Estimating Energy Savings in Compressed Air Systems

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ESTIMATING ENERGY SAVINGS IN COMPRESSED AIR SYSTEMS

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ABSTRACT

Compressed air is typically one of the most expensive utilities in an industrial facility. As a result, potential savings opportunities are aggressively sought out and identified. Once identified, projected energy savings must be calculated in order to justify the cost of implementing the savings opportunity. It is important to calculate projected energy and cost savings as accurately as possible. Unfortunately, savings are frequently overestimated because the methods used to estimate savings neglect to consider important factors such as compressor control and type, storage, and multiple compressor operation.

In this paper, a methodology is presented for modeling air compressor performance and calculating projected energy savings from easily obtainable performance data such as full-load power, no-load power, rated capacity, average fraction full-load power or average fraction rated capacity. The methodology is applied in case study examples that illustrate the difference between estimating savings using this method and rule-of-thumb methods.

Introduction

Compressed air is typically one of the most expensive utilities in an industrial facility. As a result, potential savings opportunities are aggressively sought out and identified. Once identified, the projected energy savings must be calculated in order to justify implementation expenses. To do so, the performance of the compressor(s) must be determined under existing conditions, and compared to the predicted performance after the proposed changes. Too often, savings are predicted using rule-of-thumb methods that typically overestimate savings by neglecting the effects of compressor type, compressor control, storage, multiple compressor operation, etc.

In this paper, a methodology is presented for modeling compressor performance and calculating projected energy savings using four of the five following performance metrics: full-load power (FLP), no-load power (NLP), rated capacity (FLC), fraction full-load power (FP) or fraction rated capacity (FC). The methodology is applied in various examples to illustrate the difference between estimating savings using this method and rule-of-thumb methods.

Background

In a previous paper, (Schmidt and Kissock, 2003), relationships were developed between the fraction full-load power (FP) and fraction rated capacity (FC) for various modes of compressor control. These relationships are plotted in Figure 1. The relations were derived from data compiled from various sources including Compressed Air Challenge materials, Air Master+ software, manufacturer's literature, and our measurements and observations from over 50 compressed air systems analyses. The curves in Figure 1 are drawn as continuous functions since average fraction of compressed air output, FC, can be derived from the average fraction power, FP, over a time interval.

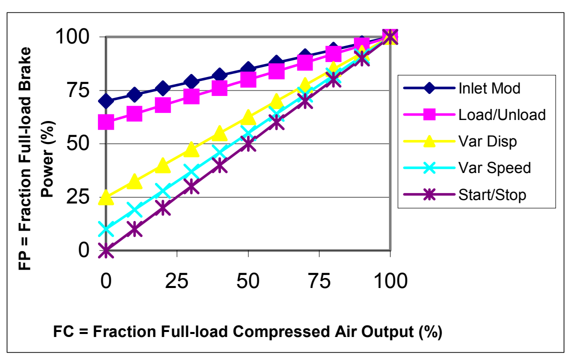

Figure 1. Fraction of full-load power versus fraction of full-load compressed air output for typical reciprocating and rotary air compressors based on type of control.

In all cases, compressed air output is greatest when the motor is fully loaded. However, as the demand for compressed air declines, the power requirement of the compressor depends on the type of control. Of the control methods shown in Figure 1, inlet-modulation control has the lowest part-load efficiency because it requires the greatest fraction of full-load power per compressed air output. Similarly, start/stop control has the best part-load efficiency because it requires the least fraction of full-load power per compressed air output. Part-load efficiency is important since most air compressors are sized for the peak load, which generally occurs infrequently, and thus run at part load most of the time.

The relationships in Figure 1 are functions of the fraction of full-load power that the compressor draws at no-load, or 0% capacity, (FPNL), and can be described by the following equation:

$$FP = [(FC \times (1 - FPNL)] + FPNL \tag{1}$$

The full-load power (FLP) to an air compressor is easily measured or calculated. If measuring power is not an option, there are a few different ways to calculate FLP. First, the current draw (A) can be measured or logged and then multiplied up by volts (V) and loaded power factor (PFL) to get power. If measuring current is not an option, typically the full-load current (FLA) and the motor power factor are listed on the compressor motor nameplate. Using these values, the FLP of the compressor can be calculated by the following equation:

$$FLP = V \times FLA \times PFL \times \sqrt{3} / 1{,}000 \tag{2}$$

It is important to note that full-load power to the compressor is generally from 105% to 120% of the input power expected based on the rated horsepower of the compressor motor.

Similarly, the no-load power (NLP) is defined as the power draw of the compressor when it is not generating air. Depending on the control type, compressors add no compressed to the system when unloaded or fully-modulated. In general, the NLP of an air compressor is also easily measured or calculated.

The average power (PAVG) is the average power drawn by a compressor over a target interval. The best method for determining the average power is to actually log the power or current draw of the compressor over the interval. However, the average power or current of a compressor in load/unload control can be calculated if the percent time loaded (PTL) is known using the following method:

$$PAVG = (FLP \times PTL) + [NLP \times (1 - PTL)]$$

Once the full-load (FLP), no-load (NLP) and average (PAVG) power have been determined, the fraction of full-load power at no-load (FPNL) and the average fraction full-load power (FP) can be calculated as:

$$FPNL = NLP / FLP$$
$$FP = PAVG / FLP$$

The rated or full-load capacity (FLC) of air compressors is typically listed on the compressor nameplate or in the compressor operating manual. If unavailable, the FLC can be estimated by applying a typical performance index of 4.2 scfm per brake horsepower (BHP).

Estimating Energy Savings in Compressed Air Systems

In order to quantify savings from changes in a compressed air system, it is necessary to quantify the performance of the existing compressor situation. To do so, an equation must be developed that describes the performance of the compressor in the current control mode. Next, the average power draw of the compressor under typical operating conditions must be measured. From this, the average fraction of rated capacity at which the compressor operates can be calculated. Both of these provide the baseline for which to compare the operation of the compressor after making the proposed changes.

Energy savings opportunities in compressed air systems generally result from one of three changes. The first change occurs when the average air demand remains constant but the compressor(s) is operated with more efficient control. The second change occurs when plant air demand is reduced with no changes to compressor control. The third change generates the largest savings and comes from implementing both of the previous changes. Methods for estimating energy savings for all three types of changes are presented below.

If the change is to the compressor control, the fraction of full-load power at no-load (FPNL) is reduced and the fraction rated capacity (FC) at which the compressor operates will remain constant. Based on FPNL and FC, an equation describing the compressor performance with more efficient control can be developed. From this equation, the average power of the compressor in the more efficient mode can be calculated and then compared to the baseline average power to quantify the energy savings.

If the change reduces plant air demand (PAD), the average fraction rated capacity (FC) at which the compressor operates is reduced and the fraction of full-load power at no-load (FPNL) will remain constant. Because the FPNL doesn't change, the compressor performance equation remains the same. Substituting the reduced FC into this equation, the average power of the compressor at the reduced plant air demand can be calculated and then compared to the baseline average power to quantify the energy savings.

If both types of changes are made, then both the FC and the FPNL would change and both of the above procedures would apply. In all cases, the energy savings are calculated as the difference between the average power of the compressor before and after the proposed changes.

The following methodology can be used to determine compressed air system performance and estimate potential energy savings. Depending on the type of change to the system, the number and order of steps used will vary.

The first step is to determine the full-load power (FLP) of the compressor; the full-load power remains constant regardless of changes in the system. Then, the no-load power of the compressor at the current operating conditions (NLP1) needs to be determined. Substituting these values into Equation (3), the fraction of full-load power at which the compressor operates at no-load in the current control mode (FPNL1) can be calculated by the following equation:

$$FPNL1 = NLP1 / FLP$$

The next step is to establish the fraction power (FP) vs. fraction capacity (FC) relationship for the compressor in the current operating conditions by substituting FPNL1 into Equation (1)

$$FP1 = [(FC1 \times (1 - FPNL1)] + FPNL1$$

Measuring the average power of the compressor while in the current control mode (PAVG1) provides a baseline from which to calculate savings. Additionally, the average fraction of full-load power (FP1) at which the compressor operates under current conditions can be calculated using Equation (4):

$$FP1 = PAVG1 / FLP$$

Substituting FP1 into Equation (6) and solving for FC, the average fraction of rated capacity (FC1) at which the compressor operates currently would be:

$$FC1 = (FP1 - FPNL1) / (1 - FPNL1)$$

Thus, based on the rated capacity (FLC) of the compressor, the average output, or average plant air demand (PAD), is determined to be:

$$PAD1 = FLC \times FC1$$

Change in Compressor Control:

If the compressor is switched to a more efficient mode of operation, the no-load power (NLP) of the compressor will be reduced. Hence, the no-load power of the compressor in the more efficient mode of control (NLP2) must be measured or estimated. Using Equation (3), the fraction of full-load power that the compressor would draw at no-load (FPNL2) would be:

$$FPNL2 = NLP2 / FLP$$

Because a change was made to the performance of the compressor, the equation describing its performance must change as well. Thus, substituting FPNL2 into Equation (1), the FP vs. FC relationship for the compressor in a more efficient control is:

$$FP2 = [(FC1 \times (1 - FPNL2)] + FPNL2$$

Previously, the fraction of rated capacity (FC) at which the compressor is operating was solved for. When switching control modes, the compressor must generate the same amount of air as before and therefore would operate at the same fraction capacity as the compressor under current conditions; that is FC2 = FC1. Thus, FP2 can be calculated by (5)
substituting FC1 into Equation (11).

Change in Air Demand:

If the change results in a reduced plant air demand (PAD2), the average fraction rated capacity (FC) is reduced. The average fraction rated capacity at which the compressor would operate (FC2) would be: (6)

$$FC2 = PAD2 / FLC$$

In this case, because the mode of control does not change, the fraction of full-load power at no-load (FPNL) would not change and thus FPNL2 = FPNL1. Thus, FP2 can be calculated by substituting FC2 into Equation (6).

(7)

Energy Savings:

In all cases, the energy savings are determined by the difference between the average power of the compressor before and after any system changes are made. Thus, to quantify energy savings, the average power of the compressor after the change (PAVG2) must be calculated. Substituting FP2 into Equation (4) and solving for PAVG2 gives:

$$PAVG2 = FP2 \times FLP \quad (13)$$

Thus, the reduction in the average power (PSAV) would be:

$$PSAV = PAVG1 - PAVG2 \quad (14)$$

Based on the annual operating hours (AOH) of the compressor and the average unit cost of electricity (COE), the annual electricity cost savings (ESAV) from operating the compressor in a more efficient control mode would be:

$$ESAV\ (\$/year) = PSAV\ (kW) \times AOH\ (hours/year) \times COE\ (\$/kWh) \quad (15)$$

Operating in Most Efficient Control Mode

Rotary compressors, which are the most common type in industry, are generally equipped to operate in two different modes of control: load/unload and modulation. Many compressors have a third mode that is some combination of the two; however, in general, the performance of a compressor can be described as operating in load/unload or modulation control. Moreover, for estimating savings, the most important measure associated with the type of control is the part-load efficiency.

A compressor in modulation control will draw between 60% and 85% of full-load power when fully-modulated – at no-load (0% capacity). On the other hand, a compressor in load/unload control will draw between 20% and 60% of full-load power when unloaded – at no-load (0% capacity). Thus, the part-load efficiency of load/unload control is always better than modulation control.

Furthermore, the majority of the compressors with load/unload control also have auto shutoff capabilities. With auto shutoff, if the compressor does not load within the auto shutoff time setting, it will automatically turn off until the demand increases again. Auto shutoff control can significantly reduce compressor energy use during periods of low demand such as during breaks, lightly-loaded shifts, or weekends.

All too often, compressors are operated in modulation control or with the auto shutoff control deactivated. This almost always results in unnecessarily high compressed air costs and wastes a significant amount of energy. In this section, we demonstrate a methodology for quantifying the savings from operating a compressor in a more efficient control mode and present two examples that illustrate the savings.

In order to quantify savings from operating a compressor in a more efficient control mode, the first step is to develop an equation describing the performance of the compressor in the current control mode. Next, the average power draw of the compressor under typical operating conditions must be measured. From this, the average fraction of rated capacity at which the compressor operates can be calculated. Both of these provide the baseline for which to compare the operation of the compressor after switching modes. Once the baseline conditions are determined, an equation describing the performance of the compressor in the more efficient control mode must be developed. Using the baseline fraction rated capacity, the average power of the compressor in the more efficient mode can be calculated. Energy savings are the difference between the average power draw of the compressor in each mode.

Example 1: Switching from Modulation to Load/Unload Control

A metal forming plant has a 60-hp rotary compressor in modulation control. The power draw of the compressor was measured over a 4.5 hour interval during production. Based on the rated full load amps from the compressor nameplate, the full-load power (FLP) was calculated to be 52 kW. The no-load power draw (NLP1) of the compressor was determined to be 42 kW. From Equation (5), the fraction of full-load power at no-load was:

$$FPNL1 = 42\ kW / 52\ kW = 81\%$$

Substituting FPNL1 into Equation (6) gives the relationship between fraction power, FP, and fraction capacity, FC, in modulation control:

$$\begin{aligned} FP1 &= [(FC \times (1 - FPNL1)] + FPNL1 \\ &= (FC1 \times 0.19) + 0.81 \end{aligned}$$

The average power draw of the compressor in modulation control (PAVG1) was measured to be about 47 kW. Thus, from Equation (7), the average fraction of full-load power (FP1) at which the compressor operates in modulation control was about:

$$FP1 = PAVG1 / FLP = 47\ kW / 52\ kW = 90\%$$

Substituting FP1 and FPNL1 into Equation (8), the average fraction of rated capacity (FC1) at which the compressor operates was about:

FC1 = (FP1 - 0.81) / 0.19 = (0.90 - 0.81) / 0.19 = 47%

Based on previous measurements of this model compressor, the fraction of full-load power when unloaded (FPNL2) is about 55%. Thus, substituting FPNL2 into Equation (11), the FP vs. FC relationship for this compressor in load/unload control was:

FP2 = [(FC1 x (1 – FPNL2)] + FPNL2
= (FC1 x 0.45) + 0.55

Using the average fraction of rated capacity (FC1) of 47% calculated above, the average fraction of full-load power (FP2) at which the compressor would operate in load/unload control would be about:

FP2 = (FC x 0.45) + 0.55 = (0.47 x 0.45) + 0.55 = 76%

From Equation (13), the average power at which the compressor would operate (PAVG2) would be about:

PAVG2 = FP2 x FLP = 52 kW x 76% = 40 kW

The compressor operated for about 4,080 hours per year and the average unit cost of electricity for the facility was \$0.07 /kWh. Thus, using Equations (14) and (15), the annual electricity cost savings from operating the compressor in load/unload control would be about:

PSAV = 47 kW – 40 kW = 7 kW
ESAV = 7 kW x 4,080 hr/yr x \$0.07 /kWh = \$2,000 /yr

The percent reduction in energy use and costs would be about:

7 kW / 47 kW = 15%

Example 2: Switching from Modulation to Load/Unload Control with Auto Shutoff

A metal stamping plant has four 100-hp rotary compressors of the same model. All four of the compressors were operating in modulation control without auto shutoff. Two of the compressors were running at close to full load power while the other two were modulating between full-load and no-load power as the air demand fluctuated. The compressors are equipped with load/unload control with auto shutoff. However, facility personnel were unaware of this and the potential savings from operating the compressors in load/unload control.

It was recommended that at least one of the compressors, beginning with the #4 trim compressor, be switched to load/unload control with auto shutoff. The power draw of each compressor was logged over a 14 hour period to monitor their performance as production changed.

During the first seven hours, the #4 compressor was run in modulation control and then was switched to run in load/unload control with auto shutoff during the second seven hours. Compressors #1, #2 and #3 remained in modulation control over the entire period. Figure 2 below shows the logged data of both the #3 and #4 compressors over the two shifts.

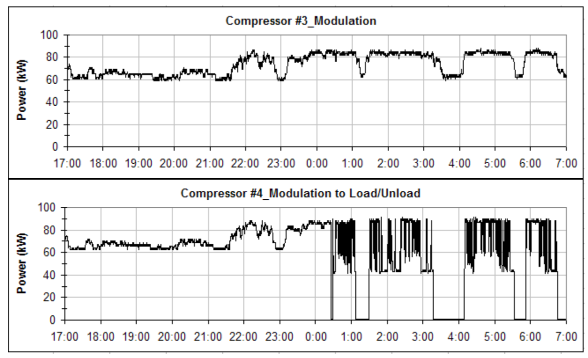

Figure 2. Comparison of compressor power draw between modulation control and load/unload with auto shut-off.

It is evident that both compressors performed virtually the same up until the point when the control mode is switched. Thus, it is reasonable to compare the average power of the #4 compressor in load/unload control to the average power of the #3 compressor in modulation control over the same period to estimate savings.

It can be seen from the figures that during the second period, there were four significant drops in power draw indicating a reduction in air demand. These are coincident with the personnel breaks in the facility. It is extremely important to note that during the low demand periods, compressor #3 fully-modulated, generating no air while drawing about 70% (62 kW / 88 kW) of full-load power. Meanwhile, compressor #4 automatically shutoff during these periods, drawing 0 kW.

The average logged power draw of compressor #3 from 0:00 to 7:00 is 79 kW and that of compressor #4 is 51 kW. The compressors operate for about 7,200 hours per year. The average unit cost of electricity for the facility is \$0.072 /kWh. Thus, using

Equations (14) and (15), the annual electricity cost savings from operating one of the four compressors in load/unload control with auto shutoff would be about:

PSAV = 79 kW – 51 kW = 28 kW
ESAV = 28 kW x 7,200 hr/yr x \$0.072 /kWh
= \$14,515 /yr

The percent reduction in energy use and costs would be about:

28 kW / 79 kW = 35%

Summary of Switching from Modulation to Load/Unload Control with Auto Shutoff

These examples shows how significant savings can be attained by operating compressors in load/unload mode with auto shut-off control compared to modulation mode. In our view, the primary advantage of operating in modulation control is that the discharge pressure of the compressor generally remains within +/- 2 psig of the target pressure, resulting in a more constant line pressure. While this may be important for some facilities, it is quite costly. In general, we suggest installing additional compressed air storage to dampen compressed air pressure swings rather than operating compressors in modulation mode at plants that desire small pressure changes. The one time cost of adding additional compressed air storage is generally much less than the continual additional cost of operating compressors in modulation mode.

FIXING LEAKS

In our experience, air leaks typically represent from 10% to 75% of compressed air demand. Leaks increase the base load on the compressors, which increases average power consumption and electricity costs. In addition, air leaks contribute to plant noise levels, contributing to an uncomfortable and unsafe work environment. Two methods for estimating the amount of air lost through leaks are discussed below.

Estimating Compressed Air Losses By Logging Compressor Power

If a compressor is left running after production has ended, it will continue to generate air to make up for the air lost through leaks. By logging the power of the compressor, and applying the methodology developed in this paper, the quantity of compressed air lost to leaks can be calculated. The following example demonstrates the use of the method proposed here to estimate compressed air losses through leaks by logging compressor power.

A plant had two 150-hp rotary screw compressors and one 250-hp reciprocating compressor. Production in the plant ended at 3:30 pm and the compressors were shut off. However, one of the compressors was turned back on in order to estimate the leak load.

The logged power draw of the 250-hp reciprocating compressor is shown in Figure 3 below. The logged interval includes its performance during production before it was shut off and then from 4:00 pm to about 4:17 pm when all production in the plant had ended.

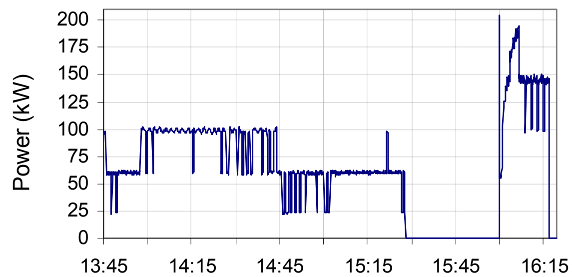

Figure 3. Compressor power draw during plant shutdown showing leak load.

The compressor operates with five-step unloading control, allowing it to operate at 100%, 75%, 50%, 25% or 0% capacity (no load). According to the logged power data, the no-load power draw (NLP) of the compressor is 23 kW and the full-load power (FLP) is 200 kW. Using Equation (5), the fraction of full-load power at no-load (FPNL) is:

FPNL = 22 kW / 200 kW = 11%

Substituting FPNL into Equation (6), the FP vs. FC relationship for this compressor is:

FP = [(FC x (1 – FPNL)] + FPNL = (FC x 0.89) + 0.11

The logged data indicate that the average power draw of the compressor (PAVG) over the leak testing interval was about 142 kW. Thus, from Equation (7), the fraction of full-load power (FP) at which the compressor operated on average was about:

FP = PAVG / FLP = 142 kW / 200 kW = 71%

Substituting FP and FPNL into Equation (8), the fraction of rated capacity (FC) at which the compressor operated over the leak interval was about:

FC = (0.71 – 0.11) / 0.89 = 67%

Thus, using Equation (9), the average compressed air output of the 1,600 cfm rated compressor when the plant was shutdown was about:

PAD = 67% x 1,600 cfm = 1,072 cfm

Therefore, the compressed air demand due to leaks and other unnecessary consumption was about 1,072 cfm. Based on this number, we estimated that one of the plant's 150-hp air compressors could be completely turned off if the major leaks were fixed. After receiving the report and fixing the leaks, management reported that, as predicted, they were able to shut off one of the 150-hp compressors that normally runs fully loaded.

Estimating Compressed Air Losses From Individual Leaks

Another approach for estimating the amount of air lost due to leaks is to inspect the plant for leaks using an ultrasonic sensor or simply by listening with an unaided ear. Once identified, the rate of compressed air flow from a leak, Q (scfm), can be calculated using the Moss Equation (Ingersoll-Rand Condensed Air Data, 1988).

$$Q = 0.5303 \text{ x } \pi / 4 \text{ x } [D \text{ (in)}]^2 \text{ x } C \text{ x } P \text{ (psia) x } 60 / [\sqrt{530(R)} \text{ x } 0.07494 \text{ lb/ft3}] \quad (16)$$

where D is the leak diameter, P is absolute pressure of the compressed air near the leak, and the coefficient of flow, C, is a function of the roughness of the hole through which the air is discharging. Most leaks are irregular in shape and have rough edges covered with grime and fluids. According to Ingersoll-Rand Compressed Air Data, a value for C of 0.61 is reasonable for leaks. For example, the rate of compressed air flow through a 1/16-inch rough-edged (C=0.61) leak at an average plant line pressure of 100 psig is about:

$$Q = 0.4165 \text{ x } (1/16")^2 \text{ x } 0.61 \text{ x } (100 + 14.7) \text{ psia x } 60 / [\sqrt{530(R)} \text{ x } 0.07494 \text{ lb/ft3}] = 3.96 \text{ scfm/leak}$$

Quantifying Energy Savings From Fixing Leaks

Once the leakage rate has been determined, the energy savings from fixing air leaks can be calculated as difference between the compressor power draw at the different levels of compressed air demand. The compressor power draw at different levels of compressed air demand is determined by the control mode. Hence, actual energy savings are highly dependent on the control mode of the compressor.

Unfortunately, the most common method of estimating savings from fixing leaks neglects this important consideration. This rule-of-thumb method simply assumes that all energy consumed to generate the air lost through leaks would be saved if the leaks were fixed. In reality, this is true only for compressors that shut completely off when unloaded; for the majority of compressors, this method will grossly overestimate savings. To illustrate the importance of the compressor control mode when calculating savings from fixing leaks, consider the following examples.

<u>Savings From Fixing Leaks Using Rule-of-Thumb Method</u>

Consider a plant has an estimated leak load of 70 cfm and a 60-hp compressor that generates 4.2 cfm per brake horsepower of work delivered to the compressor by the motor. The company pays $0.07 /kWh, and the compressor runs 4,080 hours per year. Using the rule-of-thumb method, and assuming the motor is 90% efficient, the electricity cost savings would be about:

ESAV = [(70 cfm / 4.2 cfm/hp) x 0.75 kW/hp / 90%] x 4,080 hr/yr x $0.07 /kWh = $3,945 /year

Savings From Fixing Leaks If Compressor Operates in Modulation Control Mode

Now consider if the 60-hp compressor was running in modulation control. In the *Switching from Modulation to Load/Unload Control* example, the average fraction of rated capacity (FC1) at which the same 60-hp compressor operated was 47%. The rated full-load capacity (FLC) of the 60-hp compressor was 265 cfm. Thus, from Equation (9), the average output of the compressor or average plant air demand under the current conditions (PAD1) was about:

PAD1 = FLC x FC1 = 265 cfm x 47% = 125 cfm

If the 70 cfm of leaks were fixed, the average air demand (PAD2) would be reduced to 55 cfm (125 – 70). Substituting into Equation (12), the average fraction of rated capacity at which the compressor would then operate (FC2) would be about:

FC2 = PAD2 / FLC = 55 cfm / 265 cfm = 21%

Substituting FC2 into Equation (6), the average fraction of full-load power at which the compressor would operate (FP2) would be about:

FP2 = (FC2 x 0.19) + 0.81 = (0.21 x 0.19) + 0.81 = 85%

The full-load power (FLP) of the 60-hp compressor is 52 kW. Thus, from Equation (13), the average power at which the compressor would operate if the leaks were fixed (PAVG2) would be about:

PAVG2 = 52 kW x 85% = 44.2 kW

The average power draw of the compressor currently is 47 kW. Thus, using Equations (14) and (15), the annual electricity savings would be about:

PSAV = 47 kW – 44.2 kW = 2.8 kW
ESAV = 2.8 kW x 4,080 hr/yr x $0.07 /kWh = $800 /yr

The percent reduction in energy use and costs would be about:

2.8 kW / 47 kW = 6%

Note that the "rule-of-thumb" method overestimated savings by about a factor of five.

Savings From Fixing Leaks If Compressor Operates in Load/Unload Mode

If the same 60-hp compressor were running in load/unload control, the savings would be significantly greater than in modulation control, but would still not approach the rule-of-thumb estimate. In the *Switching from Modulation to Load/Unload Control* example, the FP vs. FC relationship for this compressor in load/unload control is:

FP = [(FC x (1 - FPNL2)] + FPNL2 = (FC x 0.45) + 0.55

In the *Modulation Control Savings* example above, the average fraction of rated capacity at which the 60-hp compressor would operate with the leaks fixed (FC2) was calculated to be 21%. Substituting FC2 into Equation (6), the average fraction full-load power at which the compressor would operate in load/unload control and with the leaks fixed (FP2) would be about:

FP2 = (FC2 x 0.45) + 0.55 = (0.21 x 0.45) + 0.55 = 65%

Thus, from Equation (13), the average power at which the compressor would operate in load/unload control and with the leaks fixed (PAVG2) would be about:

PAVG2 = 52 kW x 65% = 33.8 kW

In the *Switching from Modulation to Load/Unload Control* example, the average power of the compressor in load/unload control under current conditions was calculated to be about 40 kW. Thus, using Equations (14) and (15), the electricity savings would be about:

PSAV = 40 kW – 33.8 kW = 6.2 kW
ESAV = 6.2 kW x 4,080 hr/yr x $0.07 /kWh
= $1,770 /year

The percent reduction in energy use and costs would be about:

6.2 kW / 40 kW = 16%

Therefore, because of the better part-load efficiency, operating the compressor in load/unload control more than doubles the savings from fixing the air leaks. However, in spite of this improvement, the savings still are less than 50% of those from the rule-of-thumb method.

INSTALLING AIR-SAVER NOZZLES

Many plants have applications that require a constant stream of compressed air for the removal of material, for cooling or drying, etc. Typically this stream of air is delivered through a metal or plastic tube or pipe at the plant line pressure, discharging air at a high force and flow rate. In some cases, the application does not necessarily require a strong force, but rather a large volume of cool, clean, dry air. In such cases, an air-saver nozzle can be used to reduce the amount of compressed air flow while amplifying the volume of air directed at the application. Air-saver nozzles can result in significant energy savings by reducing the plant air demand and allowing the compressor to run at a lower average load.

To demonstrate how to calculate savings from installing air saver nozzles, consider the following example. A plant uses two 1/4-inch inside-diameter copper tubes to dry off the edges of a material as it exits a wash bath. The nozzles continuously discharge compressed air at a pressure of about 110 psig. The compressed air consumption of the tubes is estimated using the Moss equation (Equation 16). The rate of compressed air flow Q (scfm) through one of the 1/4-inch relatively smooth-edged (C=0.9) tubes at pressure of 110 psig is about:

$$Q = 0.5303 \times \pi / 4 \times [D\ (in)]^2 \times C \times P\ (psia) \times 60 / [\sqrt{530(R)} \times 0.07494\ lb/ft3]$$
$$= 0.4165 \times (1/4")^2 \times 0.9 \times (110 + 14.7)\ psia \times 60 / [\sqrt{530(R)} \times 0.07494\ lb/ft3]$$
$$= 102\ scfm/tube$$

Based on data from various manufacturers of air-saver nozzles, a ¼-inch high-thrust air-saver consumes about 32 scfm of compressed air per nozzle. If so, the reduction in the average plant compressed air demand from installing air-saver nozzles on both tubes would be about:

2 tubes x (102 – 32) scfm/tube = 140 scfm

A 100-hp rotary compressor in load/unload control provides air for the plant. The power draw of the compressor was measured over a 4.5 hour interval during production. Based on the logged power data, the full-load power (FLP) of the compressor is 91 kW and the unloaded, or no-load power draw (NLP1), of the compressor was determined to be 51 kW. From Equation (5), the fraction of full-load power at no-load is:

FPNL1 = 51 kW / 91 kW = 56%

Substituting FPNL1 into Equation (6) gives the relationship between fraction power, FP, and fraction capacity, FC, in modulation control:

FP1 = [(FC x (1 - FPNL1)] + FPNL1
= (FC1 x 0.44) + 0.56

The average power draw of the compressor (PAVG1) was measured to be about 83 kW. Thus, from Equation (7), the average fraction of full-load power (FP1) at which the compressor operates is about:

FP1 = PAVG1 / FLP = 83 kW / 91 kW = 91%

Substituting FP1 and FPNL1 into Equation (8), the average fraction of rated capacity (FC1) at which the compressor operates is about:

FC1 = (FP1 – 0.56) / 0.44 = [0.91 – 0.56] / 0.44 = 80%

The rated full-load capacity (FLC) of the 100-hp compressor is 450 scfm. Thus, from Equation (9), the average output of the compressor, or average plant air demand, under the current conditions (PAD1) is about:

PAD1 = FLC x FC1 = 450 scfm x 80% = 360 scfm

If the air-saver nozzles were installed, the average air demand would be reduced by 140 scfm. Thus, the average plant air demand with the nozzles installed (PAD2) would be about:

PAD2 = PAD1 – 140 scfm
= 360 scfm – 140 scfm = 220 scfm

Substituting PAD2 into Equation (12), the average fraction of rated capacity at which the compressor would then operate (FC2) would be about:

FC2 = PAD2 / FLC = 220 cfm / 450 cfm = 49%

Substituting FC2 into Equation (6), the average fraction of full-load power at which the compressor would operate (FP2) would be about:

FP2 = (FC2 x 0.44) + 0.56 = (0.49 x 0.44) + 0.56 = 78%

Thus, from Equation (13), the average power at which the compressor would operate with air-saver nozzles installed (PAVG2) would be about:

PAVG2 = FLP x FP2 = 91 kW x 78% = 71 kW

Substituting the average power draw of the compressor before (PAVG1) and after the change (PAVG2) into Equation (14), the reduction in average power (PSAV) would be about:

PSAV = PAVG1 – PAVG2 = 83 kW - 71 kW = 12 kW

The compressor runs for 6,000 hours per year (AOH) and the facility pays $0.036 per kWh (COE). Substituting these values along with PSAV into Equation (15), the annual electricity savings would be about:

ESAV = 12 kW x 6,000 hr/yr x $0.036 /kWh
= $2,592 /year

The percent reduction in energy use and costs would be about:

12 kW / 83 kW = 14%

Summary and Conclusions

In this paper a methodology was developed for characterizing the performance of an air compressor using the following metrics: full-load power (FLP), no-load power (NLP), rated capacity (FLC), average fraction full-load power (FP) or average fraction rated capacity (FC). The methodology was then extended to show how to estimate energy savings from common changes in compressed air systems. Finally, the methodology was applied to actual case studies to demonstrate its use for estimating savings.

The case studies illustrate that energy savings of between 6% and 35% are achievable from operating compressors in load/unload control mode, fixing leaks, and using air-saver nozzles. In addition, the case studies showed that estimating savings from reducing compressed air demand using rule-of-thumb methods can overestimate savings by five times if the compressor operates in modulation mode and by two times if it operates in load/unload mode.

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