature difference between outlet and inlet ($T_{2,te} – T_{1,te}$)	2°F
Outlet pressure absolute ($p_{2,te}$)	0.5 %
Flow (q_{te})	1 %
Speed (rotational speed) (n_{te})	0.5 %
Electrical power (P_{te})	1 %
Supply voltage	2 %
GENERAL NOTES:	
A fluctuation is the percentage of difference or the difference between the minimum and maximum test reading divided by the average of all sets of readings for one test point.	

7.10 Two-speed test

For dynamic low-pressure compressors, the limits imposed on rotational speed (as defined in 7.2) may lead to a situation in which the Work coefficient, Flow coefficient, and Mach number cannot be maintained with the limits defined in 7.7.1.1.

This can happen if the conditions at test differ significantly from the preconditions of the guarantee. In this case a two-speed test can be applied.

The first test shall be conducted at the specified rotational speed, with the outlet pressure set to load the compressor at the corrected package power as calculated for test1$P_{co,g,test1}$.

The limits regarding flow, pressure, and Mach number do not apply.

The tolerance for $P_{co,g,test1}$ is ±2%.

The efficiency of the driver shall be calculated as follows:

$$\eta_{i,driver,te,x} = \frac{y_{i,te,x}}{y_{pack,x}}$$

(x signifies test1 or test2 as applicable)

$y_{i,testx}$ refers to the measured work, which is equal to the change in enthalpy Δh measured by the temperature rise.

$y_{pack,x}$ refers to the measured work, at the packaged boundary, which is equal to the packaged power divided by the mass flow:

$$y_{pack,x} = \frac{P_{pack,x}}{q_{m,x}}$$

Additionally, the internal work is calculated as follows:

$$y_{i,te,x} = \frac{y_{s,te,x}}{\eta_{s,Pr,te,x}} = \left[\frac{k}{k-1} R_1 . (T_2 - T_1)\right]_{tex}$$

The second test is set up and executed according to the limits of test setup deviation (e.g. Mach number), except for the absolute value of the rotational speed (n).

The second test is conducted at reduced / increased speed:

$$X_n = \frac{n_{red,te}}{n_{red.g}} = \frac{Ma_{u.te}}{Ma_{u.g}}$$

Note: E.g. the needed rotational speed may be in excess of compressor capability in which case the test cannot be executed.

The performance is measured, calculated and evaluated (as 7.1) as any other test point.

Additionally, the internal work and driver efficiency are calculated again, as above.

The results of test1 and test2 are combined to calculate the corrected package power.

The corrected value to use for input power as tested is then calculated as follows:

$$P_{co,g,0,1} = P_{co,g,1} * \frac{\eta_{i,driver,te2}}{\eta_{i,driver,te1}}$$

$n_{i,driver,testx}$ refers to the ratio of real work: (internal work / package work)

8 Correction of Test Results

8.1 General

The test results, measured at the test bench (subscript *te*) shall be recalculated to the corrected values (subscript *co*) with the formulas in the sections that follow. The equations take into account the guarantee conditions (subscript *g*) to calculate these corrected values.

Note: In this calculation scheme, there is no correction for the difference in Reynolds number.

8.2 Correction of measured flow

Calculate the corrected volume flow as:

$$q_{v1,co} = q_{v1,te} \cdot \frac{u_g}{u_{te}}$$

Note: Tip speed ratio $\left(\frac{u_g}{u_{te}}\right)$ is equal to the shaft speed ratio. Scaled test are not allowed.

8.3 Correction of measured pressure

As the outlet pressure during testing differs from the target test outlet pressure, this pressure with its deviations shall be corrected to the guaranteed conditions. This allows the influences these deviations have on the guaranteed performance data to be recognized.

8.3.1 For dynamic compressors (blowers)

First, calculation of the corrected compression work shall be made with:

$$y_{s,co} = y_{s,te} \cdot \left(\frac{u_g}{u_{te}}\right)^2$$

Note: Tip speed ratio $\left(\frac{u_g}{u_{te}}\right)$ is equal to the shaft speed ratio

Then, the corrected pressure ratio is calculated:

$$\pi_{co} = \left[1 + \left(\frac{\kappa_g - 1}{\kappa_g}\right)\left(\frac{y_{s,co}}{R_{1,g} \cdot T_{1,g}}\right)\right]^{\left(\frac{\kappa_g}{\kappa_g - 1}\right)}$$

8.3.2 For positive displacement compressors (blowers):

For positive displacement compressors (blowers), the corrected pressure ratio shall be calculated as follows:

$$\pi_{co} = \left[\frac{y_{comb,te}}{R_1 \cdot T_1} - \frac{\kappa_g}{\kappa_g - 1}\left(\frac{1}{k} \cdot {v_i}^{\kappa_g - 1} - 1\right)\right] \cdot v_i$$

8.3.3 For positive displacement and dynamic compressors (blowers):

To calculate corrected outlet pressure, the follow equation shall be used using the appropriate corrected pressure ratio from above:

$$p_{2,co} = \pi_{co} \cdot p_{1,g}$$

8.4 Correction of specific energy demand

The tested specific energy demand e_{te} is the ratio of the measured power during test P_{te} and the measured flow $q_{v1,te}$.

$$e_{te} = \frac{P_{te}}{q_{v1,te}}$$

The specific energy demand is then corrected for the differences in density ρ and work y between test and guarantee conditions. Although the work is set in test according to the work in guarantee conditions, any difference between actual test conditions and the set point is considered in the correction formula.

Depending on the type of compressor, a different reference process for the specific compression work shall be used.

For positive displacement compressors (blowers) with internal compression (internal volume ratio $v_i > 1$) or without internal compression (internal volume ration $v_i = 1$) the specific energy demand shall be corrected as follows:

$$e_{CO} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{comb,g}}{y_{comb,te}} \cdot e_{te}$$

For dynamic compressors (blowers), the correction is as follows:

$$e_{CO} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{s,g}}{y_{s,te}} \cdot e_{te}$$

For the guaranteed specific compression work ($y_{comb,g}$ or $y_{s,g}$), the stated guarantee values of pressure and inlet temperature are to be used. For the tested specific compression work ($y_{comb,te}$ or $y_{s,te}$), the measured test values of pressure and inlet temperature are to be used.

8.5 Calculated package power

The power consumption of the package in the guarantee conditions can be expressed in two different ways. The power consumption of the tested package at guarantee conditions (effectively delivering the corrected flow $q_{v1,co}$) will be:

$$P_{co} = e_{co} \cdot q_{v1,co}$$

The power consumption of the package at guarantee conditions and at the guarantee flow (thus this is the case in which the package matches the guarantee flow $q_{v1,g}$):

$$P_{co,g} = e_{co} \cdot q_{v1,g}$$

8.6 Comparison of corrected values with guaranteed values

The test results are corrected to the specified operating conditions with the purpose of comparability with the guaranteed or specified performance.

The comparison shall include:

- Comparison of the corrected specific energy e_{co} with the guaranteed specific energy e_g
- Comparison of the corrected volume flow rate $q_{v1,co}$ with the guaranteed volume flow rate $q_{v1,g}$
- Comparison of the corrected absolute outlet pressure $p_{2,co}$ with the guaranteed $p_{2,g}$

Acceptance tolerances for values, corrected over guarantee, are specified in the table below.

Table 4: Acceptance Tolerances

Volume flow rate at specified conditions $q_{v1,g}$ (ft3/min)	Volume flow rate $q_{v1,g}$ %	Specific energy demand e %	Outlet Pressure p_2 %
$0 < q_{v1,g} \leq 17.7$	±7	±8	0÷ +1%
$17.7 < q_{v1,g} \leq 52.9$	±6	±7	0÷ +1%
$52.9 < q_{v1,g} \leq 529.7$	±5	±6	0÷ +1%
$529.7 < q_{v1,g}$	±4	±5	0÷ +1%

The corrected values are to be compared to the guaranteed values. If the values are within the limits as defined in this standard, then the compressor passes the test and is accepted. If the values are not within the limits as defined in this standard, then the compressor (blower) fails the test.

9 Test report

9.1 Test report content

At a minimum, the test report shall include the following:

- Test piping arrangement indicating pipe sizes and lengths, pressure and temperature measurement locations, flow measurement arrangement, valve location(s), and verification of compliance with ISO measurement standards.
- Original test logs including all recorded data required for calculations.
- Detailed sample calculation for one test point (Appendix B).
- Instrument calibration certificates.

- Date of test.
- Test report number.
- Compressor (blower) type, manufacturer, model, serial number, date of manufacture.
- Manufacturer’s package checklist per Appendix A.

9.2 Test results summary

Test results shall be summarized, an example of the format is given in mandatory Appendix B.

Appendix A – Equipment Checklist (Mandatory)

A package equipment checklist shall be provided which defines clearly the scope of testing. Where items are not included or not necessary, the column "Not Applicable" shall be marked. Where an item is not included but accounted for, the column "Accounted for value" shall be filled in.
Note: For different compressor types different components may be required.

Table 5: Manufacturer's Checklist (example, to be filled in by "X")

Section	Item	Included	Accounted for value (units)	Not applicable
Process air in/out	Process air filter		Δp (psi)	
	Inlet silencer		Δp (psi)	
	Inlet guide vanes		------------	
	Inlet unload valve or throttled valve		Δp (psi)	
	Outlet guide vanes		------------	
	Blow off valve*		Δp (psi)	
	Check valve		Δp (psi)	
	Outlet silencer		Δp (psi)	
	Additional inlet losses**		Δp (psi)	
	Additional outlet losses**		Δp (psi)	
Drive train	Main drive motor efficiency		η	
	Frequency inverter		η	
	EMC filter		η	
	Choke		η	
	Starter		η	
	Sinus filter		η	
	Power Transmission (gear box, belts)		η	
Ancillaries, electrical power input	Local control system		P (kW)	
	Cooling circulation (liquid)		P (kW)	
	Lubrication system		P (kW)	
	Main drive cooling fan		P (kW)	
	Cooling air fans		P (kW)	
	Heat exchanger fans		P (kW)	
	Additional components		P (kW)	

*pressure loss when closed
** manufacturers should detail the cause of the additional losses
The complete package needs to be tested. In case some required component is not included in the test, it shall be indicated how the component has been accounted for in the testing; indicated pressure drop should be accounted for either adapted test point or adapted test structure.

Appendix B – Calculation Examples (Mandatory)

B.1 General

The following examples illustrate the calculation methods in which the variables measured data during the factory test are to be corrected to the specified conditions and comparison against performance guarantees.

B.2 Test Examples

Test Example Number	**Compressor Type**	**Is Flow and Pressure Adjustable?**
1	Dynamic Compressor	The example is valid for fixed and variable flow machines
2	Positive Displacement Compressor	The example is valid for fixed and variable flow machines
3	Dynamic Compressor	Two speed test: The example is valid for fixed and variable flow machines with variable speed

B.2.1 The guaranteed performance point

The performance is defined at the package level. The air inlet condition is the same for all examples and the guarantee is the same for examples 1 & 2.

B.3 Test calculation example 1 (Dynamic Compressor)

B.3.1 Guarantee conditions

	Symbol	Numerical Value	Units
Inlet pressure	$p_{1,g}$	14.5	psia
Inlet temperature	$T_{1,g}$	527.7	°R
Inlet relative humidity	$\varphi_{rel,g}$	50	%

B.3.2 Object of Guarantee

	Symbol	Numerical Value	Units
Inlet volume flow	$q_{v1,g}$	3000	ft^3/min
Outlet pressure	$p_{2,g}$	22.5	psia
Electric power @ the electric input terminals	P_g	101.8	kW

Table 3: Acceptance tolerances

Volume flow rate at specified conditions $q_{v1,g}$	Volume flow rate $q_{v1,g}$	Specific Energy e	Outlet Pressure p_2
ft^3/min	%	%	%
$0 < q_{v1,g} \leq 17.7$	±7	±8	-0 / +1
$17.7 < q_{v1,g} \leq 52.9$	±6	±7	
$52.9 < q_{v1,g} \leq 529.7$	±5	±6	
$529.7 < q_{v1,g}$	±4	±5	
The tolerance band on package power is defined by the tolerance on specific energy consumption			

Acceptance tolerances for this specific test point: q_{v1}=±4%; e=±5%: p_2=+1%

B.3.3 General Calculation on Guarantee Data

General calculation made on the guaranteed performance data used for verification of similarity and acceptance comparison:

Step 1: Preconditions of the guarantee

- Ambient pressure $p_{1,g} = 14.5$ psia
- Ambient temperature $T_{1,g}$ =527.7°R
- Ambient humidity $\varphi_{rel,g}$ =50%

Step 2: Mixture of air and water vapor (humid air) on guarantee data

- Ambient temperate $t = (T_{1,g} - 459.7) = 68$°F
- Calculate the vapor pressure
- The partial pressure of water vapor is found using the steam tables: Vapor pressure
  $p_{vap.sat,g} = 0.339$ psi
- Relative humidity (4.4.1) $\varphi_{rel,g} = \frac{p_{vap,g}}{p_{vap,sat,g}} = 50 \therefore p_{vap,g} = 0.1695$ psi
- Vapor content (4.4.2) $x_{wet,g} = 0.622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,g}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,g}} = 0.00736\ lb/lb$
- Isentropic exponent dry (4.4.4) $\kappa_{dry} \approx 1.4$

- Isentropic exponent (4.4.4) $\kappa_{wet,g} = \kappa_{dry} \cdot (1 - 0.11 \cdot x_{wet,g}) = 1.399$
- Gas constant $R_{air} = 53.336$ ft-lb/lbmole °R
- Gas constant (4.4.5) $R_{wet,g} = R_{air} \cdot \left(1 + \frac{x}{x+1} \cdot 0.608\right) = 53.573 \frac{ft-lbf}{lbmole\,°R}$
- Ambient air density $\rho_{1,g} = \left(\frac{p_{1,g}}{R_{wet,g} \cdot T_{1,g}}\right) = 0.07386 \frac{lb}{ft^3}$
- Inlet specific volume $v_{1,g} = \left(\frac{1}{\rho_{1,g}}\right) = 13.5396 \quad \frac{ft^3}{lb}$
- Inlet volume flow rate (4.4.8) $q_{v1,g} = \frac{q_{m2,g}}{\rho_{1,g}} = 3000 \frac{ft^3}{min}$
- Mass flow rate $\therefore q_{m2,g} = q_{v1,g} \cdot \rho_{1,g}/60 = 3.693 \frac{lb}{s}$
- Outlet pressure $p_{2,g} = 22.5$ psia
- Package input power $P_g = 101.8$ kW
- Driver speed $n_g = 20500$ rpm
- Impeller Diameter D = 0.8333 ft

Step 3: Calculate Key Performance Indicators for Guarantee Data

- Specific isentropic work (4.4.6.1)

$$y_{s,g} = \frac{k_{wet,g}}{k_{wet,g}-1} \cdot R_{wet,g} \cdot T_{1,g} \cdot \left[\left(\frac{p_{2,g}}{p_{1,g}}\right)^{\left(\frac{k_{wet,g}-1}{k_{wet,g}}\right)} - 1\right] = 13233 \frac{ft-lbf}{lbm}$$

- Specific energy (4.4.9) $e_g = \frac{P_g}{q_{v1,g}/100} = 3.39 \frac{KW}{100\frac{ft^3}{min}}$
- Specific Package Work $y_g = P_g / q_{m2,g} * 737.56 = 20329 \frac{ft-lbf}{lbm}$
- Package isentropic efficiency $\eta_g = y_{s,g} / y_g * 100 = 65.1\%$

- Rotor tip speed (4.4.10) $u_g = \pi \cdot D \cdot \left(\frac{n_g}{60}\right) = 894 \ \frac{ft}{s}$

- Mach number (4.4.13) $Ma_g = \frac{u}{c_1} = \frac{u_g}{\sqrt{g.k_{wet,g} \cdot R_{wet,g} \cdot T_{1,g}}} = 0.793$

B.3.4 General Calculations on Inlet Test Data

General calculation made on the tested performance data used for verification of similarity and acceptance comparisons.

Test Inlet Conditions

- Ambient pressure $p_{1,te} = 14.2$ psia
- Ambient temperature $T_{1,te} = 539.7$ °R
- Ambient humidity $\varphi_{rel,te} = 40\%$

Step 4: Mixture of air and water vapor (humid air) on test data

- Ambient temperate $t_{te} = (T_{1,g} - 459.7) = 80$°F

- Calculate the vapor pressure

- The partial pressure of water vapor is found using the steam tables: Vapor pressure $p_{vap.sat,te} = 0.5069$ psi

- Relative humidity (4.4.1) $\varphi_{rel,te} = \frac{p_{vap,te}}{p_{vap,sat,te}} = 40\%$
  $p_{vap,te} = 0.2028$ psi

- Vapor content (4.4.2) $x_{air,te} = 0.622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,te}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,te}} = 0.00901 \ \frac{lb}{lb}$

- Isentropic exponent dry (4.4.4) $\kappa_{dry} \approx 1.4$

- Isentropic exponent (4.4.4) $\kappa_{wet,te} = \kappa_{dry} \cdot (1 - 0.11 \cdot x_{air,te}) = 1.399$

- Gas constant (4.4.5) $R_{wet,g} = R_{air} \cdot \left(1 + \frac{x}{x+1} \cdot 0.608\right) = 53.626 \ \frac{ft-lbf}{lbmole\,°R}$

- Ambient air density $\rho_{1,te} = \left(\frac{p_{1,te}}{R_{wet,te} \cdot T_{1,te}}\right) = 0.07065 \ \frac{lb}{ft^3}$

- Inlet specific volume $v_{1,te} = \left(\frac{1}{\rho_{1,te}}\right) = 14.1539 \ \frac{ft^3}{lb}$

Step 5: Calculate outlet pressure of the test setup
(dynamic compressors only)

- Required outlet pressure (7.8.2):

$$p_{2,te,req} = p_{1,te} \cdot \left[1 + \left(\frac{k_{te}-1}{k_{te}}\right)\left(\frac{y_{s,g}\cdot\left(\frac{u_{te}}{u_g}\right)^2}{(R_{te}\cdot T_{1,te})}\right)\right]^{\left(\frac{k_{te}}{k_{te}-1}\right)} = 21.65 \text{ psia}$$

B.3.5 Test example 1 Dynamic Compressor

Volume flow and speed is adjustable

Recorded Performance Data from the Test (te)

- Mass flow $q_{m,te} = = 3.56 \ \frac{lb}{s}$
- Inlet volume flow rate (4.4.8) $q_{v1,te} = \frac{q_{m,te}}{\rho_{1,te}} = 3023 \ \frac{ft^3}{min}$
- Measured Outlet pressure $p_{2,te} = 21.7$ psia
- Measured package input power $P_{pack,te} = 97.7$ kW
- Driver speed $n_{te} = 20300$ rpm
- Impeller Diameter D = 0.8333 ft

Step 6: Calculate and confirm limit indicators on the test conditions

- Limits on test speed (7.2):

$-3\% < Lim_{n,te} < 3\%$ $\quad Lim_{u,te} := \left(\frac{n_g}{n_{te}} - 1\right)\cdot 100 = 0.99\%$

- Limits on test density (7.3.1) : $-10\% < Lim_{\rho,te} < 10\%$ $\ Lim_{\rho,te} := \left(\frac{\rho 1_g}{\rho 1_{te}} - 1\right)\cdot 100 =$ 4.54%

Step 7: Calculate test Mach number

- Rotor tip speed (4.4.10) $u_{te} = \pi \cdot D \cdot \left(\frac{n_{te}}{60}\right) = 885.7 \quad \frac{ft}{s}$

$$Ma_{te} = \frac{u}{c_1} = \frac{u_{te}}{\sqrt{g \cdot k_{wet,te} \cdot R_{wet,te} \cdot T_{1,te}}} = 0.776$$

Step 8: Calculate key performance indicator for test
(dynamic compressors only)

- Specific isentropic work:

$$y_{s,te} = \frac{\kappa_{wet,te}}{\kappa_{wet,te-1}} \cdot R_{wet,te} \cdot T_{1,te} \cdot \left[\left(\frac{p_{2,te}}{p_{1,te}} \right)^{\left(\frac{\kappa_{wet,te-1}}{\kappa_{wet,te}} \right)} - 1 \right] = 13046 \quad \frac{ft-lb}{lb}$$

- Set test within similarity limits to qualify the test point (7.7.1.1)
  - **Work coefficient: -2%, +2%**
  - **Flow coefficient: -2%, +2%**
  - **Mach number: -5%, +5%**
- Work Coefficient

$$\left[\frac{y_{s,te}}{y_{s,g}} \cdot \left(\frac{u_g}{u_{te}} \right)^2 - 1 \right] \cdot 100 = 0.054\%$$

- Flow Coefficient

$$\left[\frac{q_{v1,te}}{q_{v1,g}} \cdot \left(\frac{u_g}{u_{te}} \right) - 1 \right] \cdot 100 = 1.77\%$$

- Mach Number

$$\left[\left(\frac{Ma_{te}}{Ma_g} \right) - 1 \right] \cdot 100 = -2.12\%$$

Step 9: Calculate Package Isentropic efficiency

- Specific package work $\quad y_{pack,te} = P_{pack,te} / q_{m2,te} * 737.56 = 20239$
- Package isentropic efficiency $\quad \eta_{te} = \frac{y_{s,te}}{y_{pack,te}} \cdot 100 = 64.5\%$

Step 10: Correction of test results to precondition and object of the guarantee

- Correction of volume flow (8.2) $\quad q_{v1,co} = q_{v1,te} \cdot \frac{u_g}{u_{te}} = 3053 \ \frac{ft^3}{min}$

- Correction of reference work, dynamic low pressure (8.3.1)

$$y_{s,co} = y_{s,te} \cdot \left(\frac{u_g}{u_{te}} \right)^2 = 13304 \ \frac{ft-lb}{lb}$$

- Corrected Pressure Ratio (8.3.1)

$$\pi_{co} = \left[1 + \left(\frac{k_g - 1}{k_g}\right) \cdot \left(\frac{y_{s,co}}{R_{1,g} \cdot T_{1,g}}\right)\right]^{\left(\frac{k_g}{k_g - 1}\right)} = 1.5552$$

- Correction for outlet pressure (8.3.3)

$$p_{2,co} = \pi_{co} \cdot p_{1,g} = 22.55 \text{ psia}$$

- Correction of specific energy (8.4)

$$e_{te} = \frac{P_{pack,te}}{q_{v1,te}} = 3.23 \frac{KW}{100\frac{ft^3}{min}}$$

(dynamic machine)

$$e_{co} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{s,g}}{y_{s,te}} \cdot e_{te} = 3.43 \frac{KW}{100\frac{ft^3}{min}}$$

- Correction for package power consumption (8.5)

$$P_{co} = e_{co} \cdot q_{v1,co} = 104.6 \text{ kW}$$

$$P_{co,g} = e_{co} \cdot q_{v1,g} = 102.8 \text{ kW}$$

Step 11: Compared deviations of guarantee to corrected test value

- Package specific energy (8.6) $\left(\frac{e_{co}}{e_g} - 1\right) \cdot 100 = 0.98\%$
- Volume flow rate (8.6) $\left(\frac{q_{v1,co}}{q_{v1,g}} - 1\right) \cdot 100 = 1.769\%$
- Absolute outlet pressure (8.6) $\left(\frac{p_{2,co}}{p_{2,g}} - 1\right) \cdot 100 = 0.022\%$

Test results summary

Turbo Compressor	Symbol	Unit	Ref Section	Numerical Values	
TEST DATA					
Test Number	#	number		1g	***1te***
Test Period start/end	time	Min.		-	***10***
Barometric Pressure	$p_{.amb.}$	psia		14.5	***14.2***
Inlet Pressure	p_1	psia	4.4.14	14.5	***14.2***
Inlet Temperature	T_1	°R	4.4.14	527.7	***539.7***
Relative Humidity	$\varphi_{rel,1}$	%	4.4.1	50	***40***
Isentropic exponent	κ_{wet}	ratio	4.4.4	1.399	***1.399***
Gas Constant	R_{wet}	ft-lb/lb	4.4.4	53.573	***53.626***
Inlet density	ρ_1	lb/ft^3		0.07386	***0.07065***
Outlet Pressure	p_2	psia		22.5	***21.7***
Discharge Temperature	T_2	°R		-	***-***
Compressor Speed [rpm]	n	rev/min		20500	***20300***
Supply voltage	U	V		460	***460***
Supply frequency	f	Hz		60	***60***
Package Input Power	P_{PAC}	kW		101.8	***97.7***
External Coolant Inlet Temperature	$T_{1,cool}$	°R		-	***-***
External Coolant Flow	$q_{m,cool}$	lb/s		-	***-***
Inlet Volume Flow Rate	q_{v1}	ft^3/min	4.4.7	3000	***3023***
compression work	Y_s	ft-lb/lb	4.4.5	13233	***13046***
Package isentropic efficiency	η	%	4.4.3	65.1%	**64.5%**
				Test results corrected to pre-conditions	**Pass/Fail**
Inlet Volume Flow Rate converted	q_{v1}	ft^3/min	8.2	3053	***Pass***
Outlet Pressure	p_2	psia	8.3.3	22.55	Pass
Package Specific Energy	e_{co}	kW/100ft^3/min	8.4	3.43	***Pass***

B.4 Test calculation example 2 (Positive Displacement Compressor)

B.4.1 Guarantee Conditions

	Symbol	Numerical Value	Units
Inlet pressure	$p_{1,g}$	14.5	psia
Inlet temperature	$T_{1,g}$	527.7	°R
Inlet relative humidity	$\varphi_{rel,g}$	50	%

B.4.2 Object of Guarantee

	Symbol	Numerical Value	Units
Inlet volume flow	$q_{v1,g}$	3000	ft^3/min
Outlet pressure	$p_{2,g}$	22.5	psia
Electric power @ the electric input terminals	P_g	101.8	kW
Driver speed	n_g	5000	rpm

Table 3: Acceptance tolerances

Volume flow rate at specified conditions $q_{v1,g}$	Volume flow rate $q_{v1,g}$	Specific Energy e	Outlet Pressure p_2
ft^3/min	%	%	%
$0 < q_{v1,g} \leq 17.7$	±7	±8	-0 / +1
$17.7 < q_{v1,g} \leq 52.9$	±6	±7	
$52.9 < q_{v1,g} \leq 529.7$	±5	±6	
$529.7 < q_{v1,g}$	±4	±5	
The tolerance band on package power is defined by the tolerance on specific energy consumption			

Acceptance tolerances for this specific test point: q_{v1}=±4%; e=±5%: p_2=+1%

B.4.3 General Calculation on Guarantee Data

General calculation made on the guaranteed performance data used for verification of similarity and acceptance comparison:

Step 1: Preconditions of the guarantee

- Ambient pressure $p_{1,g} = 14.5$ psia
- Ambient temperature $T_{1,g} = 527.7$ °R
- Ambient humidity $\varphi_{rel,g} = 50\%$

Step 2: Mixture of air and water vapor (humid air) on guarantee data

- Ambient temperature $t = (T_{1,g} - 459.7) = 68°F$
- Calculate the vapor pressure
- The partial pressure of water vapor is found using the steam tables: Vapor pressure $p_{vap.sat,g} = 0.339$ psi
- Relative humidity (4.4.1) $\varphi_{rel,g} = \frac{p_{vap,g}}{p_{vap,sat,g}} = 50\%$

  $p_{vap,g} = 0.1695$ psi
- Vapor content (4.4.2) $x_{wet,g} = 0.622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,g}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,g}} = 0.00736\ lb/lb$
- Isentropic exponent dry (4.4.4) $\kappa_{dry} \approx 1.4$
- Isentropic exponent (4.4.4) $\kappa_{wet,g} = \kappa_{dry} \cdot (1 - 0.11 \cdot x_{wet,g}) = 1.399$
- Gas constant (4.4.5) $R_{wet.g} = R_{air} \cdot \left(1 + \frac{x}{x+1} \cdot 0.608\right) = 53.573\ \frac{ft-lbf}{lbmole\ °R}$
- Ambient air density $\rho_{1,g} = \left(\frac{P_{1,g}}{R_{wet,g} \cdot T_{1,g}}\right) = 0.07386\ \frac{lb}{ft^3}$
- Inlet specific volume $v_{1,g} = \left(\frac{1}{\rho_{1,g}}\right) = 13.5396\ \frac{ft^3}{lb}$
- Inlet volume flow rate (4.4.8) $q_{v1,g} = \frac{q_{m2,g}}{\rho_{1,g}} = 3000\ \frac{ft^3}{min}$
- Mass flow rate $q_{m2,g} = q_{v1,g} \cdot \rho_{1,g}\ /60 = 3.693\ \frac{lb}{s}$
- Outlet pressure $p_{2,g} = 22.5$ psia
- Measured package input power $P_g = 101.8$ kW
- Driver speed $n_g = 5000$ rpm

Step 3: Calculate Key Performance Indicators for Guarantee Data

- Specific isentropic work (4.4.6.1)

$$y_{s,g} = \frac{k_{wet,g}}{k_{wet,g}-1} \cdot R_{wet,g} \cdot T_{1,g} \cdot \left[\left(\frac{p_{2,g}}{p_{1,g}}\right)^{\left(\frac{k_{wet,g}-1}{k_{wet,g}}\right)} - 1\right] = 13233 \ \frac{ft-lbf}{lbm}$$

- Reference process work for screw type. The specific value of $\boldsymbol{v}_i$ shall be calculated for the specific machine (4.4.5.3)

$v_{i,g} = 1.36$

$$y_{comb,g} = R_{wet,g} \cdot T_{1,g} \cdot \left[\frac{\frac{p_{2,g}}{p_{1,g}}}{v_{i,g}} + \frac{k_{wet,g}}{k_{wet,g}-1}\left(\frac{1}{k} \cdot {v_{i,g}}^{k-1} - 1\right)\right] = 13234 \ \frac{ft-lbf}{lbm}$$

- Specific energy (4.4.9) $e_g = \frac{P_g}{q_{v1,g}} = 3.39 \ \frac{KW}{\frac{ft^3}{min}}$
- Specific Package Work $y_g = P_g / q_{m2,g} * 737.56 = 20329 \ \frac{ft-lbf}{lbm}$
- Package isentropic efficiency $\eta_g = y_{s,g} / y_g * 100 = 65.1\%$

B.4.4 General Calculations on Inlet Test Data

General calculation made on the tested performance data used for verification of similarity and acceptance comparisons.

Test Inlet Conditions

- Ambient pressure $p_{1,te} = 14.2$ psia
- Ambient temperature $T_{1,te} = 539.7°R$
- Ambient humidity $\varphi_{rel,te} = 40\%$

Step 4: Mixture of air and water vapor (humid air) on test data

- Ambient temperate $t_{te} = (T_{1,g} - 459.7) = 80°F$
- Calculate the vapor pressure
- The partial pressure of water vapor is found using the steam tables: Vapor pressure

$$P_{vap.sat,te} = 0.5069 \text{ psi}$$

- Relative humidity (4.4.1) $\varphi_{rel,te} = \frac{p_{vap,te}}{p_{vap,sat,te}} = 40\%$

$$p_{vap,te} = 0.2028 \text{ psia}$$

- Vapor content (4.4.2) $x_{air,te} = 0.622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,te}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,te}} = 0.00901 \ \frac{lb}{lb}$

- Isentropic exponent dry (4.4.4) $\kappa_{dry} \approx 1.4$
- Isentropic exponent (4.4.4) $\kappa_{wet,te} = \kappa_{dry} \cdot (1 - 0.11 \cdot x_{air,te}) = 1.399$
- Gas constant (4.4.5) $R_{wet,te} = R_{air} \cdot \left(1 + \frac{x}{x+1} \cdot 0.608\right) = 53.626 \ \frac{ft-lbf}{lbmole\,°R}$
- Gas constant (4.454) $R_{wet,te} = R_{air} \cdot (1 - 0.11 \cdot x_{air,te}) = 53.626 \ \frac{ft-lbf}{lbmole°R}$
- Ambient air density $\rho_{1,te} = \left(\frac{p_{1,te}}{R_{wet,te} \cdot T_{1,te}}\right) = 0.07065 \ \frac{lb}{ft^3}$
- Inlet specific volume $v_{1,te} = \left(\frac{1}{\rho_{1,te}}\right) = 14.1539 \ \frac{ft^3}{lb}$

Step 5: Calculate outlet pressure of the test setup
(positive displacement compressors only)

- Required outlet pressure (7.8.1):

$v_{i,te}$ = 1.36 (given for specific internal compression unit)

$$p_{2,te,req} = p_{1,te} \cdot v_{i,te} \cdot \left[\frac{y_{comb,g}}{R_{wet,te} \cdot T_{1,te}} - \frac{k_{wet,te}}{k_{wet,te}-1}\left(\frac{1}{k_{wet,te}} \cdot {v_{i,te}}^{k_{wet,te}-1} - 1\right)\right] = 21.825 \text{ psia}$$

B.4.5 Test example 2 Positive Displacement Compressor Volume flow and speed is adjustable

Recorded Performance Data from the Test (te)

- Mass flow $q_{m,te} = \ = 3.56 \ \frac{klb}{s}$
- Inlet volume flow rate (4.4.8) $q_{v1,te} = \frac{q_{m,te}}{\rho_{1,te}} = 3023 \ \frac{ft^3}{min}$
- Measured Outlet pressure $p_{2,te} = 21.85$ psia
- Measured package input power $P_{pack,te} = 97.7$ kW
- Driver speed $n_{te} = 5010$ rpm

Step 6: Calculate and confirm limit indicators on the test conditions

- Limits on test speed (7.2):

$$-3\% \; < \; Lim_{n,te} \; < 3\% \qquad Lim_{u,te} \coloneqq \left(\frac{n_g}{n_{te}} - 1\right) \cdot 100 = -0.20\%$$

- Limits on test density (7.3.1):

$$-10\% \; < \; Lim_{\rho,te} \; < 10\% \quad Lim_{\rho,te} \coloneqq \left(\frac{\rho 1_g}{\rho 1_{te}} - 1\right) \cdot 100 = \; 4.54\%$$

Step 7: Calculate test Mach number

- Not utilized for positive displacement machines

Step 8: Calculate key performance indicator for test

(positive displacement compressors only)

- Specific isentropic work (4.4.6.1)

$$y_{s,te} = \frac{\kappa_{wet,te}}{\kappa_{wet,te-1}} \cdot R_{wet,te} \cdot T_{1,te} \cdot \left[\left(\frac{p_{2,te}}{p_{1,te}}\right)^{\left(\frac{\kappa_{wet,te-1}}{\kappa_{wet,te}}\right)} - 1\right] = \; 13271 \; \frac{ft-lb}{lb}$$

- Specific isentropic work for screw type. The specific value of $\boldsymbol{\nu}_i$ shall be calculated for the specific machine (4.4.5.3)

$$y_{comb,te} = R_{wet,te} \cdot T_{1,te} \cdot \left[\frac{\frac{p_{2,te}}{p_{1,te}}}{v_{i,te}} + \frac{\kappa_{wet,te}}{\kappa_{wet,te-1}}\left(\frac{1}{\kappa_{wet,te}} \cdot v_{i,te}{}^{\kappa_{wet,te}-1} - 1\right)\right] = \; 13271 \, \frac{ft-lb}{lb}$$

- Set test within similarity limits to qualify the test point (7.7.1.1)
  - **Work coefficient: -2%, +2%**
  - **Flow coefficient: -2%, +2%**

- Work Coefficient

$$\left[\frac{y_{comb,te}}{y_{comb,g}} - 1\right] \cdot 100 = 0.12 \; \%$$

- Flow Coefficient

$$\left[\frac{q_{v1,te}}{q_{v1,g}} \cdot \left(\frac{n_g}{n_{te}}\right) - 1\right] \cdot 100 = 0.57\%$$

Step 9: Calculate Package Isentropic efficiency

- Specific package work $y_{pack,te} = P_{pack,te} / q_{m2,te} * 737.56 = 20239$
- Package isentropic efficiency $\eta_{te} = \frac{y_{s.te}}{y_{pack,te}} \cdot 100 = 65.6\%$

Step 10: Correction of test results to precondition and object of the guarantee

- Correction of volume flow (8.2) $q_{v1,co} = q_{v1,te} \cdot \frac{n_g}{n_{te}} = 3017 \ \frac{ft^3}{min}$

- Correction of reference work, positive displacement low pressure

$y_{comb,co} = y_{comb,te} = 13271 \ \frac{ft-lb}{lb}$

- Corrected Pressure Ratio (8.3.2)

$$\pi_{co} = v_{i,g} \cdot \left[\frac{y_{comb,te}}{R_{wet,g} \cdot T_{1,g}} - \frac{k_{wet,g}}{k_{wet,g}-1}\left(\frac{1}{k_{wet,g}} \cdot v_{i,g}{}^{k_{wet,g}-1} - 1\right)\right] = 1.5535$$

- Correction for outlet pressure (8.3.3)

$p_{2,co} = \pi_{co} \cdot p_{1,g} = 22.526$ psia

- Correction of specific energy (8.4)

$$e_{te} = \frac{P_{pack,te}}{q_{v1,te}} = 3.23 \ \frac{KW}{100\frac{ft^3}{min}}$$

(positive displacement machine)

$$e_{co} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{comb,g}}{y_{comb,te}} \cdot e_{te} = 3.368 \ \frac{KW}{100\frac{ft^3}{min}}$$

- Correction for package power consumption (8.5)

$P_{co} = e_{co} \cdot q_{v1,co} = 101.6$ kW

$P_{co,g} = e_{co} \cdot q_{v1,g} = 101.0$ kW

Step 11: Compared deviations of guarantee to corrected test value

- Package specific energy (8.6) $\left(\frac{e_{co}}{e_g} - 1\right) \cdot 100 = -0.73 \ \%$
- Volume flow rate (8.6) $\left(\frac{q_{v1,co}}{q_{v1,g}} - 1\right) \cdot 100 = 0.57 \ \%$
- Absolute outlet pressure (8.6) $\left(\frac{p_{2,co}}{p_{2,g}} - 1\right) \cdot 100 = 0.12 \ \%$

Test results summary

Screw Compressor	Symbol	Unit (Metric)	Ref Section	Numerical Values	
TEST DATA					
Test Number	#	number		1g	***1te***
Test Period start/end	time	Min.		-	***10***
Barometric Pressure	$p_{.amb}$	psia		14.5	***14.2***
Inlet Pressure	$p_{1.}$	psia	4.4.14	14.5	***14.2***
Inlet Temperature	T_1	°R	4.4.14	527.7	***539.7***
Relative Humidity	$\varphi_{rel,1}$	ratio	4.4.1	50	***40***
Isentropic exponent	κ_{wet}	ratio	4.4.4	1.399	***1.399***
Gas Constant	R_{wet}	ft-lb/lb	4.4.4	53.573	***53.626***
Inlet density	ρ_1	lb/ft^3		0.07386	***0.07065***
Outlet Pressure	$p_{2.abs}$	psia		22.5	***21.85***
Discharge Temperature	T_2	°R		-	***-***
Compressor Speed [rpm]	n	rev/min		5000	***5010***
Supply voltage	U	V		460	***460***
Supply frequency	f	Hz		60	***60***
Package Input Power	P_{PAC}	kW		101.8	***97.7***
External Coolant Inlet Temperature	$T_{1,cool}$	°R		-	***-***
External Coolant Flow	$q_{m,cool}$	lb/s		-	***-***
Inlet Volume Flow Rate	q_{v1}	ft^3/min	4.4.7	3000	***3023***
Compression work	$y_{,comb}$	ft-lb/lb	4.4.5	13234	***13271***
Package isentropic efficiency	η	%	4.4.3	65.1%	**65.6%**
				test results corrected to preconditions	***Pass/Fail***
Inlet Volume Flow Rate converted	q_{v1}	ft^3/min	8.2	3017	***Pass***
Outlet Pressure	p_2	psia	8.3.3	22.526	***Pass***
Package Specific Energy	e_{co}	kW/100ft^3/min	8.4	3.37	***Pass***

B.5 Test calculation example 3 (dynamic, 2-speed test)

The 2-speed test consists of 3 parts.

1. Test1 running at specified rotational speed $n_{te} = n_g$
2. Test2 running at a calculated rotational speed based on the ratio of Mach numbers for guarantee and test 1.
3. Test 1 and test2 are combined to evaluate the corrected package power.

B.5.1 Guarantee Conditions

	Symbol	Numerical Value	Units
Inlet pressure	$p_{1,g}$	14.7	psia
Inlet temperature	$T_{1,g}$	563.7	°R
Inlet relative humidity	$\varphi_{rel,g}$	60	%

B.5.2 Object of Guarantee

	Symbol	Numerical Value	Units
Inlet volume flow	$q_{v1,g}$	875	ft³/min
Outlet pressure	$p_{2,g}$	20.3	psia
Electric power @ the electric input terminals	P_g	19.76	kW

Table 3: Acceptance tolerances

Volume flow rate at specified conditions $q_{v1,g}$	Volume flow rate at specified conditions $q_{v1,g}$	Volume flow rate at specified conditions $q_{v1,g}$	Volume flow rate $q_{v1,g}$	Specific energy e	Outlet Pressure p_2	Idle power P_{idle}	Standby power $P_{standby}$
(m³/s) x10⁻³	**(m³/min)**	**(m³/h)**	**%**	**%**	**%**	**%**	**%**
0	0	0	n/a	n/a	n/a	±10	±10
$0 < q_{v1,g} \leq 8.3$	$0 < q_{v1,g} \leq 0.5$	$0 < q_{v1,g} \leq 30$	±7	±8	0÷ +1	n/a	n/a
$8{,}3 < q_{v1,g} \leq 25$	$0{,}5 < q_{v1,g} \leq 1.5$	$30 < q_{v1,g} \leq 90$	±6	±7			
$25 < q_{v1,g} \leq 250$	$1.5 < q_{v1,g} \leq 15$	$90 < q_{v1,g} \leq 900$	±5	±6			
$250 < q_{v1,g}$	$15 < q_{v1,g}$	$900 < q_{v1,g}$	±4	±5			

[a] The tolerance band on package power is defined by the tolerance on specific energy consumption.

Table 3: Acceptance tolerances

Volume flow rate at specified conditions $q_{v1,g}$	Volume flow rate $q_{v1,g}$	Specific Energy e	Outlet Pressure p_2
ft^3/min	%	%	%
$0 < q_{v1,g} \leq 17.7$	±7	±8	-0 / +1
$17.7 < q_{v1,g} \leq 52.9$	±6	±7	
$52.9 < q_{v1,g} \leq 529.7$	±5	±6	
$529.7 < q_{v1,g}$	±4	±5	
The tolerance band on package power is defined by the tolerance on specific energy consumption			

Acceptance tolerances for this specific test point: q_{v1}=±4%; e=±5%: p_2=+1%

B.5.3 General Calculation on Guarantee Data

General calculation made on the guaranteed performance data used for verification of similarity and acceptance comparison:

Step 1: Preconditions of the guarantee

- Ambient pressure $p_{1,g} = 14.7$ psia
- Ambient temperature $T_{1,g} = 563.7$ °R
- Ambient humidity $\varphi_{rel,g} = 60\%$

Step 2: Mixture of air and water vapor (humid air) on guarantee data

- Ambient temperate $t = (T_{1,g} - 459.7) = 104°F$
- Calculate the vapor pressure
- Vapor pressure $P_{vap.sat,g} = 1.0697$ psi
- Relative humidity (4.4.1) $\varphi_{rel,g} = \frac{p_{vap,g}}{p_{vap,sat,g}} = 60\%$

  $p_{vap,g} = 0.6418$ psi
- Vapor content (4.4.2) $x_{wet,g} = 0.622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,g}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,g}} = 0.0284 \frac{lb}{lb}$
- Isentropic exponent dry (4.4.4) $\kappa_{dry} \approx 1.4$
- Isentropic exponent (4.4.4) $\kappa_{wet,g} = \kappa_{dry} \cdot (1 - 0.11 \cdot x_{wet,g}) = 1.396$

- Gas constant $R_{air} = 53.336 \frac{ft-lbf}{lbmole°R}$
- Gas constant (4.4.5) $R_{wet,g} = R_{air} \cdot \left(1 + \frac{x}{x+1} \cdot 0.608\right) = 54.232 \frac{ft-lbf}{lbmole°R}$
- Ambient air density $\rho_{1,g} = \left(\frac{P_{1,g}}{R_{wet,g} \cdot T_{1,g}}\right) = 0.06924 \frac{lb}{ft^3}$
- Inlet specific volume $v_{1,g} = \left(\frac{1}{\rho_{1,g}}\right) = 14.442 \frac{ft^3}{lb}$
- Inlet volume flow rate (4.4.8) $q_{v1,g} = \frac{q_{m2,g}}{\rho_{1,g}} = 875 \frac{ft^3}{min}$
- Mass flow rate $\therefore q_{m2,g} = q_{v1,g} \cdot \rho_{1,g} = 1.0098 \frac{lb}{s}$
- Outlet pressure $p_{2,g} = 20.3$ psia
- Package input power $P_g = 19.76$ kW
- Driver speed $n_g = 29750$ rpm
- Impeller Diameter D = 0.524 ft

Step 3: Calculate Key Performance Indicators for Guarantee Data

- Specific isentropic work (4.4.6.1)

$$y_{s,g} = \frac{k_{wet,g}}{k_{wet,g}-1} \cdot R_{wet,g} \cdot T_{1,g} \cdot \left[\left(\frac{p_{2,g}}{p_{1,g}}\right)^{\left(\frac{k_{wet,g}-1}{k_{wet,g}}\right)} - 1\right] = 10333 \frac{ft-lb}{lb}$$

- Specific energy (4.4.9) $e_g = \frac{P_g}{q_{v1,g}} = 2.258 \frac{KW}{\frac{100 ft^3}{min}}$
- Specific Package Work $y_g = P_g / q_{m2,g} * 737.56 = 14434 \frac{ft-lbf}{lbm}$
- Package isentropic efficiency $\eta_g = y_{s,g} / y_g * 100 = 71.6\%$
- Rotor tip speed (4.4.10) $u_g = \pi \cdot D \cdot \frac{n_g}{60} = 816.2 \frac{ft}{s}$
- Mach number(4.4.13) $Ma_g = \frac{u}{c_1} = \frac{u_g}{\sqrt{k_{wet,g} \cdot g * R_{wet,g} \cdot T_{1,g}}} = 0.696$

B.5.4 General Calculations on Inlet Test Data

General calculation made on the tested performance data used for verification of similarity and acceptance comparisons.

Test Inlet Conditions

- Ambient pressure $P_{1,te} = 14.47$ psia
- Ambient temperature $T_{1,te} = 482.7$ °R
- Ambient humidity $\varphi_{rel,te} = 60\%$

Step 4: Mixture of air and water vapor (humid air) on test data

- Ambient temperate $t_{te} = (T_{1,g} - 459.7) = 23°F$
- Calculate the vapor pressure
- Vapor pressure $P_{vap.sat,te} = 0.0641$ psi
- Relative humidity (4.4.1) $\varphi_{rel,te} = \frac{p_{vap,te}}{p_{vap,sat,te}} = 60\% \therefore p_{vap,te} = 0.03846$ psi
- Vapor content (4.4.2) $x_{air,te} = 0.622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,te}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,te}} = 0.00166 \frac{lb}{lb}$
- Isentropic exponent dry (4.4.4) $\kappa_{dry} \approx 1.4$
- Isentropic exponent (4.4.4) $\kappa_{wet,te} = \kappa_{dry} \cdot (1 - 0.11 \cdot x_{air,te}) = 1.4$
- Gas constant $R_{air} = 53.336 \frac{ft-lb}{lbmole\ °R}$
- Gas constant (4.4.5) $R_{wet.te} = R_{air} \cdot \left(1 + \frac{x}{x+1} \cdot 0.608\right) = 53.390 \frac{ft-lbf}{lbmole\ °R}$
- Ambient air density $\rho_{1,te} = \left(\frac{P_{1,te}}{R_{wet,te} \cdot T_{1,te}}\right) = 0.08085 \frac{-lb}{ft^3}$
- Inlet specific volume $\nu_{1,te} = \left(\frac{1}{\rho_{1,te}}\right) = 12.368 \frac{ft^3}{lb}$

Step 5: Calculate outlet pressure of the test setup
(dynamic compressors only)

- Required outlet pressure (7.8.2):

$$p_{2,te,req} = p_{1,te} \cdot \left[1 + \left(\frac{k_{te}-1}{k_{te}}\right)\left(\frac{y_{s,g} \cdot \left(\frac{u_{te}}{u_g}\right)^2}{(R_{te} \cdot T_{1,te})}\right)\right]^{\left(\frac{k_{te}}{k_{te}-1}\right)} = 21.151 \text{ psia}$$

B.5.5 Test example 1 Dynamic Compressor –test1

Volume flow and speed is adjustable

Recorded Performance Data from the Test (te)

- Mass flow $q_{m,te} = \ = 0.882 \ \frac{lb}{s}$
- Inlet volume flow rate (4.4.8) $q_{v1,te} = \frac{q_{m,te}}{\rho_{1,te}} = 654.5 \ \frac{ft^3}{min}$
- Measured Outlet pressure $p_{2,te} = $ 21.15 psia
- Measured package input power $P_{pack,te} = $ 20.0 kW
- Driver speed $n_{te} = $ 29750 rpm
- Impellor Diameter $D = 0$ 0.524 ft

Step 6: Calculate and confirm limit indicators on the test conditions

- Limits on test speed (7.2):

$-3\% < Lim_{n,te} < 3\%$ $\quad Lim_{u,te} := \left(\frac{n_g}{n_{te}} - 1\right) \cdot 100 = 0\%$

- Limits on test density (7.3.1):

$-10\% < Lim_{\rho,te} < 10\%$ $\quad Lim_{\rho,te} := \left(\frac{\rho 1_g}{\rho 1_{te}} - 1\right) \cdot 100 = -14.36\%$

Step 7: Calculate test Mach number

- Rotor tip speed $u_{te} = \pi \cdot D \cdot \frac{n_{te}}{60} = 816.2 \ \frac{ft}{s}$

$$Ma_{te} = \frac{u}{c_1} = \frac{u_{te}}{\sqrt{k_{wet,te} \cdot 32.17 * R_{wet,te} \cdot T_{1,te}}} = 0.7577$$

Step 8: Calculate key performance indicator for test
(dynamic compressors only)

- Specific isentropic work (7.8.2):

$$y_{s,te} = y_{s,g} \cdot \left(\frac{u_{te}}{u_g}\right)^2 = 10359 \quad \frac{ft-lb}{klb}$$

OR

$$y_{s,te} = \frac{\kappa_{wet,te}}{\kappa_{wet,te-1}} \cdot R_{wet,te} \cdot T_{1,te} \cdot \left[\left(\frac{p_{2,te}}{p_{1,te}}\right)^{\left(\frac{\kappa_{wet,te-1}}{\kappa_{wet,te}}\right)} - 1\right] = 10359 \quad \frac{ft-lb}{klb}$$

- Set test within similarity limits to qualify the test point (7.7.1.1)
  - **Work coefficient: -2%, +2%**
  - **Flow coefficient: -2%, +2%**
  - **Mach number: -5%, +5%**
- Work Coefficient

$$\left[\frac{y_{s,te}}{y_{s,g}} \cdot \left(\frac{u_g}{u_{te}}\right)^2 - 1\right] \cdot 100 = 0.25\%$$

- Flow Coefficient

$$\left[\frac{q_{v1,te}}{q_{v1,g}} \cdot \left(\frac{u_g}{u_{te}}\right) - 1\right] \cdot 100 = -25.2\%$$

- Mach Number

$$\left[\left(\frac{Ma_{te}}{Ma_g}\right) - 1\right] \cdot 100 = 8.75\%$$

Step 9: Calculate Package work / efficiency

- Specific package work $y_{pack,te} = P_{pack,te} / q_{m2,te} * 737.56 = 16724$
- Package isentropic efficiency $\eta\_te = \frac{y_{s,te}}{y_{pack,te}} \cdot 100 = 61.9\%$

Step 10: Correction of test results to precondition and object of the guarantee

- Correction of volume flow (8.2) $q_{v1,co} = q_{v1,te} \cdot \frac{u_g}{u_{te}} = 654.5 \ \frac{ft^3}{min}$

- Correction of reference work, dynamic low pressure (8.3.1)

$$y_{s,co} = y_{s,te} \cdot \left(\frac{u_g}{u_{te}}\right)^2 = 10359 \quad \frac{ft-lb}{lb}$$

- Corrected Pressure Ratio (8.3.1)

$$\pi_{co} = \left[1 + \left(\frac{k_g - 1}{k_g}\right) \cdot \left(\frac{y_{s,co}}{R_{1,g} \cdot T_{1,g}}\right)\right]^{\left(\frac{k_g}{k_g - 1}\right)} = 1.38$$

- Correction for outlet pressure (8.3.3)

$p_{2,co} = \pi_{co} \cdot p_{1,g} =$ 20.316 psia

- Correction of specific energy (8.4)

$e_{co} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{s,g}}{y_{s,te}} \cdot e_{te} = 2.61$ (dynamic machine) $\frac{KW}{\frac{100ft^3}{min}}$

- Correction for package power consumption (8.5)

$P_{co} = e_{co} \cdot q_{v1,co} =$ 17.0 kW

Step 11: Compared deviations of guarantee to corrected test value

- Package specific energy (8.6) $\left(\frac{e_{co}}{e_g} - 1\right) \cdot 100 = 15.58\%$
- Volume flow rate (8.6) $\left(\frac{q_{v1,co}}{q_{v1,g}} - 1\right) \cdot 100 = -25.2\%$
- Absolute outlet pressure (8.6) $\left(\frac{p_{2,co}}{p_{2,g}} - 1\right) \cdot 100 = 0.1\%$

Step 12: Calculate additional parameters for two speed test

- Measured Outlet temperature from test 1 (needed for two speed test)

$$T_{2,te} = 563.43°R$$

- Isentropic Temperature Rise (100% efficiency) $(T_2\text{-}T_1)_{isen} = ((p_{2,te}/p_{1,te})\text{^}(k_{wet,te}-1)/k_{wet,te} - 1) * T_{1,te} = 55.41$ °R

- Actual Temperature Rise = $T_{2,te} - T_{1,te} = 80.73$ °R

- Gas efficiency internal compression work related to isentropic process

$$\eta_{s,te} = \frac{(T_2 - T1)isen}{T2,te - T1,te} = 0.686$$

- Driver efficiency package work related to specific internal compression work

$$\eta_{i,driver} = \frac{\eta_{pack,te}}{\eta_{s,te}} = 0.902$$

B.5.6 Two speed test, Test 2

Now that it has been determined that the test cannot be run without reducing the speed
Precondition of the guarantee
The data must fulfill the ideal gas las
The gas treated in the codes is ambient air

Step 1: Preconditions of the guarantee

- Ambient pressure $p_{1,g} = 14.7\text{psia}$
- Ambient temperature $T_{1,g} = 563.7\ °\text{R}$
- Ambient humidity $\varphi_{rel,g} = 60\%$

Step 2: Mixture of air and water vapor (humid air) on guarantee data

- Ambient temperate $t = (T_{1,g} - 459.7) = 104\ °\text{F}$
- Calculate the vapor pressure
- Vapor pressure $P_{vap.sat,g} = 1.0697\ \text{psi}$
- Relative humidity (4.4.1) $\varphi_{rel,g} = \frac{P_{vap,g}}{P_{vap,sat,g}} = 60\%$

  $p_{vap,g} = 0.6418\ \text{psi}$
- Vapor content (4.4.2) $x_{wet,g} = 0.622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,g}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,g}} = 0.0284\ \frac{lb}{lb}$
- Isentropic exponent dry (4.4.4) $\kappa_{dry} \approx 1.4$
- Isentropic exponent (4.4.4) $\kappa_{wet,g} = \kappa_{dry} \cdot (1 - 0.11 \cdot x_{wet,g}) = 1.396$
- Gas constant $R_{air} = 53.336\ \frac{ft-lbf}{lbmoleR}$
- Gas constant (4.4.5) $R_{wet,g} = R_{air} \cdot \left(1 + \frac{x}{x+1} \cdot 0.608\right) = 54.232\ \frac{ft-lbf}{lbmole\ °R}$
- Ambient air density $\rho_{1,g} = \left(\frac{P_{1,g}}{R_{wet,g} \cdot T_{1,g}}\right) = 0.06924\ \frac{lb}{ft^3}$

- Inlet specific volume $v_{1,g} = \left(\frac{1}{\rho_{1,g}}\right) = 14.442 \quad \frac{ft^3}{lb}$

- Inlet volume flow rate (4.4.8) $q_{v1,g} = \frac{q_{m2,g}}{\rho_{1,g}} = 875 \quad \frac{ft^3}{min}$

- Mass flow rate $\therefore\ q_{m2,g} = q_{v1,g} \cdot \rho_{1,g} = 1.0098 \ \frac{lb}{s}$

- Outlet pressure $p_{2,g} = 20.3$ psia

- Package input power $P_g = 19.76$ kW

- Driver speed $n_g = 29750$ rpm

- Impeller Diameter D = 0.524 ft

Step 3: Calculate Key Performance Indicators for Guarantee Data

- Specific isentropic work (4.4.6.1)

$$y_{s,g} = \frac{k_{wet,g}}{k_{wet,g}-1} \cdot R_{wet,g} \cdot T_{1,g} \cdot \left[\left(\frac{p_{2,g}}{p_{1,g}}\right)^{\left(\frac{k_{wet,g}-1}{k_{wet,g}}\right)} - 1\right] = 10333 \ \frac{ft-lb}{lb}$$

- Specific energy (4.4.9) $e_g = \frac{P_g}{q_{v1,g}} = 2.258 \ \frac{KW}{\frac{100ft^3}{min}}$

- Specific Package Work $y_g = P_g / q_{m2,g} * 737.56 = 14434 \ \frac{ft-lbf}{lbm}$

- Package isentropic efficiency $\eta_g = y_{s,g} / y_g * 100 = 71.6\%$

- Rotor tip speed (4.4.10) $u_g = \pi \cdot D \cdot \frac{n_g}{60} = 816.2 \quad \frac{ft}{s}$

- Mach number(4.4.13) $Ma_g = \frac{u}{c_1} = \frac{u_g}{\sqrt{k_{wet,g} \cdot g * R_{wet,g} \cdot T_{1,g}}} = 0.696$

Step 4: Mixture of air and water vapor (humid air) on test data

- Ambient temperate $t_{te} = (T_{1,g} - 459.7) = 23°F$

- Calculate the vapor pressure

- Vapor pressure $P_{vap.sat,te} = 0.0641$psi

- Relative humidity (4.4.1) $\varphi_{rel,te} = \frac{p_{vap,te}}{p_{vap,sat,te}} = 60\% \;\therefore\; p_{vap,te} = 0.03846$ psi

- Vapor content (4.4.2) $x_{air,te} = 0.622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,te}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,te}} = 0.00166 \frac{lb}{lb}$

- Isentropic exponent dry (4.4.4) $\kappa_{dry} \approx 1.4$

- Isentropic exponent (4.4.4) $\kappa_{wet,te} = \kappa_{dry} \cdot \left(1 - 0.11 \cdot x_{air,te}\right) = 1.4$

- Gas constant $R_{air} = 53.336 \frac{ft-lb}{lbmole\,°R}$

- Gas constant (4.4.5) $R_{wet,te} = R_{air} + \frac{x}{x+1} \cdot 0.608 = 53.390 \frac{ft-lb}{lbmole\,°R}$

- Ambient air density $\rho_{1,te} = \left(\frac{P_{1,te}}{R_{wet,te} \cdot T_{1,te}}\right) = 0.08085 \frac{lb}{ft^3}$

- Inlet specific volume $v_{1,te} = \left(\frac{1}{\rho_{1,te}}\right) = 12.368 \frac{ft^3}{klb}$

Step 5: Calculate outlet pressure of the test setup
(dynamic compressors only)

- Required outlet pressure (7.8.2):

$$p_{2,te,req} = p_{1,te} \cdot \left[1 + \left(\frac{k_{te}-1}{k_{te}}\right)\left(\frac{y_{s,g} \cdot \left(\frac{u_{te}}{u_g}\right)^2}{(R_{te} \cdot T_{1,te})}\right)\right]^{\left(\frac{k_{te}}{k_{te}-1}\right)} = 19.98 \text{ psia}$$

Step 6: Calculate operating speed for Test 2

$$n_{te} = n_g \times \frac{Ma_g}{Ma_{te,test\,1}} = 27315 \text{ rpm}$$

B.5.7 Test example 3 Dynamic Compressor –test2

Volume flow and speed is adjustable

Recorded Performance Data from the Test (te)

- Mass flow $q_{m,te} = 1.0826 \frac{lb}{s}$

- Inlet volume flow rate (4.4.8) $q_{v1,te} = \frac{q_{m,te}}{\rho_{1,te}} = 804.25 \frac{ft^3}{min}$

- Measured Outlet pressure $p_{2,te} = 20.0$ psia

- Measured package input power $P_{pack,te} = 18.03$ kW
- Driver speed $n_{te} = 27315$ rpm
- Impeller Diameter D = 0.524 ft

Step 7: Calculate and confirm limit indicators on the test conditions

- Limits on test speed (7.2):

$$-3\% < Lim_{n,te} < 3\% \qquad Lim_{u,te} \coloneqq \left(\frac{n_g}{n_{te}} - 1\right) \cdot 100 = 8.91\%$$

- Limits on test density (7.3.1):

$$-10\% < Lim_{\rho,te} < 10\% \quad Lim_{\rho,te} \coloneqq \left(\frac{\rho 1_g}{\rho 1_{te}} - 1\right) \cdot 100 = -14.36\%$$

Step 8: Calculate test Mach number

- Rotor tip speed (7.5) $u_{te} = \pi \cdot D \cdot n_{te}/60 = 749.4 \ \frac{ft}{s}$

$$Ma_{te} = \frac{u}{c_1} = \frac{u_{te}}{\sqrt{k_{wet,te} \cdot R_{wet,te} \cdot T_{1,te}}} = 0.695$$

Step 9: Calculate key performance indicator for test

(dynamic compressors only)

- Specific isentropic work (7.7.2):

$$y_{s,te} = \frac{\kappa_{wet,te}}{\kappa_{wet,te}-1} \cdot R_{wet,te} \cdot T_{1,te} \cdot \left[\left(\frac{p_{2,te}}{p_{1,te}}\right)^{\left(\frac{\kappa_{wet,te}-1}{\kappa_{wet,te}}\right)} - 1\right] = 8739 \ \frac{ft-lb}{lb}$$

- Set test within similarity limits to qualify the test point (7.7.1.1)
  - **Work coefficient: -2%, +2%**
  - **Flow coefficient: -2%, +2%**
  - **Mach number: -5%, +5%**

- Work Coefficient

$$\left[\frac{y_{s,te}}{y_{s,g}} \cdot \left(\frac{u_g}{u_{te}}\right)^2 - 1\right] \cdot 100 = 0.32\ \%$$

- Flow Coefficient

$$\left[\frac{q_{v1,te}}{q_{v1,g}} \cdot \left(\frac{u_g}{u_{te}}\right) - 1\right] \cdot 100 = 0.11 \ \%$$

- Mach Number

$$\left[\left(\frac{Ma_g}{Ma_{te}}\right) - 1\right] \cdot 100 = -0.15 \ \%$$

Step 10: Calculate Package work / efficiency

- Specific package work $y_{pack,te} = P_{pack,te} / q_{m2,te} * 737.56 = 12284$
- Package isentropic efficiency $\eta_{te} = \frac{y_{s,te}}{y_{pack,te}} \cdot 100 = 71.1\%$

Step 11: Correction of test results to precondition and object of the guarantee

- Correction of volume flow (8.2) $q_{v1,co} = q_{v1,te} \cdot \frac{u_g}{u_{te}} = 875.9 \ \frac{ft^3}{min}$

- Correction of reference work, dynamic low pressure (8.3.1)

$$y_{s,co} = y_{s,te} \cdot \left(\frac{u_g}{u_{te}}\right)^2 = 10366 \ \frac{ft-lb}{klb}$$

- Corrected Pressure Ratio (8.3.1)

$$\pi_{co} = \left[1 + \left(\frac{k_g - 1}{k_g}\right) \cdot \left(\frac{y_{s,co}}{R_{1,g} \cdot T_{1,g}}\right)\right]^{\left(\frac{k_g}{k_g - 1}\right)} = 1.382$$

- Correction for outlet pressure (8.3.3)

$$p_{2,co} = \pi_{co} \cdot p_{1,g} = 20.32 \text{ psia}$$

- Correction of specific energy (8.4)
  (dynamic machine)

$$e_{co} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{s,g}}{y_{s,te}} \cdot e_{te} = 2.27 \qquad \frac{KW}{\frac{100ft^3}{min}}$$

- Correction for package power consumption (8.5)

$$P_{co} = e_{co} \cdot q_{v1,co} = 19.87 \text{ kW}$$

Step 12: Compared deviations of guarantee to corrected test value

- Package specific energy (8.6) $\left(\frac{e_{co}}{e_g} - 1\right) \cdot 100 = 0.52\%$
- Volume flow rate (8.6) $\left(\frac{q_{v1,co}}{q_{v1,g}} - 1\right) \cdot 100 = 0.11\%$
- Absolute outlet pressure (8.6) $\left(\frac{p_{2,co}}{p_{2,g}} - 1\right) \cdot 100 = 0.1\%$

Step 13: Calculate additional parameters for two speed test

- Outlet temperature (needed for two speed test) $T_{2,te} = 541.29°\text{R}$

- Gas efficiency internal compression work related to isentropic process

$$\eta_{s,te} = \frac{(T_2 - T1)isen}{T2,te - T1,te} = 0.798$$

- Driver efficiency package work related to specific internal compression work

$$\eta_{i,driver} = \frac{\eta_{pack,te}}{\eta_{s,te}} = 0.893$$

B.5.8 Two speed test part 3

Object of the guarantee

- Inlet volume flow $q_{v1,g} =$ 875 ft³/min
- Specific Isentropic compression "work" $y_{s,g} =$ 10333 ft-lb/lb
- Input wire power to the package $P_{pack,g} =$ 19.76 kW

Measured performance of test 1 and 2

- Measured driver efficiency test2 $\eta_{i,driver,test2} = 0.893$
- Measure Package power test2 $P_{co,g.test2} = 19.87$ kW

- Measure drive efficiency test1 $\eta_{i,driver,test1} = 0.903$

Step 14: Calculate result of the two speed result (1 & 2)

Final result of the two speed test

- Corrected package power test 1.2

- $P_{co,g,test1.2} = P_{co,g,test2} \cdot \frac{\eta_{i,\text{driver}.test2}}{\eta_{i,\text{driver}.test1}} = 19.65$ kW

- Corrected energy density test 1.2
- $e_{co.g.test1.2} = \frac{P_{co.g.test1.2}}{q_{\mathrm{V1.g}}} = 2.246 \frac{\mathrm{kW}}{\mathrm{100ft3/min}}$
- Corrected package power efficiency test 1.2 $\eta_{te,1,2} = \eta_{te.2} \cdot \frac{\eta_{i,\mathrm{driver}.test2}}{\eta_{i,\mathrm{driver}.test1}} = 71.9\%$

Acceptance tolerances for actual tested values of the two speed test

- Energy density
- $\left(\frac{e_{co.g.test1.2}}{e_{pac.g}} - 1\right) \cdot 100 = -0.56\ \%$

Test results summary

Turbo Compressor	Symbol	Unit (Metric)	Ref Section	Numerical Values		
TEST DATA						
Test Number	#	number		1g	***1te***	2_{te}
Test Period start/end	time	Min.		-	***10***	
Barometric Pressure	$p_{.amb.}$	psia		14.7	14.47	14.47
Inlet Pressure	$p_{1.abs}$	psia	4.4.14	14.7	14.47	14.47
Inlet Temperature	T_1	°R	4.4.14	563.7	482.7	482.7
Relative Humidity	$\varphi_{rel,1}$	%	4.4.1	60.0	60	60
Isentropic exponent	κ_{wet}	ratio	4.4.4	1.396	***1.4***	1.4
Gas Constant	R_{wet}	ft-lb/lb°R)	4.4.4	54.232	53.390	53.390
Inlet density	ρ_1	lb/ft^3		0.0692	0.08085	0.08085
Outlet Pressure	$p_{2.abs}$	psia		20.3	21.17	20.0
Discharge Temperature	T_2	°R		-	563.43-	541.29
Compressor Speed [rpm]	n	rev/min		29750	***29750***	27315
Supply voltage	U	V		460	***460***	460
Supply frequency	f	Hz		60	60	60
Package Input Power	P_{PAC}	kW		19.76	20.0	18.03
External Coolant Inlet Temperature	$T_{1,cool}$	°R		-	-	
External Coolant Flow	$q_{m,cool}$	lb/s		-	-	
Inlet Volume Flow Rate tested	q_{v1}	ft^3/min	4.4.7	875	654.5	804.25
Isentropic compression work	y_s	ft-lb/lb	4.4.5	10333	10359	8739
Package isentropic efficiency	η	%	4.4.8	71.6%	61.9%	71.1%
				Test 1,2 results conrrected to pre-conditions	***Pass/Fail***	
Inlet Volume Flow Rate converted	q_{v1}	ft^3/min	8.2	875.9	Pass	
Package isentropic efficiency	η	%		71.9%		
Outlet Pressure	p_2	psia	8.3.3	20.32	Pass	
Package Specific Energy	e_{co}	kW/100ft^3/min	8.4	2.246	Pass	

Appendix C – Informative

Specific Test Point Data Collection
The compressor package shall operate at the steady state condition for the duration of data collection for each test point. The compressor (blower) shall be operated at the required conditions for a sufficient period of time to reach steady state each test point.

Steady-state is defined as demonstrating the difference between inlet and outlet temperatures $\Delta T = (T_2 - T_1)$ is within the limit Section 7.9, Table 3 for a period of three minutes interval or more.

A minimum of three sets of data shall be collected. The minimum duration of a test point, after steady state has been reached, shall be 10 minutes from the start of the first set of readings to the end of the third set of readings.

A test point considers one complete set of instrument readings obtained in a one-minute period. The individual readings are summed and divided by the total number of readings to establish an average, and used for the test point.

Recorded data of the test point shall be included in the test report to demonstrate the thermal and fluid stability at time of measurement.

Date and time of data collection shall be reported for each set of data.

The use of data acquisition systems shall be allowed and the test logs may be print outs resulting from the system.

Flow measurement
The delivered flow rate is the net mass flow rate through the process connection of the compressor package outlet. All seal losses and side streams not delivered to the process piping connection of the compressor package shall be excluded from the delivered mass flow rate evaluation.

Flow shall be measured on the process side of the compressor package outlet. It can be measured according to the principle and requirements of ISO 5167.

The mass flow rate is determined from the gas condition of the fluid at a flow meter measurement station. The general equation for mass flow is found in ISO 5167-1.

In cases of high temperature or dissimilar materials, the thermal effects of diametrical changes of the fluid meter and pipe may not be negligible in the determination of the ratio of diameters.

Measuring lines installed between the sampling point and the display instrument shall be installed with great care. Any leaks shall be eliminated. Provisions shall be made to prevent blockage by foreign bodies. Where condensate occurs in the measuring lines, such lines shall be completely filled with condensate or shall be reliably kept free of condensate (e.g. by arranging the measuring instrument at a geodetic higher level than the measuring point).

Electrical power measurement
Measurement of the total wire power supplied to the compressor package shall be recorded at test conditions with the compressor operating at the specified operating point.

Power measurement for non-inverter applications
For non-inverter applications, a standard power analyser based on 50/60Hz RMS sine wave is acceptable.

Power measurement for inverter applications
For inverter applications, the wire power measuring instrument shall be capable of handling the distorted voltage and current waveforms and phase relationship of the power factor caused by the harmonics and EMI as a result an inverters high-speed switching mode. Wire power shall be measured by a precision power analyzer with high accuracy, broad bandwidth, fast sampling rate and high-speed data update.

The frequency bandwidth shall cover 0,1 Hz to 1 MHz. The sampling rate shall be approximately 200 kS/s or greater. The maximum data updating period shall be 50 ms.

The precision power analyzer shall be capable of simultaneous measurement of normal and harmonic waves. It shall provide a variety of display formats for viewing waveforms as well as numerical values. It shall also combine the use of digital filtering and total-average methods for sampling instantaneous values.

The precision power analyzer shall have compensation functions for instrument-related losses. It shall also have a variety of integration functions for active power, current, apparent power and reactive power.

The current transducer shall be capable of measuring a dynamic range and peak greater than the maximum current consumption of the package. The measurement frequency range shall be from DC to 100 kHz or a minimum of eight times greater than the switching mode of the inverter driving the package.

Temperature measurement
A minimum of two temperature measuring instruments shall be utilized for each measurement location spaced at 180° intervals around the pipe circumference. The temperature measuring devices shall have the required resolution for accuracy.

The thermometers or the pockets shall extend into the pipe to a distance of 100 mm, or one third the diameter of the pipe, whichever is less.

Temperature measurement ambient
The compressor package ambient temperature is the atmospheric temperature measured at the compressor package in the plane of the intake system.

Temperature measurement inlet
The compressor inlet temperature is the total temperature (T_1) measured at the compressor inlet [1]. The temperature instrumentation shall be located ½ pipe diameter upstream of the compressor inlet. The location shall include all heating or cooling affecting the process fluid prior to the compressor inlet. If inadequate space is available within the package, the measurement location shall be as agreed to in advance of test by all parties including equipment owner, owner's engineer, and supplier.

Temperature measurement outlet
The compressor outlet temperature is the total temperature (T_2) measured at the compressor outlet [2]. The temperature instrumentation shall be located 1 pipe diameter downstream of the compressor outlet and 90° relatively rotated to the pressure measurement.

Pressure measurement
A minimum of two static or total pressure measuring instruments shall be utilized for each measurement location spaced at 180° intervals around the pipe circumference, and 90° to temperature instrumentation. The pressure measuring devices shall have the required resolution for accuracy.

Pressure measurement for ambient inlet
The compressor package inlet pressure p_1 is the atmospheric pressure measured by a barometer in the vicinity of the compressor package where the velocity is zero.

Pressure measurement for pipe inlet
The compressor inlet pressure is the total pressure (p_1) measured at the compressor inlet.

The compressor inlet pressure shall be measured at a location at least 1pipe diameter upstream of the compressor inlet or at a location to include all intake pressure losses impacting the process fluid prior to the compressor inlet. If inadequate space is available within the package, the measurement location shall be as agreed to in advance of test by all parties including equipment owner, owner's engineer, and supplier. The pressure measurement shall be a total measurement, or static measurement corrected to total conditions.

Pressure measurement outlet
The compressor package outlet pressure is the pressure (p_2) measured in the outlet piping at a prescribed location following the compressor package design consideration and code requirements to ensure a stable and accurate reading.

The pressure instrumentation shall be located 2 pipe diameters downstream of the compressor outlet.

Speed measurement
Where measurement of the speed of rotation is necessary for the performance test, it shall be determined with the accuracy necessary for this purpose using a cyclometer, tachometer, frequency meter; etc.

Appendix D – Thermodynamic Background (Informative)

D.1.1 Characteristic curves of dynamic compressors and positive displacement compressors

From observations it is known that at a constant speed, the fundamental "curves" (head or work or pressure rise versus volume flow or capacity) of the two technologies are different.

The positive displacement compressor (blower) will deliver nearly the same capacity regardless of pressure change. The small volume flow changes are capacity losses from internal leakages. Compressor rotational speed also has an influence on the flow delivered and power needed.

A dynamic compressor (blower) capacity changes more as the pressure changes. On rising pressure, the capacity reduces appreciably. The speed of the impeller in a dynamic compressor (blower) has a marked effect on the flow and head performance of the compressor. It is for this reason that data sheet provides the defined speed of the impeller.

Packages may be of a gear, belt, or direct drive design. To ensure all losses of the drive system are incorporated, speed of the main driver is reported, rather than the compression or compressor element.

D.1.2 Performance test through representative efficiency measurement

The intention of the performance test of a low-pressure compressor is to verify the delivered compressed air flow and the power or energy consumption at a required outlet pressure. With this test the manufacturer guarantees the performance. This performance is to be verified at a particular ambient condition, which is the guarantee or specified condition. In principle the test should be at this guarantee condition. However, in practice this cannot be achieved. The actual ambient test condition defined by the ambient pressure, temperature and humidity, will deviate from the guarantee condition due to the weather.

To convert the performance from the actual test condition to the guarantee or reference condition the test uses the efficiency of the compressor. The efficiency of the compressor (blower) is the ratio of the ideal power consumption to the real power consumption of the compressor (blower). The ideal power consumption of the compressor (blower) is the power consumption following a known thermodynamic reference process appropriate for the type of compressor (blower). The compressor (blower) in test is operated at a similarity point such that the measurement of the efficiency during this test is representative for the efficiency at the guarantee condition. Subsequently with the known thermodynamic reference process and thus the ideal power consumption, the real performance at the guarantee or reference condition can be calculated easily.

D.1.3 Low-pressure compressor (blower) in a similarity point

From the non-dimensional analysis of a compressor (blower) we know that the compressor (blower) can be set during test in a similar operating point as in the guarantee or specified condition. In this similarity point all the non-dimensional groups or numbers are identical. With the Buckingham theorem it is possible to identify 5 independent non-dimensional groups for compression of a dry, ideal gas:

- The heat capacity ratio k = cp/cv (gas property),
- The Reynolds number,
- The machine work coefficient,
- The machine Mach number.
- The machine geometry (for variable geometry compressors)

If these 5 non-dimensional groups are identical for the test condition and the guarantee condition, then all other non-dimensional groups are identical as well: Efficiency, flow coefficient, etc. … With the known measured efficiency during the test the performance at the similar operating point, the guarantee condition can be calculated.

In practice it is not possible to have a perfect match of the 5 independent non-dimensional groups. The test is to be performed in such a way to keep the non-dimensional groups in an acceptable range of values to ensure the efficiency derived from the test measurements is a representative value.

To achieve close similarity, the initial provisions for the test are:

- The test is done with ambient air. Thus we consider the deviation of the k value to be small (influenced by humidity) between test condition and guarantee condition.
- The ambient temperature is within a limited range.
- Tests are done on the actual compressor, not a scaled version.
- The mechanical speed during test is set within a limited range of the guarantee condition.
- For variable geometry compressors, the geometry is set so as to match the intended flow coefficient.

These initial provisions limit changes in the Reynolds number and Mach number. In essence with these provisions we limit the changes of the compressor efficiency due to the compressibility of the gas (Mach number) and the viscosity losses (Reynolds number).

The major, final provision is the setting of the machine work coefficient. The work coefficient is controlled by imposing the outlet pressure to the compressor.

D.1.4 Reference process for low pressure compressors

The work coefficient is the non-dimensional ratio of the work of the gas to the mechanical, kinetic energy of the compressor (blower). There are however different definitions of the work that can be used. Usually an ideal work value according to a representative, ideal reference process is chosen for the definition of the work coefficient. For instance, the isentropic or isochoric work. For low pressure dynamic compressors (blower) the work coefficient defined by the isentropic reference process is most appropriate. This is already well known and common practice for dynamic compressors [ref. ISO 5389].

For the positive displacement compressor (blower) with or without internal compression it is known that in first order the rotational speed of the compressor (blower) does not influence the work added to the gas. Also the speed difference between test and guarantee is already limited. Therefore, the matching of an ideal work value is sufficient (rather than matching a work coefficient).

For the positive displacement compressor (blower) with or without internal compression a proper combination of isentropic and isochoric work can be defined. For the positive displacement compressor (blower) with internal compression the work (W_s) on the input volume at closure of the inlet port (V_1) is isentropic up to an intermediate pressure p_i.

$$W_s = p_1 V_1 \frac{\kappa}{\kappa - 1}\left(\left(\frac{p_i}{p_1}\right)^{\frac{\kappa-1}{\kappa}} - 1\right)$$

For isentropic compression of an ideal gas there is a relation between the intermediate pressure p_i and the internal volume ratio $v_i = {}^{V_1}\!/_{V_i}$

$$\frac{p_i}{p_1} = \left(\frac{V_i}{V_1}\right)^{-k} = {v_i}^k$$

For the positive displacement compressor (blower) without internal compression the internal volume ratio and pressure ration is 1 and thus the isentropic compression work is zero.

In a second phase the compression continues isochoric (W_{isoc}) on the reduced volume as defined by the internal volume ratio.

$$W_{isoc} = (p_2 - p_1)V_i$$

The sum of the work is (W_{comb}):

$$W_{comb} = \left[p_1 V_1 \frac{\kappa}{\kappa - 1}\left(\left(\frac{p_i}{p_1}\right)^{\frac{\kappa-1}{\kappa}} - 1\right)\right] + [(p_2 - p_1)V_i]$$

$$W_{comb} = p_1 V_1 \left[\frac{\kappa}{\kappa - 1}\left(\left(\frac{p_i}{p_1}\right)^{\frac{\kappa-1}{\kappa}} - 1\right) + \left(\frac{p_2 - p_i}{p_1}\right)\frac{V_i}{V_1}\right]$$

$$W_{comb} = p_1 V_1 \left[\frac{\kappa}{\kappa - 1}\left(\left(\frac{p_i}{p_1}\right)^{\frac{\kappa-1}{\kappa}} - 1\right) + \frac{{}^{p_2}\!/_{p_1}}{v_i} - \frac{p_i}{p_1}\frac{1}{v_i}\right]$$

$$W_{comb} = p_1 V_1 \left[\frac{p_2/p_1}{v_i} + \frac{\kappa}{\kappa - 1} \left(\left(\frac{p_i}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 - \frac{\kappa - 1}{\kappa} \frac{p_i}{p_1} \frac{1}{v_i} \right) \right]$$

$$W_{comb} = p_1 V_1 \left[\frac{p_2/p_1}{v_i} + \frac{\kappa}{\kappa - 1} \left({v_i}^{\kappa-1} - 1 - \frac{\kappa - 1}{\kappa} {v_i}^{\kappa-1} \right) \right]$$

$$W_{comb} = p_1 V_1 \left[\frac{p_2/p_1}{v_i} + \frac{\kappa}{\kappa - 1} \left(\frac{1}{\kappa} {v_i}^{\kappa-1} - 1 \right) \right]$$

The corresponding specific work (work per unit mass) is:

$$y_{comb} = RT_1 \left[\frac{p_2/p_1}{v_i} + \frac{\kappa}{\kappa - 1} \left(\frac{1}{\kappa} {v_i}^{\kappa-1} - 1 \right) \right]$$

Initial, rough analysis of the predicted performance changes of positive displacement compressors (blowers) due to changes of inlet temperature, pressure etc. reveal the combined isentropic and isochoric work is a good representation of the ideal process. More detailed experimental analysis and comparison to other work definitions or work coefficient definitions may be valuable.

D.1.5 Isentropic or combined work for the defined low-pressure compressor (blower) package

Note that the isentropic or combined isentropic and isochoric work is based on the pressures at package inlet and outlet. The package includes losses of inlet filter, check valve and other components if they are part of the package. The pressure and temperature variation through the package can be illustrated as in the figure below.

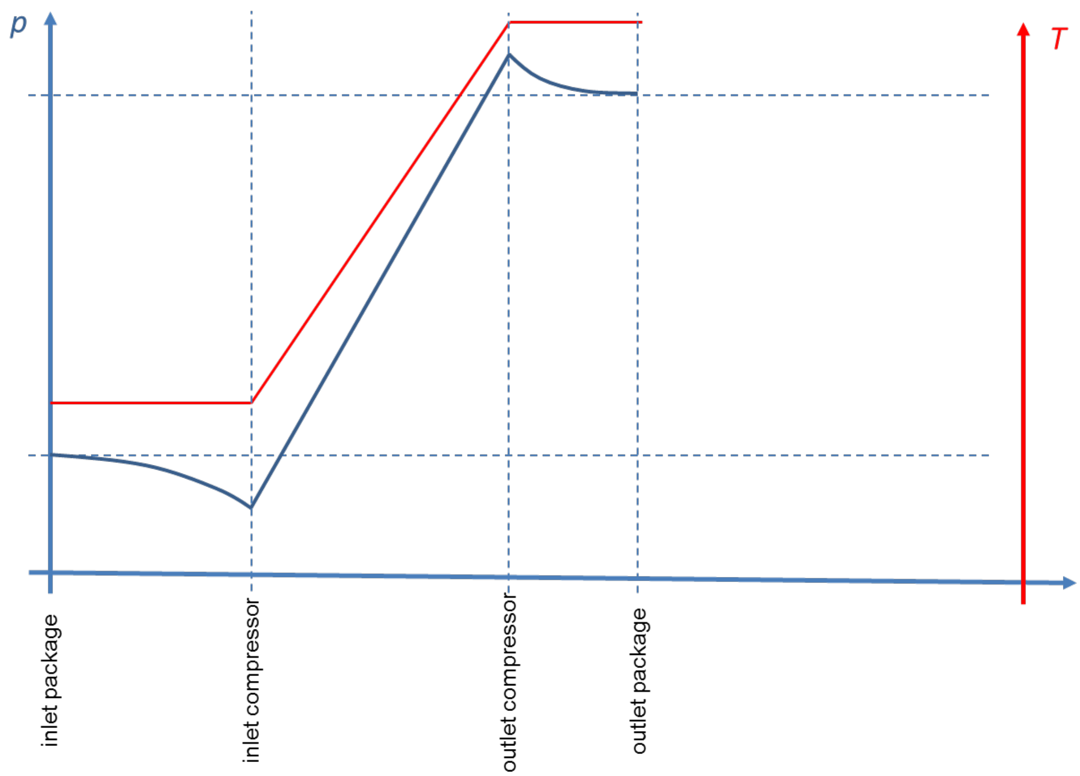


Figure 5: combined work for defined package

The pressure losses result in lowered isentropic or combined isentropic and isochoric work values based on the inlet and outlet pressures. The isentropic or combined isentropic and isochoric work is lower than the real work transferred to the gas. The real work can be determined with the temperatures of the gas, but is not used in this methodology. The package power is derived from the measured electric input power and corrected for ratios between guarantee and test conditions.

D.1.6 Correction formulas

The flow correction is based on the flow coefficient that is indifferent between test and guarantee conditions. From this equation the tested flow is corrected to the guarantee condition. This is the flow that this specific compressor will deliver under the guarantee condition (this is with another rotor tip speed).

$$\varphi_{te} = \varphi_g$$

$$\frac{q_{v1,te}}{u_{te}} = \frac{q_{v1,co}}{u_g}$$

$$q_{v1,co} = q_{v1,te} \cdot \frac{u_g}{u_{te}}$$

The efficiency η is the ratio of the ideal power P_{ideal} to the real power consumption P. The ideal power is according to the appropriate thermodynamic reference process. This is the isentropic process for dynamic compressors and the combined isentropic and isochoric process for the positive displacement compressors.

$$\eta = \frac{P_{ideal}}{P}$$

The correction for power and specific power is based on this efficiency that is identical between test and guarantee conditions as the test is in a similarity point of the guarantee condition.

$$\eta_{te} = \eta_g$$

Two cases are distinguished under guarantee conditions. In a general case, under guarantee conditions the package delivers the corrected flow $q_{v1,co}$ (as derived from the flow correction based on the flow coefficient) and will consume the corrected power P_{co}. In the specific case that the package matches the guarantee flow exactly (e.g. with a variable flow compressor), then the package delivers the guarantee flow $q_{v1,g}$ and will consume the corrected power at guarantee flow $P_{co,g}$. In both cases the efficiency matches the tested efficiency:

$$\frac{P_{ideal,te}}{P_{te}} = \frac{P_{ideal,co}}{P_{co}} = \frac{P_{ideal,g}}{P_{co,g}}$$

From this the formulas for the power corrections are:

$$P_{co} = \frac{P_{ideal,co}}{P_{ideal,te}} \cdot P_{te}$$

$$P_{co,g} = \frac{P_{ideal,g}}{P_{ideal,te}} \cdot P_{te}$$

The specific energy is the ratio of the power to the flow

$$e_{te} = \frac{P_{te}}{q_{v1,te}}$$

$$e_{co} = \frac{P_{co}}{q_{v1,co}}$$

$$e_{co,g} = \frac{P_{co,g}}{q_{v1,g}}$$

The specific energy can be written as a function of the tested power and flows:

$$e_{co} = \frac{P_{co}}{q_{v1,co}} = \frac{P_{ideal,co}}{P_{ideal,te}} \cdot P_{te} \cdot \frac{1}{q_{v1,co}}$$

$$e_{co,g} = \frac{P_{co,g}}{q_{v1,g}} = \frac{P_{ideal,g}}{P_{ideal,te}} \cdot P_{te} \cdot \frac{1}{q_{v1,g}}$$

Also the ideal powers can be written as the factor of mass flow and work. The mass flow in turn is the factor of density and volume flow. These terms are known and are used to make the correction of the specific energy from test to guarantee condition.

$$e_{co} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{q_{v1,co}}{q_{v1,te}} \cdot \frac{y_{ideal,g}}{y_{ideal,te}} \cdot P_{te} \cdot \frac{1}{q_{v1,co}}$$

$$e_{co,g} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{q_{v1,g}}{q_{v1,te}} \cdot \frac{y_{ideal,g}}{y_{ideal,te}} \cdot P_{te} \cdot \frac{1}{q_{v1,g}}$$

$$e_{co,g} = e_{co} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{ideal,g}}{y_{ideal,te}} \cdot e_{te}$$

Using the specific energy, the power consumption can be calculated as follows:

$$P_{co} = e_{co} \cdot q_{v1,co}$$

$$P_{co,g} = e_{co} \cdot q_{v1,g}$$

D.1.7 Isentropic efficiency of a compressor package

The isentropic efficiency of a compressor (blower) package can be used to characterize its energy efficiency. This is an alternative to the use of the specific energy requirement e which is an indicator for energy requirement for a given inlet and outlet pressure. Typically, the isentropic efficiency will change less to changes in inlet and outlet pressure compared to the specific energy requirement.

The isentropic efficiency of a package is defined as the inverse ratio of the corrected input power requirement of the compressor (blower) package versus the isentropic power needed to compress the same volume flow at same inlet and outlet conditions of the compressor package.

$$\eta_s = \frac{P_s}{P_{co}}$$

The isentropic power is defined as:

$$P_s = q_{v1,co} \cdot p_1 \frac{\kappa}{(\kappa - 1)} \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right]$$

With the definition of the specific energy requirement

$$e_{co} = \frac{P_{co}}{q_{v1,co}}$$

we have a direct relationship between specific energy requirement and isentropic efficiency of a compressor package:

$$\eta_s = \frac{p_1 \frac{\kappa}{(\kappa-1)} \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right]}{e_{co}}$$

$$e_{co} = \frac{p_1 \frac{\kappa}{(\kappa-1)} \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right]}{\eta_s}$$

Appendix E – Data Sheets

This section offers the reader examples of standardized data sheets that allow the customer to compare performance of same or different compressor types.

The key takeaway on the data sheet is "Specific Energy”. This value (expressed in (kW/m^3/min_ or kW/(100cfm) is the measure of compressor (blower) package efficiency. The lower the value, the more efficient the package is. This is a quick and easy way to see which blower uses less power at the stated conditions.

AIR BLOWER PACKAGE DATA SHEET

CENTRIFUGAL AIR BLOWER PACKAGE DATA SHEET

MODEL DATA - Standard Conditions - US Customary Units

1	Manufacturer:	ABC, Inc.	Date:	07/16/15
2	Model Number:	Blower Package		

3

- ☑ Main Drive Motor
- ☐ Inlet Throttle Valve
- ☑ Harmonic Filter
- ☐ No Negative Tolerance Data
- ☑ Driver Cooling
- ☐ Lubrication System
- ☐ Inlet Guide Vanes
- ☑ VFD
- ☐ Gearbox / Belt
- ☑ Inlet Air Filter

		VALUE	UNITS
4	Rated Capacity (FAD) at Rated Operating Pressure	2597	cfm
5	Rated Operating Pressure - p2	14	psig
6	Drive Motor Nameplate Rating	150.0	hp
7	Blower Rated Speed	28460	rpm

8 Performance Table[a]

Discharge Pressure p2 (psig)[b]		Delivered Air Flow - FAD (cfm)				
		100% FAD	FAD 2[c]	FAD 3[c]	FAD 4[c]	MIN FAD[d]
14 psig	FAD[f]	2597	2415	2232	2050	1867
	Spec. Power[e]	4.41	4.40	4.41	4.44	4.51
	rpm	28460	28080	27748	27454	27208
12 psig	FAD[f]	3034	2701	2368	2035	1702
	Spec. Power[e]	3.96	3.85	3.78	3.77	3.86
	rpm	28474	27401	26509	25819	25310
10 psig	FAD[f]	3256	2822	2388	1954	1520
	Spec. Power[e]	3.65	3.35	3.17	3.12	3.26
	rpm	28486	26492	24937	23861	23162
8 psig	FAD[f]	3389	2870	2352	1833	1315
	Spec. Power[e]	3.35	2.87	2.56	2.51	2.69
	rpm	28496	25492	23076	21598	20671

Notes:

a. Based on reference inlet conditions of pamb=14.7 psia, Tamb=68°F, RH=36%

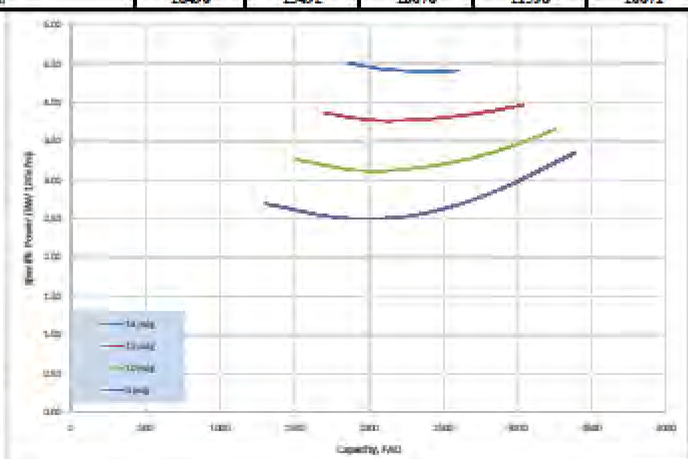
b. Discharge pressure in -2 psig increments starting at max, rated operating pressure. To include 8.0 psig

c. Intermediate points nominal equal spacing between 100% and Min. Flow (see note d.)

d. Lowest Turned Down FAD

e. Specific power (kW /100 cfm) tolerance of +/- tolerance given by Table 2 in BL 300 unless "No Negative Tolerance" box is checked.

f. Delivered air flow +/- tolerance given by Table 2 in BL 300 unless "No Negative Tolerance" box is checked



Member

BL 072
07/16 R5

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AIR BLOWER PACKAGE DATA SHEET

CENTRIFUGAL AIR BLOWER PACKAGE DATA SHEET

MODEL DATA - Standard Conditions - (SI Units)

1	Manufacturer:	ABC, Inc.	Date:	07/16/15
2	Model Number:	Blower Model		

3

- ☑ Main Drive Motor
- ☑ Driver Cooling System
- ☑ VFD
- ☐ Inlet Throttle Valve
- ☐ Lubrication System
- ☐ Gearbox / Belt Drive
- ☑ Harmonic Filter
- ☐ Inlet Guide Vanes
- ☑ Inlet Air Filter
- ☐ No Negative Tolerance Data

		VALUE	UNITS
4	Rated Capacity (FAD) at Rated Operating Pressure		m^3/min
5	Rated Operating Pressure - p_2		mbar(g)
6	Drive Motor Nameplate Rating		kW
7	Blower Rated Speed		rpm

8 Performance Table[a]

Discharge Pressure p2 (mbar)[b]		Delivered Air Flow - FAD (m^3/min) 100% FAD	FAD2[c]	FAD3[c]	FAD4[c]	MIN FAD[d]
1000 mbar(g)	FAD[f]	841	675	634	450	322
	Spec. Power[e]	5.13	5.27	5.32	5.69	6.17
	rpm	5375	4455	4225	3219	2500
900 mbar(g)	FAD[f]	1067	870	777	511	324
	Spec. Power[e]	4.74	4.78	4.81	5.11	5.66
	rpm	6725	5575	5035	3556	2500
800 mbar(g)	FAD[f]	1068	871	779	514	328
	Spec. Power[e]	4.38	4.40	4.41	4.65	5.11
	rpm	6725	5575	5035	3556	2500
700 mbar(g)	FAD[f]	1200	984	863	554	334
	Spec. Power[e]	3.79	3.75	3.72	3.83	4.18
	rpm	7535	6245	5521	3759	2500

Notes:

a. Based on reference inlet conditions of pamb=1 bar(a), Tamb=20°C, RH=0%

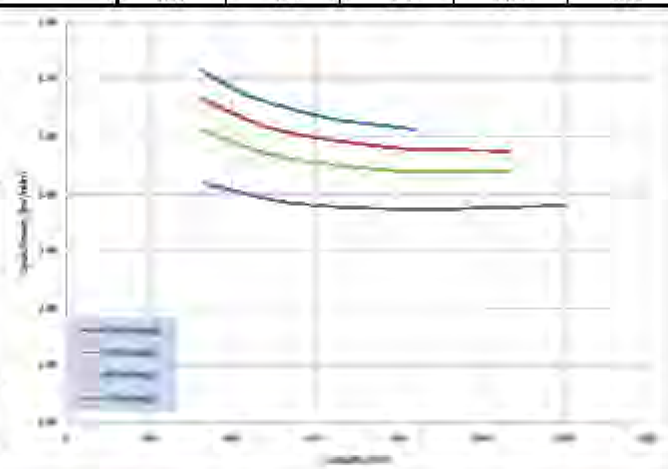
b. Discharge pressure at 1000 mbar(g) increments starting at max. rated operating pressure. To include 800 mbar

c. Intermediate points at equal spacing between 100% and Min. Flow (see note d)

d. Lowest Turned Down FAD

e. Specific power (kW/100 cfm) tolerance of +/- tolerance given by Table 2 in BL 300 unless "No Negative Tolerance" box is checked

f. Delivered air flow +/- tolerance given by Table 2 in BL 300 unless "No Negative Tolerance" box is checked



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AIR BLOWER PACKAGE DATA SHEET

Positive Displacement Fixed Speed Blower

MODEL DATA - Standard Conditions - US Customary Units				
1	Manufacturer:	ABC, Inc.	Date:	7/16/2015
2	Model Number:	Blower Model		
3	☑ Main Drive Motor ☐ Driver Cooling System ☐ Lubrication System ☑ Starters ☑ Inlet Air Filter ☑ Gearbox / Belt Drive ☑ Control Cubicle ☑ Check Valve			
			VALUE	UNITS
4	Rated Capacity (FAD) at Rated Operating Pressure		200	cfm
5	Rated Operating Pressure - p_2		12.00	psig
6	Drive Motor Nameplate Rating		20.0	hp
7	Blower Rated Speed		3000	rpm
8	Performance Table [a]			
	Discharge Pressure [c]		VALUE	UNITS
	10 psig	Delivered Air Flow - FAD [d]	212	cfm
		Specific Power [b]	5.20	kW / 100 cfm
		Blower Speed	3020	rpm
	8 psig	Delivered Air Flow - FAD [d]	250	cfm
		Specific Power [b]	6.40	kW / 100 cfm
		Blower Speed	3100	rpm
	6 psig	Delivered Air Flow - FAD [d]	270	cfm
		Specific Power [b]	7.30	kW / 100 cfm
		Blower Speed	3150	rpm

Notes:

a. Based on reference inlet conditions of pamb= 14.7 psia, Tamb = 68°F, RH=36%.

b. Specific power (kW / 100cfm.) +/- tolerance given by Table 2 in BL 300.

c. Zero negative tolerance for discharge pressure. An 8 psig data point is required.

d. Delivered air flow +/- tolerance given by Table 2 in BL 300.

Member

BL 062 (US Units)
07/16 R5

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AIR BLOWER PACKAGE DATA SHEET

Positive Displacement Fixed Speed Blower

MODEL DATA - Standard Conditions (SI Units)				
1	Manufacturer:	ABC, Inc.	Date:	7/16/2015
2	Model Number:	Blower Model		
3	☑ Main Drive Motor ☐ Driver Cooling System ☐ Lubrication System ☑ Starters ☑ Inlet Air Filter ☑ Gearbox / Belt Drive ☑ Control Cubicle		VALUE	UNITS
4	Rated Capacity (FAD) at Rated Operating Pressure		36	m^3 / min
5	Rated Operating Pressure - p_2		800	mbar(g)
6	Drive Motor Nameplate Rating		37.0	kW
7	Blower Rated Speed		3000	rpm
8	Performance Table [a]			
	Discharge Pressure[c]		VALUE	UNITS
	700 mbar(g)	Delivered Air Flow - FAD[d]	36.4	m^3 / min
		Specific Power[b]	1.53	kW / m^3/min
		Blower Speed	3020	rpm
	600 mbar(g)	Delivered Air Flow - FAD[d]	36.7	m^3 / min
		Specific Power[b]	1.50	kW /m^3/min
		Blower Speed	3100	rpm
	400 mbar(g)	Delivered Air Flow - FAD[d]	37	m^3 / min
		Specific Power[b]	1.40	kW /m^3/min
		Blower Speed	3150	rpm

Notes:

a. Based on reference inlet conditions of pamb = 1 bar (a), Tamb=20°C, RH=0%.

b. Specific power (kW / m^3/min) +/- tolerance given by Table 2 in BL 300.

c. Zero negative tolerance for discharge pressure. A 600 mbar(g) data point is required.

d. Delivered air flow +/- tolerance given by Table 2 in BL 300.

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BL 062
07/16 R5

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AIR BLOWER PACKAGE DATA SHEET

Positive Displacement Variable Speed Blower

MODEL DATA - Standard Conditions (US Units)

1	Manufacturer:	ABC, Inc		Date:	02/28/20	
2	Model Number:	Blower Model				
3	☑ Main Drive Motor ☐ Driver Cooling System ☐ Harmonic Filter ☐ No Negative Tolerance Data	☑ Control Cubicle ☐ Lubrication System ☑ Discharge Check Valve		☑ VFD ☑ Gearbox / Belt Drive ☑ Inlet Air Filter		
					VALUE	**UNITS**
4	Rated Capacity (FAD) at Rated Operating Pressure				800	cfm
5	Rated Operating Pressure - p_2				15.00	psig
6	Drive Motor Nameplate Rating				100.0	hp
7	Blower Rated Speed				5375	rpm

			Performance Table[a]				
	Discharge Pressure p2 (psig)[b]		Delivered Air Flow - FAD (cfm)				
			100% FAD	FAD[c]	FAD[c]	FAD[c]	MIN FAD[d]
8	14 psig	FAD[f]	841	675	634	450	322
		Spec. Power[e]	5.13	5.27	5.32	5.69	6.17
		Blower Speed (rpm)	5375	4455	4225	3219	2500
	12 psig	FAD[f]	1067	870	777	511	324
		Spec. Power[e]	4.74	4.78	4.81	5.11	5.66
		Blower Speed (rpm)	6725	5575	5035	3556	2500
	10 psig	FAD[f]	1068	871	779	514	328
		Spec. Power[e]	4.38	4.40	4.41	4.65	5.11
		Blower Speed (rpm)	6725	5575	5035	3556	2500
	8 psig	FAD[f]	1200	984	863	554	334
		Spec. Power[e]	3.79	3.75	3.72	3.83	4.18
		Blower Speed (rpm)	7535	6245	5521	3759	2500

Notes:

a. Based on reference inlet conditions of pamb=14.7 psia, Tamb=68°F, RH=36%

b. Discharge pressure in -2 psig increments starting at max. rated operating pressure. To include 8 psig

c. Intermediate points at equal spacing between 100% and Min. Flow (see note d.)

d. Lowest Turned Down FAD

e. Specific power (kW /100 cfm) tolerance of +/- tolerance given by Table 2 in BL 300 unless "No Negative Tolerance" box is checked

f. Delivered air flow +/- tolerance given by Table 2 in BL 300 unless "No Negative Tolerance" box is checked

g. dashed lines indicate not within label tolerance

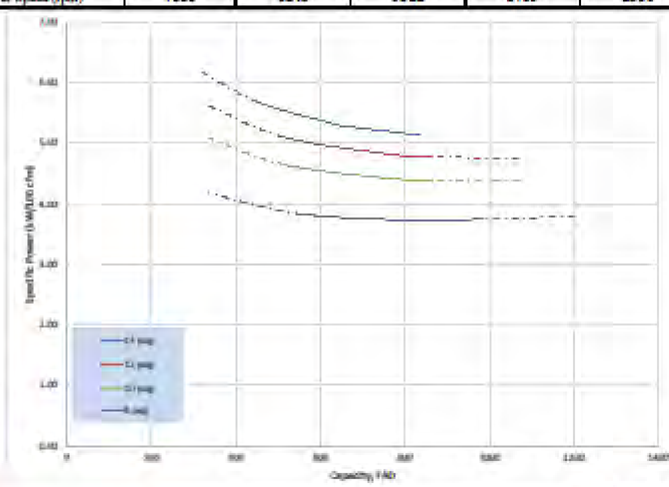

Table 2 from BL 300:	Delivered Air Flow at specified conditions	[illegible]	Specific Power Consumption	Discharge Pressure
m^3/min	ft^3/min	%	%	%
Below 0.5	Below 15	+/- 7	+/- 8	-0 / +12
0.5 to 1.5	15 to 50	+/- 6	+/- 7	-0 / +12
1.5 to 15	50 to 500	+/- 5	+/- 5	-0 / +1
Above 15	Above 500	+/- 4	+/- 5	-0 / +1

CAGI

BL 082 (US Units)
07/16 R3

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AIR BLOWER PACKAGE DATA SHEET

Positive Displacement Variable Speed Blower

MODEL DATA - Standard Conditions - SI Units

1	Manufacturer:	ABC, Inc.	Date:	07/16/15
2	Model Number:	Blower Model		

3
- ☑ Main Drive Motor
- ☑ Control Cubicle
- ☑ VFD
- ☐ Driver Cooling System
- ☐ Lubrication System
- ☑ Gearbox / Belt Drive
- ☑ Harmonic Filter
- ☑ Discharge Check Valve
- ☑ Inlet Air Filter
- ☐ No Negative Tolerance Data

		VALUE	UNITS
4	Rated Capacity (FAD) at Rated Operating Pressure	24.18	m³ / min
5	Rated Operating Pressure - p_1	600	mbar(g)
6	Drive Motor Nameplate Rating	30.0	kW
7	Blower Rated Speed	4455	rpm

8 Performance Table[a]

Discharge Pressure p2 (mbar(g))[b]		Delivered Air Flow - FAD (m³ / min) 100% FAD	FAD 2[c]	FAD 3[c]	FAD 4[c]	MIN FAD[d]
600 mbar(g)	FAD[f]	24.18	19.71	15.72	12.25	9.84
	Spec. Power[e]	1.18	1.21	1.24	1.30	1.36
	Blower Speed (rpm)	5346	4455	3673	2989	2500
500 mbar(g)	FAD[f]	24.33	19.86	15.88	12.41	10.00
	Spec. Power[e]	1.02	1.03	1.06	1.10	1.15
	Blower Speed (rpm)	5346	4455	3673	2989	2500
400 mbar(g)	FAD[f]	24.48	20.01	16.03	12.57	10.16
	Spec. Power[e]	0.88	0.88	0.90	0.94	0.97
	Blower Speed (rpm)	5346	4455	3673	2989	2500
0 mbar(g)	FAD[f]	0.00	0.00	0.00	0.00	0.00
	Spec. Power[e]	0.00	0.00	0.00	0.00	0.00
	Blower Speed (rpm)	0	0	0	0	0

Notes:

a. Based on reference inlet conditions of pinlet=1 bar(a), Tinlet=20°C, RH=0% See CAGI BL 300 Standard

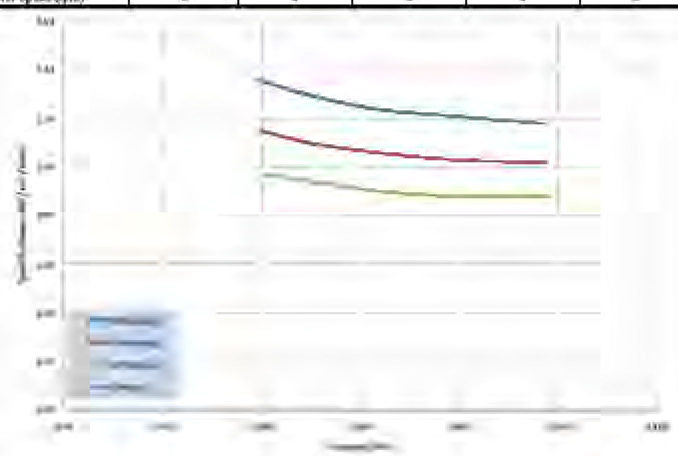
b. Discharge pressure in -100 mbar increments starting at max. rated operating pressure. To include 600 mbar(g)

c. Intermediate points at equal spacing between 100% and Min. Flow (see note d.)

d. Lowest Turned Down FAD

e. Specific power (kW /m³/min); tolerance of +/- tolerance given by Table 2 in BL 300 unless "No Negative Tolerance" box is checked

f. Delivered air flow +/- tolerance given by Table 2 in BL 300 unless "No Negative Tolerance" box is checked



Member CAGI

BL 300 (SI Units)

For example only. Double click in Word Document for Clearer PDF.