

Optimizing Compressed Air Storage for Energy Efficiency

2011-01-0323

Published
04/12/2011

Brian James Abels and Kelly Kissock
Univ. of Dayton

Copyright © 2011 SAE International

doi:10.4271/2011-01-0323

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.

$$m_i = \frac{P_i V}{RT_i} \quad (1)$$

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.

$$\dot{m} = \dot{V}_{net} \rho \quad (2)$$

$$\Delta m = \dot{V}_{net} \rho \Delta t \quad (3)$$

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:

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

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](#).

$$\dot{V}_{net} \rho \Delta t = \frac{V}{RT} (P_2 - P_1) \quad (5)$$

$$V = \frac{\dot{V}_{net} \rho \Delta t RT}{(P_2 - P_1)} \quad (6)$$

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](#).

$$V = \frac{\dot{V}_{net} \rho \Delta t}{(P_2 - P_1)} \left(\frac{P_s V}{m} \right) = \frac{\dot{V}_{net} \Delta t P_s V}{(P_2 - P_1) m} \left(\frac{m}{V} \right) \quad (7)$$

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](#).

$$V = \frac{\dot{V}_{net} \Delta t P_s}{(P_2 - P_1)} \quad (8)$$

$$\Delta t = \frac{V \Delta P}{\dot{V}_{net} P_s} \quad (9)$$

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.

$$\Delta t_{tot} = \Delta t_L + \Delta t_U = \frac{V \Delta P}{P_s} \left(\frac{1}{\dot{V}_L} + \frac{1}{\dot{V}_U} \right) \quad (10)$$

During unload operation the net change in volume of the tank is the plant demand, or the maximum output of the compressor, $\dot{V}_m$, 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](#).

$$\dot{V}_U = \dot{V}_m * FC \quad (11)$$

$$\dot{V}_L = \dot{V}_m (1 - FC) \quad (12)$$

Substituting [Equations 11](#) and [12](#) into [Equation 10](#) and factoring out the maximum output of the compressor, $\dot{V}_m$, the total cycle time can be described as:

$$\Delta t_{tot} = \frac{V \Delta P}{P_s} \left(\frac{1}{\dot{V}_m (1 - FC)} + \frac{1}{\dot{V}_m * FC} \right) = \frac{V \Delta P}{P_s \dot{V}_m} \left(\frac{1}{(1 - FC)} + \frac{1}{FC} \right) \quad (13)$$

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.

$$\Delta t_{tot} = \frac{V \Delta P}{P_s \dot{V}_m (1 - FC)} \left(\frac{(1 - FC)}{(1 - FC)} + \frac{(1 - FC)}{FC} \right) = \frac{V \Delta P}{P_s \dot{V}_m (1 - FC)} \left(\frac{(1 - FC)}{(1 - FC)} + \frac{1}{FC} - \frac{FC}{FC} \right) \quad (14)$$

$$\Delta t_{tot} = \frac{V \Delta P}{P_s \dot{V}_m (1 - FC)} \left(\frac{1}{FC} \right) = \frac{V \Delta P}{P_s \dot{V}_m (FC - FC^2)} \quad (15)$$

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

$$k = \frac{V \Delta P}{P_s \dot{V}_m} \quad (16)$$

$$\Delta t_{tot} = \frac{k}{(FC - FC^2)} \quad (17)$$

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.

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

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).

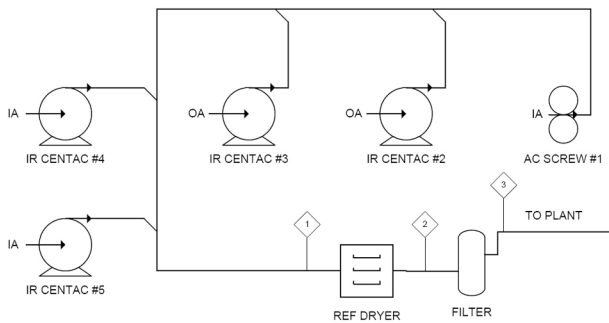


Figure 1. Logged pressure locations

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.

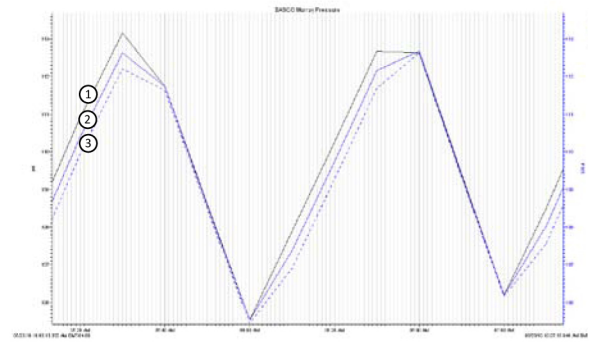


Figure 2. Logged pressure of load-unload operation

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.

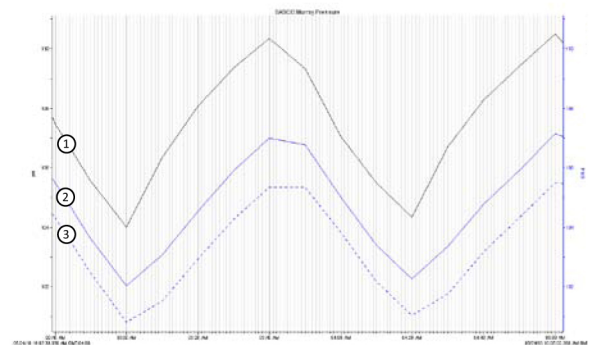


Figure 3. Logged pressure draw at high flow

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.