Chapter 22: Compressed Air Evaluation Protocol

The Uniform Methods Project: Methods for Determining Energy Efficiency Savings for Specific Measures

September 2011 – August 2020

This version supersedes the version originally published in January 2015 and subsequent 2017 publication. The content in this version has been updated.

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Disclaimer

Preface

This document was developed for the U.S. Department of Energy Uniform Methods Project (UMP). The UMP provides model protocols for determining energy and demand savings that result from specific energy-efficiency measures implemented through state and utility programs. In most cases, the protocols are based on a particular option identified by the International Performance Verification and Measurement Protocol; however, this work provides a more detailed approach to implementing that option. Each chapter is written by technical experts in collaboration with their peers, reviewed by industry experts, and subject to public review and comment. The protocols are updated on an as-needed basis.

The UMP protocols can be used by utilities, program administrators, public utility commissions, evaluators, and other stakeholders for both program planning and evaluation.

To learn more about the UMP, visit the website, https://energy.gov/eere/about-us/ump-home, or download the UMP introduction document at http://www.nrel.gov/docs/fy17osti/68557.pdf.

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Acronyms

ACFM	actual cubic feet per minute
CAGI	Compressed Air and Gas Institute
CCAF	Compressor Control Type Adjustment Factor
CFM	cubic feet per minute
ECM	electronically commutated motor
gal	gallons
hp	horsepower
IBV	inlet butterfly valve modulation
IGV	inlet guide vane modulation
kW	kilowatt
kWh	Kilowatt-hour
psi	pounds per square inch
psia	pounds per square inch absolute
psig	pounds per square inch gauge
RH	Relative humidity
RMS	root mean square
SCFM	standard cubic feet per minute
VSD	variable speed drive

Protocol Updates

The original version of this protocol was published in November 2014 and updated in October 2017.

This chapter has been updated from the 2017 version to incorporate the following revisions:

- Added equation 3 and 4 to calculate ACFM based on average operating pressure, site elevation, temperature, and humidity. Added guidance on effects of site conditions on compressor air flow.
- Updated interpolation calculation in Example 2.

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1 Measure Description

Compressed-air systems are used widely throughout industry for many operations, including pneumatic tools, packaging and automation equipment, conveyors, and other industrial process operations. Compressed-air systems are defined as a group of subsystems composed of air compressors, air treatment equipment, controls, piping, pneumatic tools, pneumatically powered machinery, and process applications using compressed air. A compressed-air system has three primary functional subsystems: supply, distribution, and demand.

Air compressors are the primary energy consumers in a compressed-air system and are the primary focus of this protocol.[1] The two compressed-air energy efficiency measures specifically addressed in this protocol are:

- High-efficiency/variable speed drive (VSD) compressor replacing modulating, load/unload, or constant-speed compressor
- Compressed-air leak survey and repairs.

This protocol provides direction on how to reliably verify savings from these two measures using a consistent approach for each.

1.1 High-Efficiency/Variable-Speed Drive Compressor Replacing Modulating Compressor

This measure pertains to the installation of a rotary screw compressor with a VSD. Most incentive programs and technical reference manuals use a baseline system definition of a standard modulating compressor with blowdown valve. The energy-efficient compressor is typically defined as an oil-flooded, rotary-screw compressor with variable-speed control.

This measure is frequently offered for the replacement of an existing unit at the end of its useful life or for the installation of a new system in a new building (i.e., time of sale).

Several control methods are available for air compressors, and control methods greatly affect the overall operating efficiency of a compressor. To accurately estimate energy savings, it is important to know the baseline method of control. A brief description of each common control method is provided below.

1.1.1 Reciprocating – On/Off Controls

The simplest method of control is to use an on/off control to start and stop a compressor to maintain system pressure. The compressor starts and generates air when the pressure falls below a certain set point, and it turns off when pressure is above a certain set point. Using an on/off control is an efficient way to ensure the compressor is either fully loaded or off; however, this form of control is only suitable for small compressors (typically less than approximately 5

[1] As discussed in "Considering Resource Constraints" in the introduction of this report, small utilities (as defined under the Small Business Administration regulations) may face additional constraints in undertaking this protocol. Therefore, alternative methodologies should be considered for such utilities. http://www.sba.gov/category/navigation-structure/contracting/contracting-officials/small-business-size-standards

horsepower [hp] and most common in residential settings). This method of control is uncommon in industrial settings.

1.1.2 Reciprocating – Load/Unload Control

Reciprocating compressors can be unloaded by holding open the inlet valve. Air is still pushed in and out of the compression chamber, but it is not compressed and discharged to the system. Depending on the number of cylinders and controls, the system may have multiple loading steps, such as 0%–50%–100% or 0%–25%–50%–75%–100%. Some compressors have a variable clearance volume, which impacts the amount of compressed air discharged at the end of the piston stroke and allows for additional capacity adjustment. Regardless of the specific type or steps of control, the standard performance curve shown in Table 2, later in this document, represents the energy usage.

1.1.3 Rotary-Screw – Inlet Valve Modulation/Inlet Throttling

Inlet valve modulation throttles off the air inlet to a compressor as discharge pressure rises above the set point pressure. The part-load performance of modulating compressors is relatively poor. Some modulation-controlled machines may be adjusted to fully unload if capacity reduces to a certain level, such as 40%. This reduces energy consumption compared to modulation-only compressors but requires the use of air storage receivers to meet demand when in the fully unloaded state.

1.1.4 Rotary-Screw – Load/Unload Control

Load/unload controls require significant storage receiver volume and operate a compressor at full capacity until the unload pressure (cutout) set point is reached. The compressor then unloads and blows down the oil separator and operates at minimum power while producing no air. Oil-free screw compressors nearly instantly unload due to no oil separator blowdown. The air loss associated with blowing down the oil separator is also eliminated.

1.1.5 Rotary-Screw – Variable-Displacement Control

Variable-displacement controls change compressor capacity by opening ports in the compressor that limit the amount of the cylinder or air-end that is used for compression. This can be implemented in either discrete steps (e.g., 50%, 75%, and 100%) or by continuously varying capacity. Compressor-specific power is typically good within the variable displacement range, but these compressors typically have a limited turndown range. At minimum turndown, the compressor commonly uses inlet modulation to further reduce flow, resulting in poor specific power, or kilowatt (kW) per cubic feet per minute (CFM).

1.1.6 Rotary-Screw – Variable-Speed Control

VSD or variable-frequency drive compressor controls use an integrated variable frequency alternating current or switched-reluctance direct current drive to control the electrical signal to the motor and, in turn, vary the speed of the motor and compressor. Compressors equipped with VSD controls continuously adjust the drive motor speed to match variable demand requirements. VSD compressors typically have an excellent turndown range and efficiently produce air over the entire range of operating speeds. Below the minimum turndown speed, the compressor typically cycles between off and minimum-load states. This method of control is typically the high-efficiency case and not the base case.

1.1.7 Centrifugal Controls

Most centrifugal compressors use a form of inlet throttling to vary capacity. Inlet butterfly valve and inlet guide vanes are both similar methods of control that reduce flow while also reducing power. Due to limitations in centrifugal compressor design, flow can only be reduced to a minimum level before surging occurs. To meet system flow below the throttling range, which is typically below approximately 70% of full load capacity, variable demands are met by the compressor by operating at the minimum throttled position and blowing off excess air produced through a blow-off valve. Therefore, most centrifugal compressors use a constant amount of power below the throttle limit regardless of actual demand. The standard curves shown in Table 2 are reflective of these common methods of control.

Another method of control used for centrifugal compressors is inlet throttling with unloading. Some centrifugal compressors can unload and recycle compressed air back to the compressor inlet instead of blowing off, and wasting, generated air. This control method can be more efficient but loading cycles do not allow for constant system pressure control.

A newer centrifugal compressor type uses a high-speed variable-speed rotor supported by magnetic bearings. The compressor varies speed to meet loads within the throttling range and unloads to a reduced speed instead of blowing off excess air. This type of control can be highly efficient, although it is not a compressor type commonly available at the time of this writing. It is important to note that variable-frequency drives cannot be retrofitted to existing fixed-speed centrifugal compressors; a special type of compressor is needed to utilize this advanced method of control.

For all centrifugal compressors, obtaining the actual performance curve is recommended as the performance of different compressor models varies significantly.

1.2 Compressed-Air Leak Survey and Repairs

Leaks are a significant cause of wasted energy in a compressed-air system and can develop in many parts of a compressed-air system. The most common problem areas are couplings; hoses; tubes; fittings, pipe joints, quick disconnects; filters, regulators, and lubricators; condensate traps; valves; flanges; packings; thread sealants; and other point-of-use devices.

Leakage rates are a function of the supply pressure, typically quantified in standard cubic feet per minute (SCFM), and proportional to the square of the orifice diameter (hole or crack size).

There are three common methods of compressed-air leak detection: auditory and sensatory observation, soapy water test, and ultrasonic leak detection. The industry standard and best practice is ultrasonic leak detection. This relies on the ability of specialized directional microphones and amplifiers to detect high-frequency noise generated by the turbulent flow of compressed air escaping a compressed-air system through an orifice or crack. The high-frequency sound produced by a compressed-air leak is both directional and localized to the source.

2 Application Conditions of Protocol

2.1 High-Efficiency/Variable-Speed Drive Compressor Replacement Measures

Demand-side management programs typically offer a prescriptive compressor replacement measure. Many programs and technical reference manuals assume the baseline compressor system to be a modulating, load/unload, or constant-speed compressor. New energy-efficient compressors are assumed to be VSD controlled.

Incentives for air compressor replacements are typically paid on a dollar-per-compressor-horsepower basis, dollar-per-kilowatt hour-saved basis, or a fixed percentage of project cost. Common eligibility requirements for compressor replacement measures include:

- The air compressor must be a primary system component and not a backup system component.
- Replaced equipment must be removed or the customer must attest that the baseline system, if remained connected, will be used only for emergency backup purposes and will rarely (if ever) operate.
- Only one VSD compressor per system is eligible for incentive.

This measure is commonly offered for retrofit (or early replacement) projects and new construction or replace on burnout/time-of-sale projects. For a new construction project or if the baseline unit has failed or is near the end of its useful life, the baseline efficiency should be determined from:

- The market industry standard/common practice for the given baseline control type
- Compressed Air and Gas Institute (CAGI) performance sheet data for an equivalently sized new compressor with load/unload or modulating controls.

This protocol is also applicable to projects involving the addition of a VSD controlled trim compressor to a multiple compressor central plant and to projects where an existing air compressor is retrofitted with an add-on VSD.

2.2 Compressed-Air Leak Surveys and Repairs

Compressed-air leak surveys are typically performed by a program-approved third party or a trade ally. Programs typically establish specific guidelines for conducting the survey and reporting the findings.

Energy savings from compressed-air system repairs are determined by multiplying the estimated reduction in compressed air loss in SCFM by the power input per CFM (also known as efficacy) of the air compressor serving the system for the range of loading experienced by the system.

Incentives are typically paid as the least of:

- A fixed dollar amount per rated compressor horsepower

- Full reimbursement for the cost of the leak survey
- A program-defined maximum, not-to-exceed dollar amount.

3 Savings Calculations

This section describes the calculation methods for estimating gross savings from compressed-air projects.

3.1 Savings Calculations for Installing a High-Efficiency Air Compressor

3.1.1 Compressor Power at Full Load

Energy use reduction for all compressor projects can be calculated by the difference between the energy consumed in the baseline operation minus the energy consumed in the post-retrofit operation. Generally, information is required for compressor capacity in both the baseline and post-retrofit scenarios. Appropriate adjustments are made to ensure the flow profile is equivalent between pre- and post-retrofit conditions unless demand improvements have been made that result in a change in the flow profile.

Compressor power at full load can be calculated as follows:

$$\text{Full Load } kW_{rated} = \frac{(\text{Compressor hp}) \times LF_{rated} \times (0.746 \text{ kW/hp})}{(\eta_{motor})} \tag{1}$$

$$\text{Full Load } kW_{rated} = \frac{(\text{Compressor hp}) \times LF_{rated} \times (0.746 \text{ kW/hp})}{(\eta_{motor}) \times (\eta_{VSD})} \tag{2}$$

where:

Compressor hp	= compressor horsepower, nominal rating of the prime mover (motor)
0.746	= horsepower to kW conversion factor
η_{motor}	= motor efficiency (%)
η_{VSD}	= variable-speed drive efficiency (%)
LF_{rated}	= load factor of compressor at full load (typically 1.0 to 1.2)

VSDs have losses, just like other electronic devices that transform voltage. VSD efficiency decreases with decreasing motor load. The decline in efficiency is more pronounced with drives of smaller horsepower ratings. VSD efficiencies typically range from 95% to 97% at full load depending on compressor horsepower (DOE 2020).

Alternatively, full load power may be available from manufacturers or CAGI performance sheet data. Measuring full- and part-load power is even more accurate for a specific site.

Air compressor full-load performance values provided on CAGI data sheets are reported at standard atmospheric conditions (14.5 pounds per square inch absolute [psia], 68 degrees Fahrenheit (ºF), and 0% relative humidity [RH] at sea level). Typically, air compressor operating conditions will differ from the standard conditions, which will impact the rated compressor capacity. To account for these changes, the rated capacity from manufacturers or CAGI performance sheet data should be adjusted from the SCFM to ACFM using Equation 3 (CAGI 2020).

SCFM and ACFM are the two most common methods for rating compressor capacity and performance. SCFM reflects the standard operating conditions used to benchmark compressor performance and is presented in the manufacturer technical specification sheets and CAGI sheets. ACFM represents adjustments to compressor capacity to account for operating conditions at the location of the air compressor. Conditions like location atmospheric pressure (psia), relative humidity (%), and atmospheric temperature (ºF) should be considered when calculating the adjusted air compressor capacity. Some potential pitfalls of using SCFM instead of ACFM include incorrect compressor sizing and design, lower than expected air flow, potential for air shortages during seasonal changes, and incorrect sizing of auxiliary air compressor equipment such as dryers. These issues can disrupt site operations.

Use Equation 3 to convert SCFM to ACFM. Equation 3 accounts for compressor capacity adjustments due to elevation, temperature, and humidity.

$$ACFM = SCFM \times \frac{P_s}{[P_a - (ppm \times RH)]} \times \frac{(T_a + 460)}{(T_s + 460)} \tag{3}$$

where:

P_s	= standard pressure, psia (CAGI and ISO use 14.5)
P_a	= atmospheric pressure, psia
ppm	= partial pressure of moisture at atmospheric temperature
RH	= relative humidity (CAGI and ISO use zero RH)
T_a	= atmospheric temperature, ºF
T_s	= standard temperature, ºF (CAGI and ISO use 68 ºF)

By converting SCFM to ACFM, the actual air flow for site conditions is determined for a given point in time. It is important to note that site conditions are dynamic. As such, compressors should be designed with several factors in mind: year-round conditions including winter and summer extremes, the load profile of the facility, and a factor of safety to account for significant shifts in demand. Weather considerations are especially important for compressors located in facility areas that are not temperature controlled.

The relationship between air capacity and pressure is also highly important. The manufacturer or CAGI performance sheet provides performance values at a fixed pressure setpoint and capacity should be adjusted to account for a change in the operating pressure setpoint at the facility.

One way to adjust capacity is to reference the manufacturer or CAGI data sheets for different pressure setpoints and use linear interpolation to determine the appropriate capacity at the site operating pressure. Manufacturer or CAGI data sheets are normally provided for two or three different pressure setpoints.

Alternatively, pressure adjustments can be done using Boyle's Law (Equation 4), which states that volume is inversely proportional to pressure. This means that if operating pressure is higher than the pressure specified on the CAGI sheet, output capacity of the air compressor will be lower.

$$P_1 \times V_1 = P_2 \times V_2 \qquad (4)$$

where:

P_1 = initial pressure, psig (pressure specified on CAGI sheet)

P_2 = operating pressure, psig (pressure setpoint at site)

V_1 = initial volumetric flow rate, acfm (air volume at CAGI rated pressure adjusted to site ambient conditions)

V_2 = final volumetric flow rate, acfm (air volume accounting for operating pressure)

It is important to note that pressure changes and site conditions have varying effects on different types of compressors. The equations above are reflective of adjustments for positive displacement compressors such as rotary screw and reciprocating compressors. Generally, positive displacement compressor capacity will be affected significantly by variations in ambient conditions, while the full load power will only be affected marginally. For dynamic compressors, like the centrifugal compressor, variations in site conditions and pressure could have a significantly larger swing in both capacity and power, impacting performance even further. The variations in capacity for centrifugal compressors are especially important to consider due to the efficiency losses during low turndown for these machines.

An example capacity adjustment calculation is provided below.

Example 1: Converting SCFM from CAGI sheets to ACFM to account for site elevation, temperature, humidity, and operating pressure.

Assume site personnel want to determine the adjusted capacity of a 75 hp rotary screw compressor with a rated flow of approximately 360 SCFM at 100 psia. The site is located 1,000 feet above sea level, with an average temperature of 60 ºF and relative humidity of 45%. The operating pressure setpoint at the facility is 90 psig.

Step 1: Convert SCFM to $\mathrm{ACFM_{Rated}}$ at the rated manufacturer or CAGI operating pressure using Equation 3:

$$ACFM_{Rated} = 360_{SCFM} \times \frac{14.5_{psia}}{[14.2_{psia} - (0.2563_{ppm} \times 0.45_{\%RH})]} \times \frac{(60_{^\circ\mathrm{F}} + 460)}{(68_{^\circ\mathrm{F}} + 460)}$$

$$ACFM = 365$$

Step 2: Adjust ACFM for operating pressure setpoint using either CAGI sheet interpolation or Equation 4:

$$100_{psig} \times 365_{ACFMr} = 90_{psig} \times V_2$$

$$ACFM = 405.6$$

As Example 1 illustrates, site conditions have a significant impact on compressor capacity.

Alternatively, if adjustment of compressor kW performance is preferred, use Equation 5 to adjust full load compressor kW of the air compressor.

The following expressions are used to correct the compressor full-load performance based on site elevation and operating pressure.

$$\mathrm{kW_{adjusted}} = \mathrm{Full\ Load\ kW_{rated}} \times \frac{\left[\left(\frac{\mathrm{P_{discharge}} + \mathrm{P_{alt}}}{\mathrm{P_{alt}}}\right)^{\frac{0.395}{1.395}} - 1\right]}{\left[\left(\frac{\mathrm{P_{rated}} + 14.5}{14.5}\right)^{\frac{0.395}{1.395}} - 1\right]} \tag{5}$$

where:

Full Load kW_{rated}	= full-load kW of air compressor at full-load capacity and pressure (per CAGI data sheet or manufacturer specifications)
$P_{discharge}$	= actual system discharge pressure (psig)
P_{alt}	= atmospheric pressure based on site elevation above sea level (psia)
P_{rated}	= pressure at rated flow (psig) per CAGI data sheet or manufacturer specified design inlet pressure
14.5	= standard atmospheric conditions (psia) at sea level
(0.395/1.395)	= based on the ratio of specific heat for air at standard atmospheric conditions and isentropic compression with constant specific heats

For pressure-specific adjustments, a common rule of thumb for systems in the 80 to 140 pounds per square inch gauge (psig) range is: for every 2 pounds per square inch (psi) increase (or decrease) in discharge pressure, energy consumption will increase (or decrease) by approximately 1% at full output flow. This rule of thumb closely approximates Equation 5 within this range. Outside this range, Equation 5 is preferred. Equation 6 demonstrates how the "rule-of-thumb" adjustment is calculated:

$$\mathrm{kW_{adjusted}} = \mathrm{Full\ Load\ kW_{rated}} \times [1 - (((\underline{\mathrm{P_{rated}} - \mathrm{P_{discharge}}})/2) \times 0.01)] \tag{6}$$

3.1.2 Compressor Power at Part Load

The rated full-load power of a compressor represents the energy use of the system when operating at full load. At part-load conditions, compressor power is generally lower with common control types. To determine power at part load, the part-load fraction, calculated as the supplied CFM divided by the rated CFM for a given compressor, is matched to the percentage of power using an appropriate table (see Table 1 and Table 2). The operating power can then be calculated at a given capacity using Equation 7:

$$kW_{operating} = kW_{adjusted} \times \% \text{ Power} \tag{7}$$

where:

$kW_{adjusted}$	= Adjusted full-load kW based on actual operating conditions or measured data
% Power	= percentage of power input (%), ratio of the load that the compressor is actually drawing relative to the rated full load Note: % power is not a parameter that can be physically measured, although measuring power and then testing the compressor at full-load will provide the variables needed to calculate percentage of power.

Percentage of power is also influenced by equipment type (reciprocating, rotary screw, etc.) and method of control (throttling, on/off, variable speed, etc.). Table 1 presents typical power versus capacity distributions for rotary screw compressors with multiple control methods. Table 2 presents typical percentage of power versus percentage of capacity curves for centrifugal and reciprocating air compressors. The data in Tables 1 and 2 were developed from standard percentage of power versus percentage of capacity performance curves extracted from Scales and McCulloch (2013) and Smith (2012). It is important to note that a centrifugal compressor can only turn down to approximately 70% and blowdown of compressed air is required to operate at lower capacities. Figure 1 shows examples of percentage of power versus percentage of capacity curves for lubricated rotary screw air compressors.

Table 1. Average Percentage of Power Versus Percentage of Capacity for Rotary Screw Compressors with Various Control Methods

(Scales and McCulloch 2013)

Percentage of Capacity	Load/Unload – Oil-Free	Load/Unload (1 gal/CFM)	Load/Unload (3 gal/CFM)	Load/Unload (5 gal/CFM)	Load/Unload (10 gal/CFM)	Inlet Valve Modulation (w/o Blowdown)	Inlet Valve Modulation (w/Blowdown)	Variable Displacement	VSD w/Unloading	VSD w/Stopping
0%	27%	27%	27%	27%	27%	71%	26%	25%	12%	0%
10%	34%	32%	34%	36%	35%	74%	40%	34%	20%	12%
20%	42%	63%	54%	44%	42%	76%	54%	44%	28%	24%
30%	49%	74%	64%	53%	52%	79%	62%	52%	36%	33%
40%	56%	81%	73%	65%	60%	82%	82%	61%	45%	41%
50%	64%	87%	79%	70%	68%	86%	86%	63%	53%	53%
60%	71%	92%	85%	77%	76%	88%	88%	69%	60%	60%
70%	78%	95%	90%	85%	83%	92%	92%	77%	71%	71%
80%	85%	98%	94%	90%	89%	94%	94%	85%	80%	80%
90%	93%	100%	98%	97%	96%	97%	97%	91%	89%	89%
100%	100%	100%	100%	100%	100%	100%	100%	100%	100%	100%

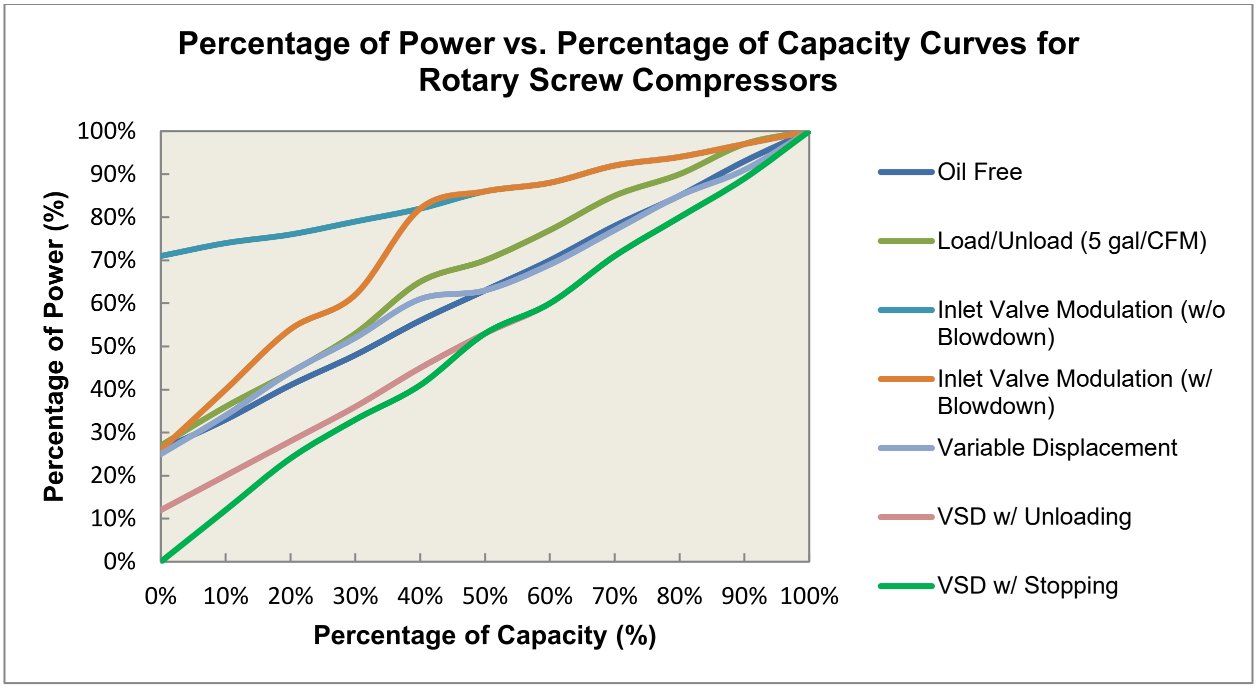

Figure 1. Example Operation Curve (percentage of power versus percentage of capacity curve) for Lubricated Rotary Screw Air Compressor

Table 2. Average Percentage of Power Versus Percentage of Capacity for Reciprocating and Centrifugal Compressors with Various Control Methods

(Compressed Air Challenge)

	Percentage of Power			
Percentage of Capacity	**Reciprocating On/Off Control**	**Reciprocating Load/Unload**	**Centrifugal IBV[a] w/ Blowdown**	**Centrifugal IGV[b] w/ Blowdown**
0%	0%	26%	80%	74%
10%	10%	33%	80%	74%
20%	20%	41%	80%	74%
30%	30%	48%	80%	74%
40%	40%	56%	80%	74%
50%	50%	63%	80%	74%
60%	60%	70%	80%	74%
70%	70%	78%	80%	74%
80%	80%	85%	87%	83%
90%	90%	93%	93%	91%
100%	100%	100%	100%	100%

[a] IBV – Inlet Butterfly Valve Modulation

[b] IGV – Inlet Guide Vane Modulation

In situations where the receiver storage capacity per CFM of supplied air for a load/unload controlled compressed air system does not match one of the default performance curves provided in Tables 1 and 2, it is recommended that a unique profile is developed using the process of interpolation. An example interpolation calculation is provided below.

Example 2: Using Linear Interpolation to Develop Project-Specific Performance Curve for Load/Unload Compressor with Compressed Air Storage

Assume the base-case system on a VSD compressor replacement project consists of a load/unload-controlled rotary screw air compressor with a rated flow of 360 SCFM and approximately 1,000 gallons of receiver storage. The ratio of compressed air receiver capacity (gallons [gal]) to supplied SCFM is approximately 2 gal per SCFM. Using interpolation and the values from Table 1 for 1 gal/SCFM and 3 gal/SCFM load/unload systems; approximate the %Power of a 2 gal/SCFM load/unload-controlled system when operating at 60% capacity.

General Formula for Linear Interpolation

$$\%\mathrm{Pwr}_{z,cap\%} = \%\mathrm{Pwr}_{x,cap\%} + \left[\frac{\mathrm{gal/CFM}_z - \mathrm{gal/CFM}_x}{\mathrm{gal/CFM}_y - \mathrm{gal/CFM}_x} \times (\%\mathrm{Pwr}_{y,cap\%} - \%\mathrm{Pwr}_{x,cap\%})\right] \quad (8)$$

where:

%cap	= specified operating point (% capacity)
$\%\mathrm{Pwr}_{z,cap\%}$	= % power of z gal/CFM system at specified % capacity
$\%\mathrm{Pwr}_{x,cap\%}$	= % power of x gal/CFM system at specified % capacity
$\%\mathrm{Pwr}_{y,cap\%}$	= % power of y gal/CFM system at specified % capacity
$\mathrm{gal/CFM}_x$	= lower bound receiver capacity
$\mathrm{gal/CFM}_y$	= upper bound receiver capacity
$\mathrm{gal/CFM}_z$	= receiver capacity of subject system being evaluated

Using the default performance curves for 1 gal/CFM and 3 gal/CFM load/unload compressed air systems and the known receiver capacities, we can approximate the %Power of a 2 gal/CFM system while operating at 60% capacity as follows.

%cap	= 60%	
$\%\mathrm{Pwr}_{x,cap\%}$	= $\%\mathrm{Pwr}_{1\ gal,60\%}$	= 92% (from Table 1)
$\%\mathrm{Pwr}_{y,cap\%}$	= $\%\mathrm{Pwr}_{3\ gal,60\%}$	= 85% (from Table 1)
$\mathrm{gal/CFM}_x$	= 1 gal/CFM	
$\mathrm{gal/CFM}_y$	= 3 gal/CFM	
$\mathrm{gal/CFM}_z$	= 2 gal/CFM	

$$\%\text{Load}_{2\ gal,60\%} = 0.92 + \left[\left(\frac{2\text{-}1}{3\text{-}1}\right) \times (0.85\text{-}0.92)\right]$$

$$\%\text{Load}_{2\ gal,60\%} = 0.885$$

This process can be repeated for all other common operating points (percentage of capacity values) relevant to the given project or a unique performance curve can be developed by interpolating %Power values for the full range of 0% to 100% in 10% increments (as shown in Table 3).

Table 3. Interpolated Percentage of Power Versus Percentage of Capacity Curve for Rotary Screw Compressor with Load/Unload Controls and Receiver Capacity of 2 gal/CFM

Percentage of Capacity	Percentage of Power Load/Unload (2 gal/CFM)
0%	27%
10%	33%
20%	59%
30%	69%
40%	77%
50%	83%
60%	89%
70%	93%
80%	96%
90%	99%
100%	100%

3.1.3 CFM-bin Hour Profile Analysis Approach

The above methods for determining the instantaneous demand of an air compressor at a given load can be repeated for many bins of hour-CFM operation. This is commonly referred to as a CFM demand profile. A demand profile must be developed to provide accurate estimates of annual energy consumption. A demand profile typically consists of a CFM-bin hour table summarizing hours of usage under all common loading conditions throughout a given year.

Table 4 provides an example of a compressed air CFM-bin hour profile based on the following assumptions:

- The base-case compressor system consisted of a 75 hp rotary screw compressor with inlet valve modulation (with blowdown) controls, an adjusted full-load power of approximately 65.5 kW, and a rated flow of approximately 360 SCFM.
- The post-retrofit case compressor system consists of a 75 hp rotary screw compressor with VSD (with stopping) controls, an adjusted full-load power of approximately 67.5 kW, and a rated flow of approximately 360 SCFM.

The annual CFM profile is used to determine base case and proposed case energy use. For both, compressor electricity demand for each CFM-bin should be determined from actual metering data, spot power measurements, or CFM-to-kW lookup tables. When analyzing metered trend data, the hourly average percentage of power should be used to determine which CFM-bin an individual hour is assigned.

The difference in energy consumption between an air compressor operating in idling mode and being physically shut down can be significant depending on the base case and post-retrofit case methods of system control (as demonstrated by CFM-bin6) where base case consumption includes 13,113 kilowatt-hours (kWh) when the inlet valve modulation (with blowdown) compressor is operating in idling mode for approximately 770 hours per year; whereas the post-retrofit case VSD-controlled system (with stopping) has zero energy consumption for the same bin-hours. It is also common to differentiate between compressor systems operating in “timed-out” mode versus “shut-down” mode. “Timed-out" mode is generally determined from metering. “Shut-down” mode is typically determined from staff interviews and is verified from metering.

Table 4. Sample Compressed Air CFM-Bin Hour Table Base and Post Cases

CFM-bin #	CFM Load Profile	Base Case: Rotary Screw Compressor With Inlet Valve Modulation (w/Blowdown)				Post Case: VSD Rotary Screw Compressor w/Stopping			
		Percentage of Power	H/Yr	Input Power (kW)	kWh	Percentage of Power	H/Yr	Input Power (kW)	kWh
CFM-bin 1	324	97%	200	63.5	12,707	89%	200	60.1	12,015
CFM-bin 2	288	94%	2,440	61.6	150,231	80%	2,440	54.0	131,760
CFM-bin 3	216	88%	170	57.6	9,799	60%	170	40.5	6,885
CFM-bin 4	180	86%	430	56.3	24,222	53%	430	35.8	15,383
CFM-bin 5	144	82%	1,100	53.7	59,081	41%	1,100	27.7	30,443
CFM-bin 6	0 idling *	26%	770	17.0	13,113	0%	0	0.0	0.0
CFM-bin 7	0 shutdown	0%	3,650	0.0	0.0	0%	4,420	0.0	0.0
Total kWh/yr		**269,153**				**196,486**			

The energy consumption for each CFM-bin is determined from the product of the average compressor demand and the number of hours in each bin (Equation 9. The sum of the kWh bin values gives the annual consumption (Equation 10).

$$\Delta kWh_{bin1} = (\text{Base } kW_{operating\_bin1} - \text{Post } kW_{operating\_bin1}) \times \text{CFM-bin 1 H} \quad (9)$$

$$\Delta kWh_{binN} = (\text{Base } kW_{operating\_binN} - \text{Post } kW_{operating\_binN}) \times \text{CFM-bin N H}$$

where:

Base $kW_{operating\_bin1}$ = baseline demand at part-load associated with CFM-bin 1

Post $kW_{operating\_bin1}$ = post demand at part-load associated with CFM-bin 1

Base $kW_{operating\_binN}$ = baseline demand at part-load associated with CFM-bin N

$\text{Post kW}_{operating\_binN}$ = post demand at part-load associated with CFM-bin N

Total energy reduction:

$$kWh/yr = \sum_{1\text{-}n} [\ \Delta kWh_{bin1} + \Delta kWh_{bin2} + \ldots + \Delta kWh_{binN}\ ] \quad (10)$$

where:

ΔkWh_{bin1} = energy reduction for CFM-bin 1

ΔkWh_{binN} = energy reduction for CFM-bin N

Another common practice is to incorporate day-types into the CFM-bin analysis as compressed air demands are often tied to facility operations and production schedules. This approach can be particularly useful when developing 8,760 load shapes and when calculating peak demand savings. Day-type analysis is also beneficial when estimating savings from leak repairs and upgrading compressed air dryers.

The CFM-bins should be carefully developed to be applicable to the facility operation. Enough CFM-bins should be present to adequately characterize the granularity of operations. At a minimum, characterizing each individual shift and variances between day types (e.g., weekdays vs. weekends) is needed. A consistent method that nearly always provides appropriate granularity is the daily profile analysis, which obtains the average hourly profile for each hour of each day of the week.

3.1.4 Addressing Uncertainty

During compressed air energy efficiency project evaluations, a common issue arises from a lack of information about baseline energy consumption and lack of airflow data. In the absence of measured or trended CFM data, parameters such as load profile and operating hours must be developed by the evaluator, based on interviews with on-site facility personnel, reviews of historical operations/production levels, reported operating schedules, and short-term (two weeks or more) individual compressor power recordings.

Another common finding from compressed air program evaluations is the fact that baseline and post-installation energy savings calculations are not normalized to account for changes in facility production levels. A best practice when estimating the energy savings of a project is to develop correlations between, not only energy usage and airflow, but also production whenever possible. This allows the evaluator to select the optimal normalization parameter to improve the accuracy of estimated savings.

One common method is to measure compressor power. The percentage of power can be correlated to percentage of flow using the appropriate compressor curve for the given control type. In this way, a load profile can be developed that can be used to compare the baseline and post systems at equivalent flow.

For systems with load and unload compressors, timing the load/unload cycles can be an effective way of determining percentage of capacity. A load/unload compressor either produces full flow or no flow; thus, the percentage of measured time when the compressor is loaded is equivalent to percentage of capacity.

3.2 Savings Calculations for Compressed-Air Leak Surveys and Repairs

3.2.1 Quantifying the Compressed-Air Leakage

Before a compressed-air leak survey is conducted, a system leak-down test should be performed to estimate the combined loss (CFM) of compressed-air leaks. Leak-down tests are best performed at the air receiver by isolating the receiver from the supply side of the system. The basic procedures for conducting a leak-down test are:

- Estimate the total storage volume of the compressed-air system, receivers, main headers, etc., in cubic feet.
- During nonproduction hours, start the system and allow it to reach normal operating system pressure.
- Turn off all production loads.
- Shut off the compressor(s).
- Allow the system to "leak down" to approximately half the full load pressure (psig) and record the time it takes to reach this point.
- Use the following formula:

$$\text{Leak Flow SCFM (Free Air)} = [(V \times \Delta P)/(\text{Time} \times P_{alt})] \times 1.25 \tag{11}$$

where:

V	=	total storage volume of compressed-air system in cubic feet
ΔP	=	drop in line pressure during leak down test in psig ($P_1 - P_2$)
P_{alt}	=	atmospheric pressure (psia) corrected for local altitude (elevation)
T	=	time it takes for line pressure to drop by 50% from normal system operating pressure (minutes)

The 1.25 multiplier corrects leakage to normal system pressure, allowing for reduced leakage with system pressure falling to 50% of the initial reading.

In many cases, a leak-down test is impractical or critical users must have air at all times. In these instances, flow should be estimated by measuring compressor power and correlating to flow (reference table/methods above). This should be done during a non-production period, such as a weekend. During this test, it is important to identify any non-leak users of air. The measured compressor flow should be reduced by the total air use of the non-leak applications to determine the actual leak volume.

Leakage is expressed in terms of the percentage of system capacity. The percentage lost to leakage should be less than 10% in a well-maintained system (Marshall 2013). Poorly maintained systems can have losses as high as 20% to 30%.

3.2.2 Quantifying the Energy Impacts of Compressed Air Leak Repairs

Energy savings resulting from the repair of compressed-air leaks can be significant. The best method for estimating impacts is the CFM-bin approach highlighted in Section 3.1.3. The baseline load profile is developed and simulated to determine baseline energy usage. The upgrade load profile is then generated showing the flow reduction resulting from the leak repair and simulated to give the energy usage post-repair. The difference in energy usage between the baseline and post-energy simulation is the energy use reduction associated with the leak repair.

The full CFM-bin approach is highly accurate, but it can be time consuming and overly complicated for small projects. It also works best when full trend data are available to develop a CFM demand profile. A simplified method, outlined below, closely approximates the CFM-bin approach. This simplified approach is only applicable under these conditions:

- The compressed air system is well-controlled and operates predictably
- The system uses a single compressor to meet variable loads and functions as the trim compressor
- The flow reduction is small enough that the quantity of compressors operating is unchanged (if the flow reduction is significant enough to shut off a compressor, the CFM-bin method must be used).

If the above conditions are met, use the simplified savings algorithm below to estimate the energy savings of a leak repair:

$$\text{kWh Saved} = \text{repaired leak volume} \times kW_{FL}/CFM_{rated} \times \text{Hours} \times \text{CCAF} \qquad (12)$$

where:

kWh Saved	= kWh saved per year
repaired leak volume	= rate of air loss from leaks repaired (SCFM)
kW_{FL}	= rated full load kW of the trim air compressor
CFM_{rated}	= rated CFM output of the trim air compressor
Hours	= annual operating hours of the flow reduction (typically the compressed air system operating hours for leak repair measures)
CCAF	= trim compressor control type adjustment factor

The adjustment factor will vary based on the method of system control. Table 5 presents typical adjustment factors for common control strategies. An adjustment factor should be used to ensure that energy savings estimates accurately represent savings. It is common for vendors to use an average measured kW/CFM value, but this frequently results in overestimated savings. The adjustment factors provided in Table 5 were developed using data from the percentage of power versus percentage of capacity curves in Section 3.1.2 (Table 1 & Table 2). Each CCAF value represents the slope of the performance curve when operating within the 40% to 80% capacity range as this is a common operating range for a trim compressor.

Table 5. Recommended Adjustment Factors for Determining Energy Savings from Compressed Air Leak Repairs

Control Method	CCAF
Reciprocating—on/off control	1.00
Reciprocating—load/unload	0.74
Screw – load/unload oil free	0.73
Screw – load/unload 1 gal/CFM	0.43
Screw – load/unload 3 gal/CFM	0.53
Screw – load/unload 5 gal/CFM	0.63
Screw—load/unload 10 gal/CFM	0.73
Screw—inlet modulation	0.30
Screw—inlet modulation w/unloading	0.30
Screw—variable displacement	0.60
Screw—variable speed drive	0.97
Centrifugal Compressors	Varies[a]

[a] Centrifugal part-load performance should be reviewed individually depending on the facility load. Centrifugal compressors have good part-load performance within the throttle range of about 0.86 for IGV and 0.67 for IBV controls. Below the throttle range, a centrifugal compressor simply discharges excess compressed air generated through the blowoff valve; therefore, if the compressor is operating in blowoff, the CCAF would be 0. A value between the throttle range and blowoff CCAF may be applicable depending on the time a specific compressor typically operates within each range of control.

Below is an example calculation of the estimated energy savings resulting from compressed air leak repairs based on the following assumptions:

- Compressed air is supplied to the system by a 75 hp rotary screw compressor with VSD controls, a full-load power of approximately 67.5 kW, and a rated flow of approximately 360 SCFM. The compressor runs 4,160 hours per year. The estimated rate of air loss from leaks repaired is approximately 58 SCFM.

$$\text{kWh Saved} = \text{repaired leak volume} \times \text{kW}_{FL}/\text{CFM}_{rated} \times \text{Hours} \times \text{CCAF}$$

Per Table 5, CCAF for “Screw—variable speed drive” = 0.97

$$\begin{aligned}\text{kWh Saved} &= (58\ \text{SCFM}) \times (67.5\ \text{kW}/360\ \text{SCFM}) \times (4{,}160\ \text{hours}) \times (0.97)\\ &= 43{,}883\ \text{kWh}\end{aligned}$$

The methods shown for the energy impact of repairing leaks can also be applied to other compressed air measures that reduce flow, such as installing high-efficiency air nozzles or installing no-loss condensate drain valves.

3.2.3 Leak Volume Quantification Best Practices

The following basic procedures should be followed when quantifying energy savings resulting from leak repairs:

- Impacts from leaks should be supported with formal documentation. The rated power input to CFM output (air compressor specific power) should be supported by trended system data whenever possible.
- The leakage rate (CFM) from a compressed-air leak can be estimated based on the system line pressure and approximate orifice diameter of the crack or leak identified. Leakage rate is proportional to the square of the measured orifice diameter. Table 6 shows the leakage rates for various line pressures (psig) and leak orifice diameters (inches). Correction factors for well-rounded versus sharp orifice shapes must be applied to the leakage rates to ensure estimates are conservative.

Table 6. Leakage Rates (CFM) for Different Supply Pressures and Approximately Equivalent Orifice Sizes

(DOE 2013)

Pressure (psig)	Orifice Diameter (in.)					
	1/64	1/32	1/16	1/8	1/4	3/8
70	0.29	1.16	4.66	18.62	74.4	167.8
80	0.32	1.26	5.24	20.76	83.1	187.2
90	0.36	1.46	5.72	23.1	92	206.6
100	0.40	1.55	6.31	25.22	100.9	227
125	0.48	1.94	7.66	30.65	122.2	275.5

Values should be multiplied by 0.97 for well-rounded orifices and by 0.61 for sharp orifices (DOE 2013).

- Once leak repair work is complete the combined air loss (CFM) of the logged leaks that were repaired should be summed and compared to the total leakage determined from the preliminary leak-down test. Identifying all leaks in a compressed-air system is nearly impossible, so it is appropriate to allocate a portion of the leak-down test CFM to "undetected leakage." A post-repair leak-down test should also be performed to quantify leak reduction.

4 Measurement and Verification Plan

This protocol describes methods for estimating gross savings from compressed air projects. When choosing an option, consider the following factors:

- The equation variables used to calculate savings
- The uncertainty in the claimed estimates of each parameter
- The cost, complexity, and uncertainty in measuring each variable
- The interactive effects of concurrently implementing multiple compressed-air efficiency measures.

4.1 International Performance Measurement and Verification Protocol Option

The preferred approach for evaluating compressed air electronically commuted motors (ECMs) is International Performance Measurement and Verification Protocol Option A: Retrofit Isolation (Key Parameter Measurement). Options B, C, and D can be used in limited applications, but Option A is the preferred approach. Discussions on the feasibility and applicability of the other approaches are provided below.

4.1.1 Option A: Retrofit Isolation (Key Parameter Measurement)—Preferred Approach

International Performance Measurement and Verification Protocol Option A (Retrofit Isolation Key Parameter Measurement) offers the best approach for measuring the energy consumption of compressed-air system. Option A relies on field measurements of key performance parameters and estimates of key parameters not selected for field measurements. Field measurements are typically collected for compressor load current (amps) or true root mean square (RMS) power (Watts).

Parameters such as airflow, line pressure, compressor specific power, part-load performance, and operating hours are typically determined from a combination of one-time spot measurements, historical production data, manufacturers' specifications, CAGI standard data sheets, and interviews with the customer. Using Option A, the measurement boundary is established on the line side of the power supply feeding the air compressor or VSD.

Interval field measurements of compressor load current (amps) coupled with spot power measurements or true RMS power (Watts) measurements are used to determine the instantaneous operating load of an air compressor and to develop trends of energy consumption over time (minimum metering period of two weeks). Equation 11 is used to convert interval measurements of load current (amps) and one-time spot measurements of line voltage and power factor into operating load ($kW_{operating}$) for three-phase motors.

$$kW_{operating} = \sqrt{3} \times Amps \times Volts_{RMS} \times PF \tag{13}$$

where:

Amps = measured load current

$Volts_{RMS}$ = measured True RMS phase-to-phase voltage

PF = measured power factor

True RMS voltage, load current, and power factor should be measured with the system operating under all common loading conditions. Each "common loading condition" should correlate with an established bin of hour-CFM operation. The derived operating load for each CFM-bin is then inserted into Equation 7 (most commonly as the parameter "Post $kW_{operating\_binN}$") to determine annual consumption and energy reduction.

4.1.2 Option B: Retrofit Isolation (All Parameter Measurement)

The savings created by compressed air ECMs can be determined using Option B (Retrofit Isolation – All Parameter Measurement); however, the degree of difficulty and costs associated with enhanced measurement and verification will increase. By definition Option B requires "field measurement of all key performance parameters which define the energy use of the ECM-affected system." This implies that in addition to measuring load current or true RMS power, the evaluator is required to measure airflow (SCFM) and operating hours. Option B also requires pre-retrofit metering before the measure is implemented.

4.1.3 Option C: Whole Facility

Typically, Option C is not applicable because compressed air is generally not more than 10% of a typical facility's energy consumption.

4.1.4 Option D: Calibrated Simulation

Option D can be used in circumstances where multiple ECMs are concurrently implemented; however, this approach can be cost prohibitive and is less common when evaluating ECMs only affecting compressed air systems.

4.2 Verification Process

In accordance with Option A, the first step of the protocol entails verifying key data collected on typical program application or rebate forms, including information on the baseline compressor system. This typically includes:

- Number of shifts per day, shift-hours per week, weekend hours per week, and estimated total operating hours per year
- Average air demand (SCFM) for each shift
- Baseline equipment use pre- and post-retrofit (lead, trim, or backup compressor)
- Baseline compressor system type (reciprocating, screw oil-less/oil-flooded, two-stage, centrifugal, vane, etc.)
- Baseline compressor system control type (load/no load, inlet modulating dampers, other)
- Baseline compressor system operating pressure (psig) at rated SCFM
- Manufacturer, model number, system type, control method, nominal horsepower, rated SCFM, operating pressure at rated SCFM, and installation date for the new energy-efficient air compressor.

For compressed-air leak survey and repair projects, the following information is also frequently requested:

- Whether the facility currently has a formal compressed-air leak detection program in place
- An estimate of total plant air leakage as a percentage of total use
- Type and model of leak detection instrument used by the trade ally to conduct the survey.

Some of these data can be verified using a desk review of invoices, manufacturer specifications sheets (which are typically required for rebate/incentive payments), compressed-air survey reports, or an on-site audit of a sample of participants to verify the quality of self-reported information. If efficiency and unit capacity are not collected for each participant, program application requirements should be modified to include these important data.

4.3 Data Requirements

The energy use of a compressed-air system is typically governed by plant production levels. The actual recommended metering duration for any given compressed-air project should be established to represent all operating modes of the facility. This period should span two full operating cycles from maximum energy use (e.g., weekday production) to minimum (e.g., weekend nonproduction) to confirm the rate of recurrence in the metered data. This is also done to evaluate the consistency of operations on a cycle-to-cycle basis and avoid circumstances where data collected during a single cycle coincided with abnormal operations. For most non-weather-dependent compressed-air applications, a metering period of one month or less is acceptable.

Sampling intervals of 15 to 60 seconds are recommended[2], although sampling should occur at a high enough frequency to avoid aliasing errors associated with rapidly fluctuating system demand. In general, the sampling frequency should be at least twice the frequency of events in the system, such as compressor load and unload cycles. In most applications, a sampling interval of 15 to 60 seconds satisfies this requirement.

The minimum data required to evaluate a high efficiency air compressor replacement project are:

- Equipment manufacturer, model, and serial number
- Compressor system type (e.g., reciprocating, oil-flooded rotary screw, centrifugal)
- Prime mover (motor) efficiency
- Rated compressor shaft horsepower (brake horsepower) or rated compressor horsepower and prime mover (motor) load factor
- Rated fully loaded SCFM output

[2] The DOE AIRMaster+ tool (https://www.energy.gov/eere/amo/articles/airmaster) recommends sampling at 15 second intervals. A lower bound of 15 seconds is recommended to capture instantaneous changes in air compressors.

- Rated input power of the compressor in kW over output flow rate in CFM (at rated pressure)
- Annual operating hours of constant speed or modulating compressors at a range of loadings
- Load factor of baseline constant speed or modulating compressor
- Percentage of CFM versus percentage of kW curve of new variable displacement capacity or VSD compressor
- Type of control system (modulation, load/no-load, VSD, variable displacement, etc.).

All of the above listed parameters should be gathered for both the baseline and energy-efficient equipment.

Parameters to be spot measured during the verification include:

- Integrated true RMS kW three-phase power under all common compressor loading conditions.

Parameters to be metered or trended:

- Preferred method: True poly-phase RMS power (kW): This protocol prefers a trend log of true poly-phase RMS power for the circuit powering the VSD compressor. The selected sampling interval should be at a high enough frequency to avoid aliasing errors and at least twice the frequency of events in the system. In general, a sampling interval of once per minute is preferred.
- Alternative method #1: In lieu of true power metering, trending of current (amperage) combined with several one-time true power measurements can be used for base-loaded/constant speed systems. This method can also be used with variable frequency drive compressors as long as true-RMS current transducers are used.
- Alternative method #2: If independent true power metering or trending of current (amperage) coupled with spot power measurements is not possible, it is acceptable to use trend data from a central master control or building automation system. It is preferable to have building automation system trend logs of true poly-phase RMS power with a maximum sampling interval of once per five minutes, and one minute or less is required for load/unload controls. One-time spot power measurements should be performed to verify the accuracy of the control system values.

Additional data required to evaluate compressed air leak survey and repair projects include:

- Compressed-air system specific power (kW/CFM), including compressors, dryers, and significant end uses over a range of CFM loadings
- Supply and demand side one-line diagram showing all generation equipment and significant end uses
- Presence of intermediate pressure and/or flow controllers

- Delivery pressure
- Historical production data for systems affecting compressed-air consumption (number of products produced, active equipment, etc. as appropriate for facility). Production data should be collected for both the pre-and post-retrofit measurement period and appropriate production adjustments should be made to the collected data.

Data to be collected and utilized, when available:

- Measured or trended airflow (SCFM) data can be quite advantageous when evaluating compressed-air ECMs; however, this information can be difficult to obtain and is not generally collected unless the existing compressed-air system controls already have the capability. In the absence of measured or trended CFM data, the evaluator must develop parameters such as load profile and operating hours, based on interviews with on-site facility personnel; reviews of historical operations/production levels; reported operating schedules, and short-term (2 weeks or more) individual compressor power recordings.

5 Data Collection Methods

5.1 Metering

The typical metering equipment used to measure and trend the energy consumption of a VSD compressor are:

- Handheld (or portable) power meters to measure true RMS voltage, current, power, and power factor at all common loading conditions.
- Current transducers for measuring load current while metering (preferably with a linearity accuracy of ±1.0% of the reading). Recording amp loggers are acceptable as long as spot measurements of compressor power are performed with a handheld kW meter at various loadings.
- Watt-hour transducers to measure true power (kW) of one, two, or three phases of a system.
- Meter recorders (data loggers) with adequate storage capacity to match logging interval and measurement frequency.

The selected measurement equipment should always be installed on the line side of a VSD compressor, not on the load side. Measurements from the output of a VSD compressor can lead to significant data errors. In the pre- and post-retrofit measurement periods, all regularly operating compressors serving a common system should be logged simultaneously regardless of quantity of compressors. Compressors that are used only for backup purposes do not need to be logged, although it is good practice to do so to validate that the equipment was never used. Often post-retrofit only measurements are taken and the pre-retrofit power profile is estimated using the post-retrofit CFM (from kW to CFM conversions), data from the CAGI data sheet for the baseline air compressor system, and generic control curves from Table 1 for the baseline control method.

5.2 Ultrasonic Leak Detectors for Compressed Air Leak Surveys

An ultrasonic leak detector with a frequency response of 35 to 45 kHz should be used to conduct compressed air leak surveys. It is also beneficial to use a set of noise attenuating headphones designed to block intense sounds that often occur in industrial environments so that the user may easily hear the sounds received by the instrument.

Ultrasonic leak detectors are an effective tool for identifying and locating leaks in a compressed air system but should not be relied upon for quantifying the rate of leakage. The accuracy of these devices is dependent on operator experience and proximity to source; they are inherently inaccurate as leakage rates are not directly measured and instead are correlated based on the amount of sound produced by a given leak in decibels. The best practice for quantifying rate of leakage is to conduct leak-down tests prior to and immediately following leak repairs to determine the actual system impact.

6 Methodology

6.1 General Discussion

The primary energy savings verification method is to monitor, by metering, energy use over a time period that reflects a full or complete range of the underlying operations within a specific industrial facility. Monitoring for periods of less than one year, as is most often the case, will require that annual energy use be approximated based on the results of short-term metering and historical production data.

A common issue encountered during compressed-air energy efficiency project evaluations is a lack of information about baseline energy consumption. In many instances, baseline consumption must be derived based on pre-retrofit production levels, reported equipment performance, as well as equipment and component specifications. Key parameters to be determined include motor efficiencies, load factors, load profiles, operating hours, total system SCFM and compressor efficacies (kW/CFM). Often, this information must be gathered through interviews with the program participant, implementer, or energy advisor directly involved with the project.

Other resources frequently used to inform baseline assumptions include:

- Equipment tags[3]
- Historical trending from an energy management system
- Engineering reports and calculations generated during the design and application phases of the project
- Rebate or incentive program application forms.

When determining energy savings for VSD compressors, production data must be normalized to an independent normalizing variable. A unit indicating a relative level of production should be obtained from the site, often provided as units produced, hours of machine operation, or labor hours, depending on the site and the availability of information.

Preferably, the independent variable would be collected with sufficient granularity so a correlation can be developed between the measured compressed air energy consumption and the independent variable. The correlation should have a coefficient of determination (R^2) value of at least 0.90 to be of value to the analysis. The pre- and post-retrofit periods should then be normalized to an annual variable for units of production to determine the annual effect of the system improvement. If an annual value is unavailable, using an average of production between the pre- and post-retrofit periods can be acceptable.

Many sites may not be able to provide an independent variable for normalization. In these cases, normalizing to flow is an acceptable alternative. Two methods are used depending on the type of ECM implemented:

[3] It is common for baseline compressor systems to be salvaged or kept in service and converted to an emergency backup role. This provides an opportunity for the evaluator to observe and collect information from equipment tags.

- ECMs that reduce system flow (leaks, air nozzles, condensate drains): For this type of upgrade, the individual installed components should be inspected, and CFM reduction confirmed. The flow reduction can then be modeled via a bin table approach using the measured compressor data and simulating the decrease in energy consumption caused by the decrease in flow.
- ECMs that improve system specific power (new air compressors, compressor controls): For this type of upgrade, the system CFM should be determined at each measured point for both the baseline and the installed systems. The CFM should then be compared. The pre- and post-retrofit periods should be normalized to an annual CFM demand profile. The system should then be simulated via a bin table approach at the normalized CFM level using the correlation between flow and power for the respective system.

In a new construction situation where past process production volume and past energy consumption data are unavailable, the determination of energy use per unit of production will have to be based on some form of comparable site such as a similar process in-house or in-company at another facility. For new construction or normal end-of-life replacement projects the baseline system efficiency is determined from the minimum allowed by current local jurisdictions.

The key parameters from Equation 3 are: % Power, ΔkW, and annual operating hours. Each will fluctuate based on the operating load profile of the VSD compressor. Actual post-retrofit consumption can be determined from the sum of multiple iterations of Equation 3, where a unique calculation must be performed for each common loading condition (i.e., using a bin table method). The compressor load profile dictates the number of iterations. Metering generally provides this information.

6.2 Step-by-Step Procedures for Evaluating High-Efficiency/Variable-Speed Drive Air Compressor Installation Projects

This section of the protocol summarizes the basic step-by-step procedures to be performed when evaluating a high efficiency/VSD compressor replacing a modulating compressor measure.

Step 1: Collect product performance data for baseline and new high efficiency/VSD air-compressor equipment. If product literature is not available, data should be collected from the equipment nameplate. Product literature may be obtainable online after leaving the site using the manufacturer and model number. A sample data collection form is shown in Table 7. Note that the data fields shown in Table 7 should be collected for both the baseline and new equipment.

Table 7. General On-Site Data Collection Form for Air Compressor

Air Compressor General Data Collection Form			
Manufacturer:		Rated Flow (SCFM):	
Model Number:		Pressure at Rated Flow (psig):	
Nominal hp:		Full Load kW_{rated}:	
Drive Motor Efficiency:		Fan Motor hp and Efficiency (if applicable):	
Air-Cooled/Water-Cooled:	❒ Air-cooled ❒ Water-cooled		
Duty:	❒ Lead (Primary) ❒ Trim (Secondary) ❒ Back-up		
Compressor Type:	❒ Rotary Screw (oil-flooded) ❒ Rotary Screw (oil-less) ❒ Centrifugal ❒ Other______		
Control Type (Screw Compressors)	❒ On/Off		
	❒ Load/Unload Total Storage Volume (gallons): _______		
	❒ Inlet Modulating Dampers ❒ w/blowdown ❒ w/o blowdown		
	❒ Variable Speed Drive (VSD) ❒w/unloading ❒ w/stopping		
	❒ Variable Displacement ❒ Other		

Step 2: Determine compressor power at full load for baseline and new high efficiency/VSD air-compressor units using either CAGI performance sheet data, metered full-load and fully unloaded kW data, or derived using Equations 1 and 2. On projects involving the replacement of an older air compressor system, the evaluator may encounter some difficulty in locating CAGI data sheets, product literature, or manufacturer specifications for the baseline system. In the absence of historical metering data or product literature, the full-load kW for an air compressor system can be derived using Equation 1 or 2:

$$\text{Full Load } kW_{rated} = \frac{(\text{Compressor hp}) \times LF_{rated} \times (0.746 \text{ kW/hp})}{(\eta_{motor})} \qquad (1)$$

$$\text{Full Load } kW_{rated} = \frac{(\text{Compressor hp}) \times LF_{rated} \times (0.746 \text{ kW/hp})}{(\eta_{motor}) \times (\eta_{VSD})} \qquad (2)$$

where:

Compressor hp = compressor horsepower, nominal rating of the prime mover (motor)

0.746 = horsepower to kW conversion factor

η_{motor} = motor efficiency (%)

η_{VSD}	= variable-speed drive efficiency (%)
LF_{rated}	= load factor of compressor at full load (typically 1.0 to 1.2)

Typically, the compressor hp will be known by the customer or on-site personnel. Motor efficiency and load factor may or may not be known by on-site personnel and may need to be estimated using engineering judgment informed by known parameters such as system type, method of control, and age.

Step 3: Once rated compressor power at full load for the baseline and new high efficiency/VSD air compressor have been determined, correct these values for site-specific conditions using Equation 5 or the "rule-of-thumb" approach (Equation 4). The two primary adjustments that must be made pertain to atmospheric pressure based on site elevation above sea level and actual system discharge pressure (psig).

Preferred Approach

$$\mathrm{kW_{adjusted}} = \text{Full Load } \mathrm{kW_{rated}} \; \frac{\left[\left(\frac{P_{discharge} + P_{alt}}{P_{alt}}\right)^{\frac{0.395}{1.395}} - 1\right]}{\left[\left(\frac{P_{rated} + 14.5}{14.5}\right)^{\frac{0.395}{1.395}} - 1\right]} \qquad (5)$$

where:

Full Load kW_{rated}	= full load kW of air compressor at full load capacity and pressure (per CAGI data sheet)
$P_{discharge}$	= actual system discharge pressure (psig)
P_{alt}	= atmospheric pressure based on site elevation above sea level (psia)
P_{rated}	= pressure at rated flow (psig) per CAGI data sheet
14.5	= standard atmospheric conditions (psia) at sea level
(0.395/1.395)	= based on the ratio of specific heat for air at standard atmospheric conditions and isentropic compression with constant specific heats

Alternate "Rule-of-Thumb" Approach for Correcting for Discharge Pressure

Although not the preferred approach, a general rule of thumb for air compressors with a rated pressure capacity of 100 psig is: for every 2 psi increase or decrease in discharge pressure, energy consumption will increase or decrease by approximately 1% at full output flow. A sample calculation is shown below:

$$kW_{adjusted} = \text{Full Load } kW_{rated} \times [1 + (((P_{rated} - P_{discharge})/2) \times 0.01)] \qquad (6)$$

Step 4: Once the rated compressor power at full load for the baseline and new high efficiency/VSD air-compressor equipment has been adjusted for site-specific conditions, develop a CFM demand profile. A demand profile consists of a CFM-bin hour table, summarizing hours of usage under all common loading conditions throughout a given year for the base and post-retrofit case conditions.

Table 8 provides an example of a Compressed Air CFM-bin Hour Profile. The base and post-retrofit case profiles shown in Table 10 were developed based upon the following assumptions:

- The base-case compressor system consisted of a 75 hp rotary screw compressor with inlet valve modulation (with blowdown) controls, an adjusted full-load power of approximately 65.5 kW, and a rated flow of approximately 365 SCFM.
- The post-retrofit case compressor system consists of a 75 hp rotary screw compressor with VSD (with stopping) controls, an adjusted full-load power of approximately 69.2 kW, and a rated flow of approximately 365 SCFM.

Table 8. Example Compressed Air CFM-Bin Hour Table - Base and Post Cases

CFM-bin Number	Air Demand Load Profile (SCFM)	%Capacity[a]	Base Case Hours per Year	Post Case Hours per Year
CFM-bin 1	324	90%	2,640	2,640
CFM-bin 2	288	80%	150	150
CFM-bin 3	216	60%	170	170
CFM-bin 4	180	50%	430	430
CFM-bin 5	144	40%	1,130	1,130
CFM-bin 6	0 idling	26%	770	0
CFM-bin 7	0 shut-down	0%	3,650	4,420
	Total Hours		**8,760**	**8,760**

[a]Percentage of flow (part-load fraction) values were determined assuming a rated output flow of 365 SCFM.

Step 5: Once the base and post-retrofit case CFM demand profiles have been developed, calculate the base case and proposed case energy usage. For both base and post-retrofit cases, compressor electricity demand for each CFM-bin should be determined from actual metering data, spot power measurements, or CFM-to-kW lookup tables (refer to Sections 4.3 and 5.1 for guidance on measurement and verification data requirements and data collection methods).

When actual meter or spot power measurement data are unavailable, the percentage of power at part-load for each CFM-bin is typically determined using the calculated percentage of flow values and generic CFM-to-kW lookup tables (see Table 1 and Table 2 in Section 3.1). Percentage of power is influenced by equipment type and method of control. Percentage of capacity versus percentage of power profiles pertinent to the example project for the base and post-retrofit cases are provided in Table 9.

Table 9. Average Percentage of Power Versus Percentage of Capacity for Base Case and Post Case for Example Project

(Scales and McCulloch 2013)

Percentage of Capacity	Base Case: Rotary Screw w/Inlet Valve Modulation (w/Blowdown)	Percentage of Power for Post Case: VSD Rotary Screw Compressor w/Stopping
0%	26%	0%
10%	40%	12%
20%	54%	24%
30%	62%	33%
40%	82%	41%
50%	86%	53%
60%	88%	60%
70%	92%	71%
80%	94%	80%
90%	97%	89%
100%	100%	100%

Using the percentage of power values from Table 9 and the percentage of capacity values calculated in Step 4, the power at part load (kW) for each CFM-bin is determined using Equation 7:

$$kW_{operating} = kW_{adjusted} \times \%\ Power \qquad (7)$$

where:

$kW_{adjusted}$ = Adjusted full load kW

% Power = percentage of power input (%), ratio of the load that a motor is actually drawing relative to the rated full load.

Note: % Power is not a parameter that can be physically measured.

Revisiting the example problem introduced in Step 4, the part-load power (kW) for each CFM-bin is calculated below and is shown in Table 10.

Table 10. Percentage of Power and Operating Load or Base Case and Post-Retrofit Case for Example Project

CFM-bin Number	CFM Load Profile		Base Case		Post Case	
		Percentage of Capacity	Percentage of Power	$kW_{operating}$	Percentage of Power	$kW_{operating}$
CFM-bin 1	324	90%	97%	63.5	89%	60.1
CFM-bin 2	288	80%	94%	61.6	80%	54.0
CFM-bin 3	216	60%	88%	57.6	60%	40.5
CFM-bin 4	180	50%	86%	56.3	53%	35.8
CFM-bin 5	144	40%	82%	53.7	41%	27.7
CFM-bin 6	0 idling	0%	26%	17.0	0%	0.0
CFM-bin 7	0 shutdown	0%	0%	0.0	0%	0.0

Obtaining an actual percentage of power versus percentage of capacity performance curve for the specific air compressor system being evaluated is recommended (if available). A system-specific curve can also sometimes be developed based on information provided on CAGI data sheets. The data presented in Tables 1 and 7 within this protocol could also be used to chart percentage of power versus percentage of capacity in a spreadsheet platform (MS Excel) and develop polynomial fit curves to better estimate part-load values as opposed to using lookup tables.

Step 6: Once the percentage of power and operating load for each CFM-bin have been determined, calculate the corresponding energy consumption using the product of the average compressor demand and the number of hours in each bin for the base and post cases (Equation 9). The sum of the kWh bin values gives the annual consumption (Equation 10).

$$\Delta kWh_{binN} = (\text{Base } kW_{operating\_binN} - \text{Post } kW_{operating\_binN}) \times \text{CFM-bin N H} \qquad (9)$$

where:

Base $kW_{operating\_binN}$ = baseline demand at part-load associated with CFM-bin N

Post $kW_{operating\_binN}$ = post demand at part-load associated with CFM-bin N

Total Energy Reduction:

$$kWh/yr = \sum [\ \Delta kWh_{bin1} + \Delta kWh_{bin2} + \ldots + \Delta kWh_{binN}\ ] \qquad (10)$$

where:

ΔkWh_{bin1} = energy reduction for CFM-bin 1

ΔkWh_{binN} = energy reduction for CFM-bin N

Using the data from our example project (summarized in Table 11) and Equation 9, the CFM-bin level energy reduction for each bin would be as follows:

$$\Delta kWh_{bin1} = (63.5 \text{ kW} - 60.1 \text{ kW}) \times 200 \text{ h} = 692 \text{ kWh}$$

$$\Delta kWh_{bin2} = (61.6 \text{ kW} - 54.0 \text{ kW}) \times 2{,}440 \text{ h} = 18{,}471 \text{ kWh}$$

$\Delta kWh_{bin3} = (57.6\ kW – 40.5\ kW) \times 170\ h \quad = 2,914\ kWh$

$\Delta kWh_{bin4} = (56.3\ kW – 35.8\ kW) \times 430\ h \quad = 8,839\ kWh$

$\Delta kWh_{bin5} = (53.7\ kW – 27.7\ kW) \times 1,100\ h \quad = 28,639\ kWh$

$\Delta kWh_{bin6} = (17.0\ kW – 0.0\ kW) \times 770\ h \quad = 13,090\ kWh$

$\Delta kWh_{bin7} = (0.0\ kW – 0.0\ kW) \times 4,420\ h \quad = 0\ kWh$

Table 11. Example Project Compressed-Air CFM-Bin Hour Table and Consumption - Base and Post-Retrofit Cases

		Base Case: Rotary Screw Compressor with Inlet Valve Modulation (w/Blowdown)				Post Case: VSD Rotary Screw Compressor w/Stopping			
CFM-bin #	**CFM Load Profile**	**Percentage of Power**	**H/Yr**	**Input Power (kW)**	**kWh**	**Percentage of Power**	**H/Yr**	**Input Power (kW)**	**kWh**
CFM-bin 1	324	97%	200	63.5	12,707	89%	200	60.1	12,015
CFM-bin 2	288	94%	2,440	61.6	150,231	80%	2,440	54.0	131,760
CFM-bin 3	216	88%	170	57.6	9,799	60%	170	40.5	6,885
CFM-bin 4	180	86%	430	56.3	24,222	53%	430	35.8	15,383
CFM-bin 5	144	82%	1,100	53.7	59,081	41%	1,100	27.7	30,443
CFM-bin 6	0 idling	26%	770	17.0	13,090	0%	0	0	0
CFM-bin 7	0 shutdown	0%	3,650	0.0	0.0	0%	4,420	0.0	0.0
Total kWh/yr					**269,153**				**196,486**

Using Equation 9 the Total Energy Reduction resulting from the example project would be:

Total Energy Reduction (kWh/yr) =

$= \sum_{0\text{-}7} [\Delta kWh_{bin1} + \Delta kWh_{bin2} + \Delta kWh_{bin3} + \Delta kWh_{bin4} + \Delta kWh_{bin5} + \Delta kWh_{bin6}]$

$= \sum_{0\text{-}7} [692 + 18,471 + 2,914 + 8,839 + 28,639 + 13,090 + 0]\ kWh$

$= 72,644\ kWh$

7 Sample Design

See Uniform Methods Project Chapter 11: Sample Design for guidance on designing samples to evaluate a program.

Confidence and precision levels are typically determined by specific regulatory or program administrator requirements. In most jurisdictions, evaluation samples should be designed to estimate operating hours and load profiles with a sampling precision of ±10% at the 90% confidence interval.

In addition to sampling errors, errors in measurement and modeling can also occur. In general, these measurement errors are lower than the sampling error; thus, sample sizes are commonly designed to meet sampling precision levels alone.

Sample sizes for achieving the required precision should be determined by estimating the coefficient of variation. These generally range from 0.5 to 1.06 for compressed-air measures, with lower values for more homogeneous populations.

7.1 Program Evaluation Elements

To ensure the validity of data collected, establish procedures at the beginning of the study to address the following issues:

- Quality of an acceptable regression curve fit (based on R^2, missing data, etc.).
- Procedures for filling in limited amounts of missing data.
- Meter failure (the minimum amount of data from a site required for analysis).
- High and low data limits (based on meter sensitivity, malfunction, etc.).
- Units to be metered not operational during the site visit; for example, determine whether this should be brought to the owner's attention or whether the unit should be metered as is.
- Units to be metered malfunction during the mid-metering period and have (or have not) been repaired at the customer's direction.

An additional 10% of the number of sites or units should be put into the sample to account for data attrition.

At the beginning of each study, determine whether metering efforts should capture short-term measure persistence. That is, decide how the metering study should capture the impacts of nonoperational rebated equipment (due to malfunction, equipment never installed, etc.). For non-operational equipment, these could be treated as equipment with zero operating hours, or a separate assessment of the in-service rate could be conducted.

7.2 Net-to-Gross Estimation

The cross-cutting chapter, *Estimating Net Savings – Common Practices*, discusses various approaches for determining net program impacts.

8 Looking Forward

VSD air compressor incentive offerings may become less common in the future as regional and state energy codes and standards begin to adopt minimum efficiency requirements similar to those already in effect in California via *Title 24 Energy Efficiency Standards for Residential and Nonresidential Buildings*. The 2019 version of *Title 24* requires that every newly installed compressed air system larger than 25 hp be equipped with at least one trim compressor that is efficient at part loads (i.e. has VSD control) and that compressed air systems with more than one compressor, and a combined capacity of greater than 100 hp, be equipped with a master controller that is capable of determining the most energy efficient combination of compressors to operate within the system based on current air demands.

However, VSD air compressors still remain a popular measure offering amongst commercial and industrial demand side management programs and have the potential to offer continued, significant savings in many jurisdictions for the foreseeable future.

Measurement and Verification Studies

The following evaluations are examples of studies that utilize the methodologies described in this protocol:

- Impact Evaluation of National Grid's 2014 Rhode Island Prescriptive Compressed Air Installations (DNV GL 2016)
- ComEd's Industrial Comprehensive Systems Studies Program – Implementation Contract – Nexant, Inc.
- Duke Energy Non-Residential Custom Program Impact Evaluation.

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