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Optimizing Compressed Air Storage for Energy Efficiency

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Optimizing Compressed Air Storage for Energy Efficiency

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ABSTRACT

Compressed air storage is an important, but often misunderstood, component of compressed air systems. This paper discusses methods to properly size compressed air storage in load-unload systems to avoid short cycling and reduce system energy use. First, key equations relating storage, pressure, and compressed air flow are derived using fundamental thermodynamic relations. Next, these relations are used to calculate the relation between volume of storage and cycle time in load-unload compressors. It is shown that cycle time is minimized when compressed air demand is 50% of compressor capacity. The effect of pressure drop between compressor system and storage on cycle time is discussed. These relations are used to develop guidelines for compressed air storage that minimize energy consumption. These methods are demonstrated in two case study examples.

INTRODUCTION

In the United States, nearly 90 billion kWh are consumed each year by compressed air systems at a cost of about $4.5 billion [Lawrence Berkeley National Laboratory, 1999]. Compressed air storage is an important, but often misunderstood, component of these compressed air systems. System storage is especially important in systems with compressors that operate in load/unload operation where the compressor cycles between periods of compressed air generation and idling. This paper establishes the relationship of storage, pressure, and compressed air flow through fundamental thermodynamic relations. Theoretical minimum cycle time is calculated to be 50% of compressor capacity. The effects of blowdown time and pressure drops before storage are discussed in relation to cycle time and energy use. These relations are used to develop guidelines for compressed air storage that minimize energy consumption. Other considerations that affect proper storage sizing are also discussed. The use of these relations and guidelines are demonstrated with two examples from the industrial sector.

UTILIZING COMPRESSED AIR STORAGE FOR ENERGY MINIMIZATION

FUNDAMENTAL STORAGE RELATIONS

Consider a compressed air system operating in load-unload mode between two pressures, P1 and P2. The system contains a storage tank of volume V, and air with mass, m and temperature, T. According to the ideal gas law, the mass in the tank is described by Equation 1.

$$m_i = \frac{P_i V}{R T_i} \qquad (1)$$

A mass balance on the system yields Equation 2, where ρ is the density of the air and V_{net} is net volume change of air, each at atmospheric conditions. During load cycle, the volume of air entering the storage tank can be described as the difference between the volume of air the compressor produces during load operation, V_L, and the volume of compressed air consumed by the plant V_D. During unload operation the change in volume is described solely by the air leaving the tank, V_D. Therefore, over a given time, Δt, the change in mass can be describes as Equation 3.

$$m = V_{net}\rho \qquad (2) \qquad\qquad \Delta m = V_{net}\rho\Delta t \quad (3)$$

The change in mass can also be described as the difference in the mass at states 1 and 2, as shown in Equation 4 below:

$$\Delta m = V_{net}\rho\Delta t = m_2 - m_1 = \frac{P_2 V}{RT_2} - \frac{P_1 V}{RT_1} \qquad (4)$$

As air enters the tank, flow work is converted to heat and the temperature rises. However, over time heat is transferred from the tank. Therefore it is assumed that the difference in temperature between states 1 and 2 is negligible. Consequently, factoring out constants in Equation 4 and solving for storage volume yields Equations 5 and 6.

$$V_{net}\rho\Delta t = \frac{V}{RT}\ P_2 - P_1 \quad (5) \qquad\qquad V = \frac{V_{net}\rho\Delta tRT}{(P_2 - P_1)} \qquad (6)$$

In order to simplify Equation 6, the ideal gas law is once again utilized to solve for RT at atmospheric conditions. Because the state is described in terms of atmospheric conditions, the pressure can be written as the standard atmospheric pressure, or P_s. Then, substituting into Equation 6 and displaying air density as mass of air over storage volume yields Equation 7.

$$V = \frac{V_{net}\rho\Delta t}{P_2 - P_1}\ \frac{P_s V}{m} = \frac{V_{net}\Delta t P_s V}{P_2 - P_1\ m}\ \frac{m}{V} \qquad (7)$$

Thus, both mass and volume cancel to produce Equation 8. Next, solving for the change in time and displaying the difference between P_2 and P_1 as ΔP yields Equation 9.

$$V = \frac{V_{net}\Delta t P_s}{P_2 - P_1} \qquad (8) \qquad\qquad \Delta t = \frac{V\Delta P}{V_{net}P_s} \qquad (9)$$

It is seen that load and unload cycle time depends upon four main components: first, the volume of storage in the system; second, the pressure band between which the compressor operates; third, both the amount of compressed air produced by the compressor and demanded by the plant (as they comprise the change in atmospheric volume of air in the system) and finally, the atmospheric pressure.

CYCLE TIME AND VOLUME OF STORAGE:

In order to analyze the effect system storage volume has on compressor cycle time, it is helpful to break total cycle time into two components: the time the compressor spends in each load and unload operation. Therefore, by defining total cycle time to include both load time, Δt_L, and unload time, Δt_U, and factoring out constants, Equation 9 can be rewritten as Equation 10 below.

$$\Delta t_{tot} = \Delta t_L + \Delta t_U = \frac{V\Delta P}{P_s}\left(\frac{1}{V_L} + \frac{1}{V_U}\right) \qquad (10)$$

During unload operation the net change in volume of the tank is the plant demand, or the maximum output of the compressor, V_m, multiplied by the fraction capacity at which the compressor is operating, FC, as shown in Equation 11. During load, the net change in volume is the maximum volume output of the compressor subtracted by the plant demand, as shown in Equation 12.

$$V_U = V_m * FC \qquad (11) \qquad\qquad V_L = V_m(1 - FC) \quad (12)$$

Substituting Equations 11 and 12 into Equation 10 and factoring out the maximum output of the compressor, V_m, the total cycle time can be described as:

$$\Delta t_{tot} = \frac{V\Delta P}{P_s}\ \frac{1}{V_m(1-FC)} + \frac{1}{V_m * FC} = \frac{V\Delta P}{P_s V_m}\ \frac{1}{(1-FC)} + \frac{1}{FC} \qquad (13)$$

MINIMUM CYCLE TIME:

In order to find the minimum cycle time, the first derivative of Equation 13 was taken and set equal to zero. First, however, to simplify derivation Equation 13 was multiplied by the multiplicative identity, (1-FC)/(1-FC). The result is shown in Equations 14 and 15 below.

$$\Delta t_{tot} = \frac{V\Delta P}{P_s V_m(1-FC)} \quad \frac{(1-FC)}{(1-FC)} + \frac{(1-FC)}{FC} \quad = \frac{V\Delta P}{P_s V_m(1-FC)} \quad \frac{(1-FC)}{(1-FC)} + \frac{1}{FC} - \frac{FC}{FC} \qquad (14)$$

$$\Delta t_{tot} = \frac{V\Delta P}{P_s V_m \; 1-FC} \quad \frac{1}{FC} \quad = \frac{V\Delta P}{P_s V_m \; FC-FC^2} \qquad (15)$$

The equation can be simplified further by letting k equal the constants shown in Equation 16 to form Equation 17:

$$k = \frac{V\Delta P}{P_s V_m} \qquad (16) \qquad\qquad \Delta t_{tot} = \frac{k}{FC-FC^2} \quad (17)$$

In order to find minima, the derivative of the equation is set to zero. By employing the power rule Equation 18 is formed. Solving for fraction capacity, FC, it is found that the critical point of one half of full capacity causes the total cycle time to be minimized.

$$\frac{d}{d\,FC} \; k \;\; FC - FC^2 \;\; ^{-1} \;\; = -k \;\; FC - FC^2 \;\; ^{-2} * (1 - 2FC) = 0 \qquad (18)$$

SIZING COMPRESSED AIR STORAGE FOR EFFECTIVE PRESSURE BAND:

When performing sizing calculations, one aspect that can seriously affect results is the presence of a pressure drop due to friction between the air compressor and the main compressed air storage [Beals, 2009]. The figure below shows a compressor system with a refrigerated dryer located downstream of a combination of centrifugal and load-unload compressors. A filter is also located downstream of the dryer. Pressure was logged upstream of the dryer (1), between the dryer and filter (2), and downstream of the filter (3).

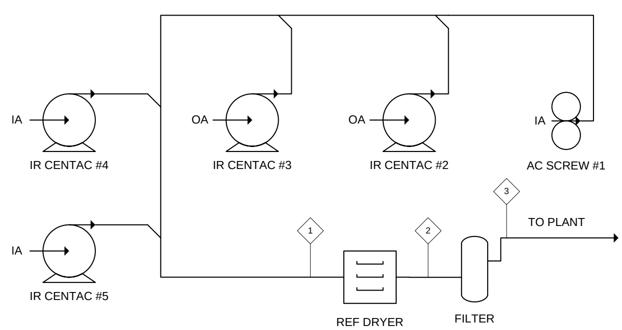

Figure 1 – Logged pressure locations

Figure 2 shows recorded pressure data when only the load-unload screw compressor was in operation. During load operation, the pressure in the system is increasing and compressed air flows through the dryer and filter. Friction causes the pressure loss across the dryer and filter. During unload operation, no air is being produced, and the pressure in the system decreases. There is no pressure drop due to friction across the dryer and filter because no air is flowing. Hence the pressures before the dryer, after the dryer, and after the filter, are equal.

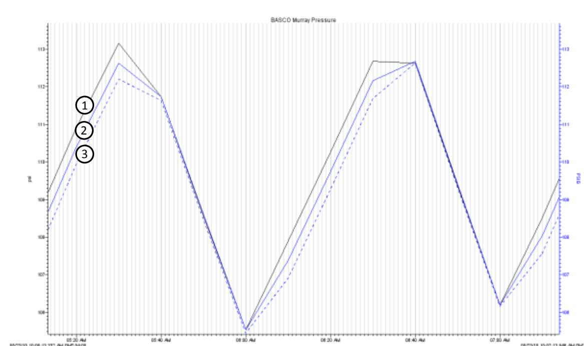

Figure 2 – Logged pressure of load-unload operation

The pressure drop due to friction increases with air flow volume. Figure 3 shows pressure from the same locations, when a centrifugal compressor is operating in addition to the load/unload compressor. In this case, friction pressure drop exists during both load and unload operation, and the pressure drops are larger because the volume flow rate is larger.

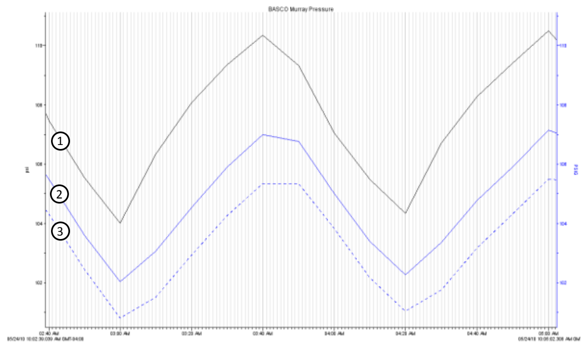

Figure 3 – Logged pressure draw at high flow

These pressure friction drops in the dryer and filter influence the effective pressure band of load/unload compressors. To demonstrate this, consider a compressed air system operating in load-unload between the pressure set-points of 100 and 110-psig. If no pressure drop existed between the compressor and the system storage, the pressure profile that the compressor experiences would follow Figure 4. When compressed air demand decreases system pressure to the set point of 100 psig, the compressor loads. Load operation is maintained until system pressure rises to the set-point of 110-psig. The compressor then unloads until the minimum pressure set-point is once again experienced.

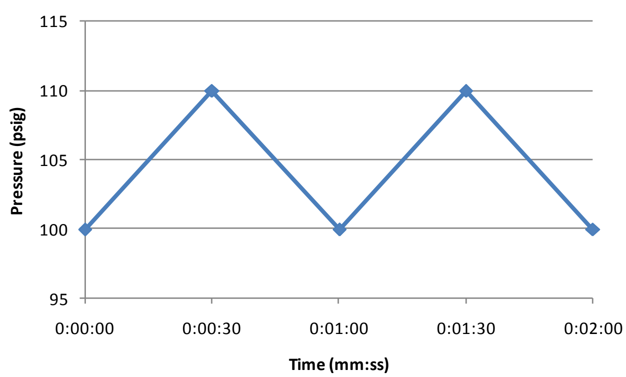

Figure 4 – Ideal compressor pressure pattern

Next, consider the same system with a 5 psig friction pressure drop during load conditions between the compressor and system storage. This pressure at the outlet of the compressor is depicted in Figure 5. When the pressure at the compressor drops to 100 psig, the compressor loads and starts generating compressed air. Because of the friction between the compressor and storage tank, the compressor quickly builds pressure to 105 psig before compressed air will flow into the system. The compressor then continues to run loaded until the set-point pressure of 110 psig is achieved, at which point the compressor unloads. In unload condition, flow stops and the friction pressure drop disappears. Hence, the pressure at the compressor outlet quickly drops to 105 psig. The system continues to operate unloaded until compressed air demand causes the pressure at the outlet of the compressor to drop to 100 psig, at which point the cycle is repeated.

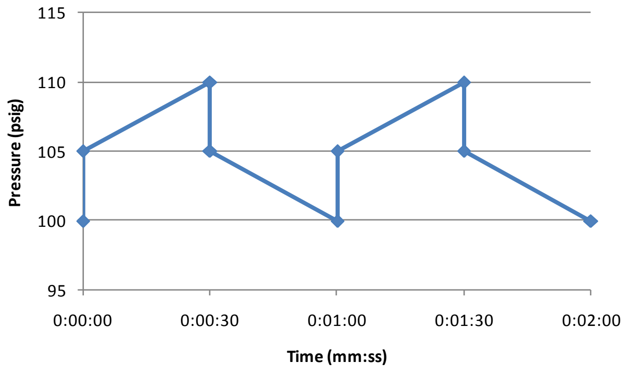

Figure 5 – Compressor pressure pattern with 5 psig drop before storage

Because of this effect, the compressor loads between 105 and 110 psig, over a pressure band of 5 psi. Similarly, the compressor unloads between 105 and 100 psig, over a pressure band of 5 psi. Thus, the effective pressure band is 5 psi instead of the 10 psi settings of the compressor. This effective pressure band, and the volume of compressed air storage, influence the rate of cycling between load and unload operation. Thus, it is important to use the effective pressure band at the compressor outlet when sizing compressed air storage to avoid short cycling.

REQUIRED MINIMUM CYCLE TIME:

When a lubricated rotary screw compressor unloads, pressure in the air and oil sump tank after the screw is slowly reduced to atmospheric pressure. The time required for the pressure in the air and oil sump tank to reach atmospheric pressure and the power draw to decline to full unload power is called blowdown time. This blowdown process generally takes about 30 to 90 seconds. During blowdown the screw pushes against less and less pressure, and the unloaded power draw decreases accordingly. When the pressure in the sump is at atmospheric pressure, a constant power draw great enough to overcome friction in the screws is established. This pattern is depicted in Figure 6 below. The total unload time is 60 seconds and the blowdown time is 30 seconds.

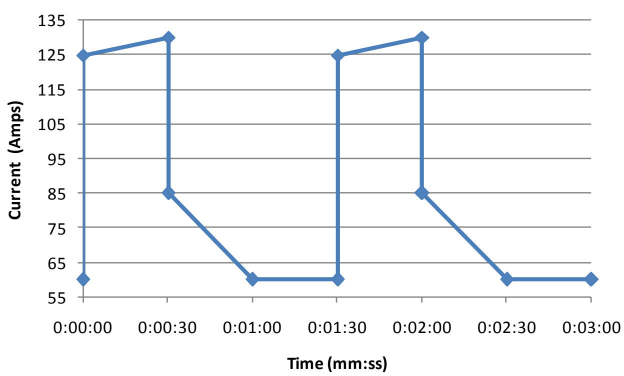

Figure 6 – Current draw with 30 second blowdown

If the unload time is less than the blowdown time, the compressor never achieves minimum power draw and will consume a higher average power. Figure 7 depicts the current draw of a compressor with a 60 second blowdown time at FC = 0.5, such that the load and unload cycle times are of equal length. The figure shows the current draw with storage amount such that the total cycle time is (A) 60 seconds long and (B) 120 seconds long.

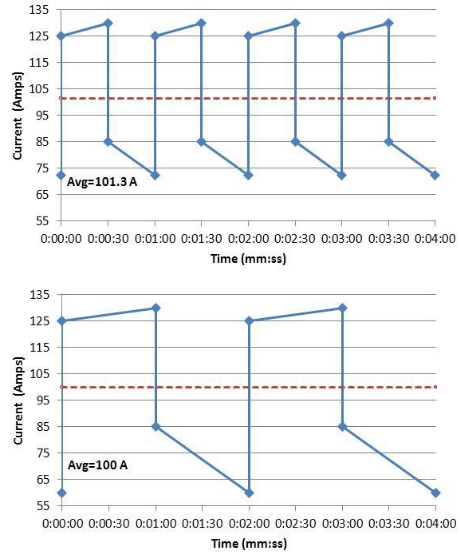

Figure 7 – Current draw for cycle times (A) 60 seconds, and (B) 120 Seconds

In Figure 7 (A) the blowdown time is cut short and the system never reaches minimum amperage draw. To decrease energy consumption and reduce wear on the system, compressed air storage should be sized so that when the compressor is operating at 50% fraction capacity the unloaded cycle time is greater than that of the time it takes the blowdown process to complete. This effectively ensures that at no point in its operation will the compressor cycle faster than it can fully blow down. For non-lubricated compressors storage should be sized to follow manufacture's guidelines for minimum cycle time to reduce wear on the system.

In order to determine an asymptotic point of minimum power draw a theoretical system was modeled using the compressor system simulation software, AirSim [Kissock 2003]. The system comprised a 100 hp load-unload flooded compressor with a 60 second blowdown time producing 4.5 scfm/hp. The system was modeled over a range of storage for a logged period of 150 minutes. It is shown that compressed air storage follows a pattern of diminishing returns. Savings are achieved by allowing the system to fully blow down but increased savings are still achieved with additional storage because the system enters the blowdown process less often. These results are shown in Figure 8.

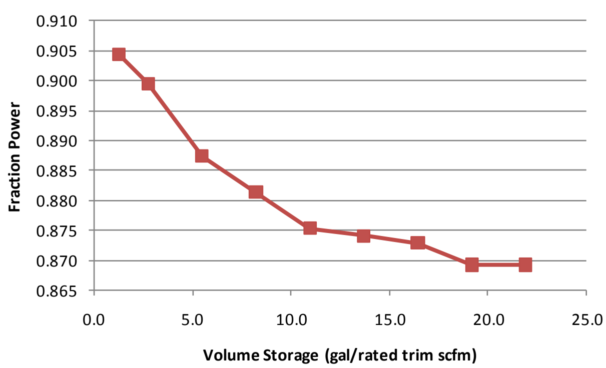

Figure 8 – Average fraction power per storage ratio at FC = 0.5.

SIZING GUIDELINES:

Equation 13 can be used to calculate minimum required storage for various compressor sizes (as they affect V_{net}) and minimum cycle times (i.e. blowdown time). A fraction capacity of 50%, a pressure band of 10 psi, and a compressor output of 4.5 standard cubic feet per minute per horsepower, scfm/hp, are assumed. These results can be seen in Table 1 below.

Table 1 - Required storage per blowdown time (gal)

Required Storage Per Blowdown Time (gal)				
Compressor Size (hp)	**Blowdown Time (sec)**			
	30	**45**	**60**	**90**
10	124	186	247	371
50	619	928	1,237	1,856
100	1,237	1,856	2,474	3,711
150	1,856	2,783	3,711	5,567
200	2,474	3,711	4,948	7,422
250	3,093	4,639	6,185	9,278
300	3,711	5,567	7,422	11,133
350	4,330	6,494	8,659	12,989
400	4,948	7,422	9,896	14,844

With these results, normalized guidelines for storage sizing can be calculated by dividing required volume by rated scfm of the compressor. Because of the assumption of 4.5 scfm/hp over horsepower range, required storage per rated air output is independent of compressor size. Eliminating the independent variable of compressor size, Equation 13 generates Table 2 below. These guidelines match very closely to previously developed guidelines of 4.13 and 5.5 gallons/rated-scfm for 90 and 120 second full cycle times assuming a 10 psi pressure band and 50% fraction capacity [Beals, 2009].

Table 2 - Volume storage required for given pressure band (gal/rated-scfm)

Volume Storage Required per Rated Scfm (gal/rated-scfm)				
Pressure Band (psi)	**Blowdown Time (sec)**			
	30	**45**	**60**	**90**
5	5.5	8.2	11.0	16.5
10	2.7	4.1	5.5	8.2
15	1.8	2.7	3.7	5.5
20	1.4	2.1	2.7	4.1

APPLICATION

CASE STUDY 1: UTILIZING GUIDELINES FOR DESIGNED CYCLE TIME

To demonstrate the role blowdown time has on power draw, the following example is considered. A GA-75FF 100-hp air-cooled Atlas Copco combined rotary screw air compressor with integrated refrigerated dryer provided compressed air to a plant. The compressor operated in load/unload mode. The compressor operated between 100 and 110 psig with no significant compressed air storage. Current draw of the compressor was logged for about 3 hours. During the logged period the compressor served a bag house with automatic shake-down. Plant personnel closed the valve to the bag house at around 1:15 pm. Figures 9, 10 and 11 below show the current draw over the entire logged period and 15 minute snapshots both before and after the bag house valve was closed.

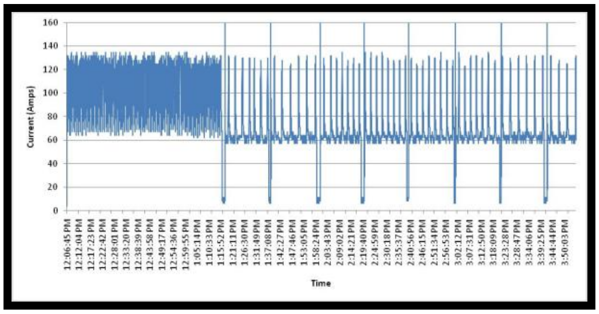

Figure 9 – 100-hp Atlas Copco air compressor – (entire log time)

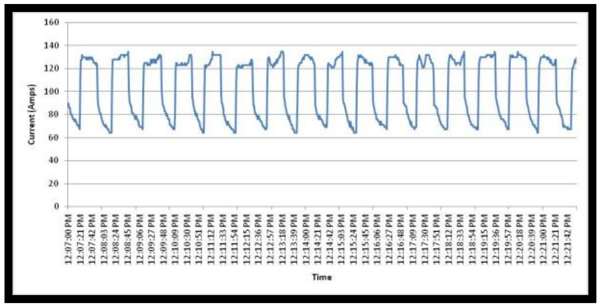

Figure 10 – 100-hp Atlas Copco air compressor before shutting valve to bag house– (15 minute log time)

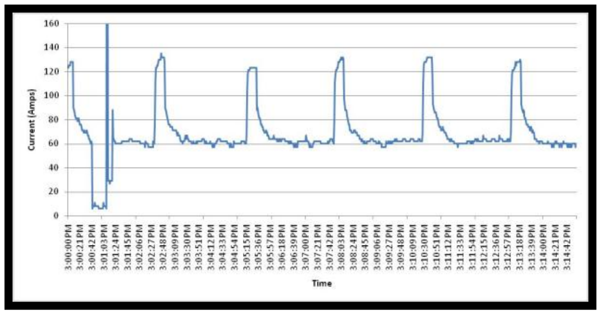

Figure 11 – 100-hp Atlas Copco air compressor after shutting valve to bag house– (15 minute log time)

During the period in which the bag house was in operation, the system operated at around FC = 51%. Therefore, when the bag house was consuming compressed air the system operated around the fraction capacity for which cycle time is minimized. It is shown in Figure 10 that during the periods in which the bag house was consuming compressed air the compressor was not able to fully blow down each cycle. Energy consumption and mechanical wear would be reduced if the system could fully blowdown before cycling into load operation. In order to estimate expected savings, the performance of the compressor was simulated using AirSim and calibrated to the logged current draw. The simulated current draw and system air pressure for the baseline condition are shown in the Figure 12. The average simulated power draw of the interval is 55.9 kW.

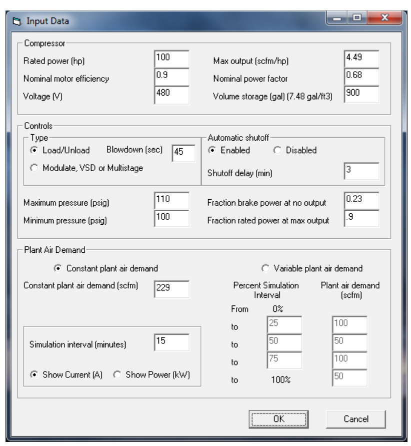

Figure 12 – AirSim input

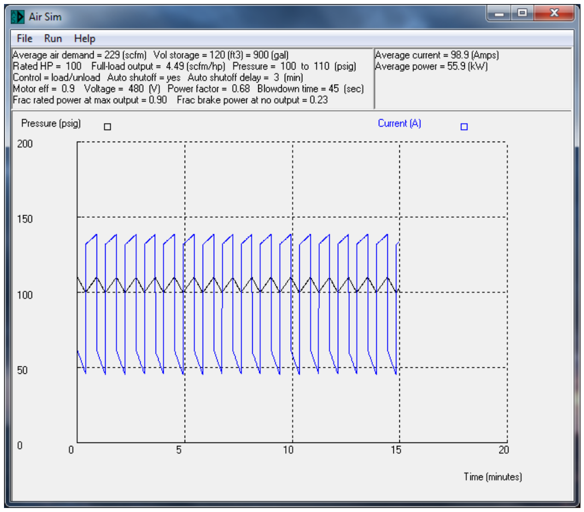

Figure 13 – AirSim output for baseline conditions

Utilizing guidelines from Table 2 it is seen that for a pressure band of 10 psig and a blowdown time of 45 seconds, 4.1 gallons per full load scfm are recommended to avoid short cycling. Therefore, the required storage to allow the compressor to fully blow down before cycling is at least 1,841 gallons. Figure 14 shows the simulated system with the suggested amount of storage.

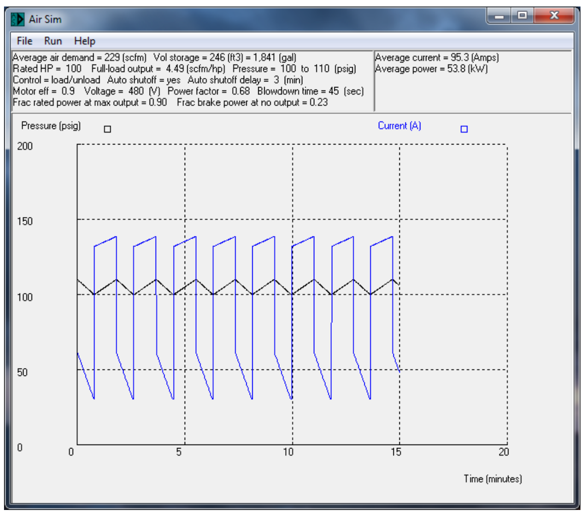

Figure 14 – AirSim output with 1,841 gallons of storage

The average expected power draw with increased storage is 53.8 kW resulting in an average reduction in power consumption of 2.1 kW.

CASE STUDY 2: DESIGNING SYSTEM STORAGE TO MINIMIZE ENERGY USE DURING PERIODS OF LOW COMPRESSED AIR CONSUMPTION

To demonstrate the power of storage calculations, the following example is considered [Schmidt, Kissock, 2005]. Four 30-hp rotary-screw air compressors provided compressed air for a plant. The compressors were set to operate in load/unload control with the activation pressures staged so that three of the compressors were base loaded and one ran as the trim compressor. The activation pressures on the trim compressor were 85 psig and 92 psig. Figure 15 shows the measured current draw of the 30-hp trim compressor over a time interval of about 55 minutes. The compressor alternates between loaded, unloaded and auto-shutoff. When unloaded, the compressor draws about 50% of full-load power, and when loaded draws about 117% of rated motor power. The compressor enters auto-shutoff when unloaded for more than 2.5 minutes. The average power draw over this interval was 17 kW. The short cycle times indicate relatively little storage capacity in the compressed air distribution system.

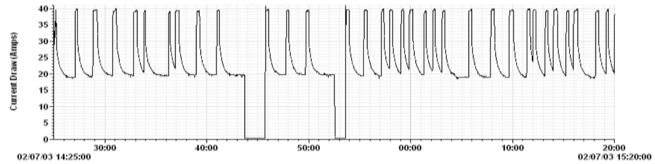

Figure 15 – Current draw of a 30-hp lag compressor operating in load/unload control with auto-shutoff activated

During the logged period it was observed that there were periods of lower compressed air consumption. Overall energy consumption would be reduced if the compressor system were designed so that during these periods the compressor entered auto-shuttoff mode. In order to estimate expected savings, the performance of the trim compressor was simulated using AirSim and calibrated to the logged current draw. The AirSim input screen is shown in Figure 16. The input screen allows the user to define the values of the key system parameters, and to select the most appropriate output for calibration.

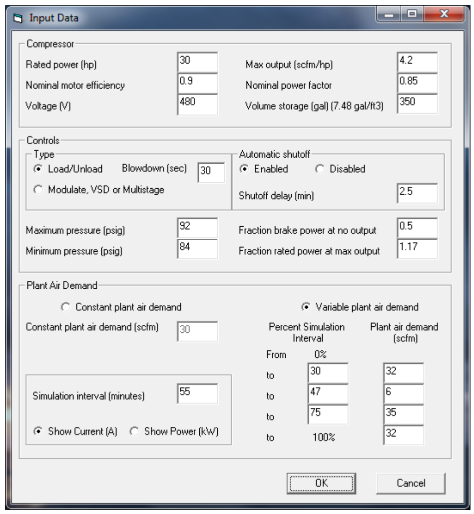

Figure 16 – AirSim input screen

The simulated current draw and system air pressure for the baseline condition are shown in the Figure 17. The simulation results compare well to the measured data. The compressor draws about 40 Amps when loaded, 20 Amps when unloaded, and runs in auto-shut off mode for a few minutes during the middle of the period. The average simulated power draw of the interval is 17.2 kW.

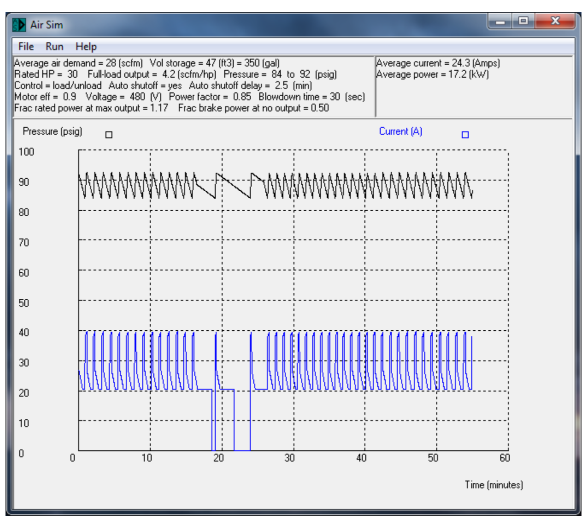

Figure 17 – Baseline simulated current draw and system air pressure

Next, Equation 13 was used to calculate the amount of storage required to extend cycle length to 10 minutes during the period of low compressed air consumption as shown below in Equation 19.

$$V = \frac{10\ minutes \quad 14.7\ psi \ (126\ scfm)}{(7\ psi)} \left(\frac{1}{(1-0.048)} + \frac{1}{0.048} \right)^{-1} \qquad (19)$$

It is discovered that the total required volume of storage is around 120 cubic feet, or around 896 gallons. Next, compressor performance was simulated with an additional 546 gallons of storage. The simulated current draw and system pressure are shown in Figure 18. The results indicate that adding 546 gallons of storage would allow the compressor to run unloaded longer and shut off during the period of low consumption, reducing energy use while delivering the same amount of compressed air.

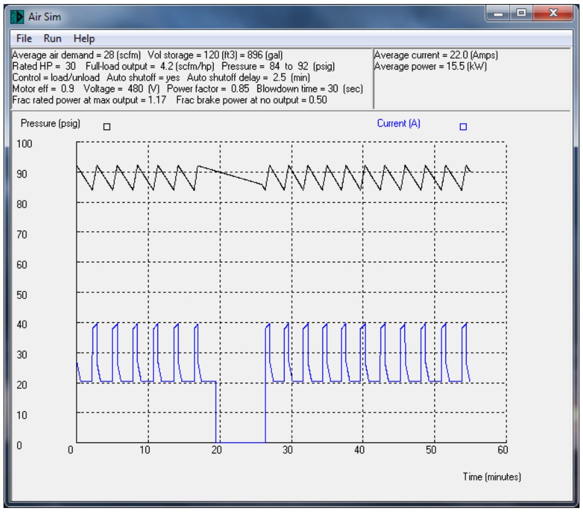

Figure 18 – Expected current draw and system pressure after adding 546 gallons of storage.

The average expected power draw with increased storage is 15.5 kW. Assuming the loading in Figure 15 is typical over the 8,400 hours per year the plant operates, the energy savings from adding storage would be about:

(17.2 kW – 15.5 kW) x 8,400 hours/year = 14,280 kWh/year

SUMMARY/CONCLUSIONS

It has been shown that minimum cycle time for load-unload compressors occurs when a compressor is operating at 50% of full load capacity, and that if a compressed air system lacks sufficient storage, the compressor will short cycle. This short cycling increases both wear on equipment and overall energy consumption. Short cycling increases energy use greatly in lubricated compressors that can take up to 90 seconds to fully blow down the air and oil reservoir because during the blowdown period the compressor does not achieve fully unloaded power draw. As such, by following suggested guidelines for minimum system storage, cycle time at 50% capacity can be designed to exceed that of the blowdown time or other required cycle time minimum. This effectively reduces wear on the compressor and minimizes energy use. For this paper compressor power is modeled varying linearly as the compressor blows down. Future study into this area should include the effects of non-linear power draw during blowdown time to more accurately predict savings from increased cycle time.

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