Compressed Air Storage

Storage can be used to control demand events (peak demand periods) in a compressed air system by reducing both the amount of pressure drop and the rate of decay. Storage can be used to protect critical pressure applications from other events in the system. Storage can also be used to control the rate of pressure drop in demand while supporting the speed of transmission response from supply. For some systems, it is important to provide a form of refill control such as a flow control valve. Many systems have a compressor operating in modulation to support demand events, and sometimes strategic storage solutions can allow for this compressor to be turned off.

Primary Air Receivers

The basic purpose of an air receiver is to store a volume of compressed air for use when needed. The most common example is a small, air-cooled, piston type compressor, mounted on a tank or air receiver. The compressor operates on a start/stop control system, usually controlled by a pressure switch having a fixed differential. Because of automotive applications, the pressure at which the compressor is stopped normally is 175 psig. The compressor is restarted when the use of the compressed air causes the pressure to fall to about 145 psig (a differential of 30 psi).

On larger compressor sizes, the compressor may be loaded and unloaded in a range of 145-160 psig but continues to run. The tank provides radiant cooling and requires an automatic drain to remove condensate. The problem from an energy standpoint is that all of the air is being compressed to at least 145 psig and most of it to 160 psig, although most applications require a much lower pressure. Pneumatic tools normally are designed for operation at 90 psig, so energy is being expended to compress the air well beyond what is needed. A rule of thumb for systems in the 100 psig range is for every 2 psi increase in discharge pressure, energy consumption will increase by approximately 1 percent at full output flow (check performance curves for centrifugal and two-stage lubricant injected rotary screw compressors). There is also another penalty for higher-than-needed pressure. Raising the compressor discharge pressure increases the demand of every **unregulated usage**, including leaks, open blowing, etc. if no pressure/flow controller is present. Although it varies by plant, unregulated usage is commonly as high as 30-50 percent of air demand. For systems in the 100 psig range with 30-50 percent unregulated usage, a 2 psi increase in header pressure will increase energy consumption by about another 0.6 – 1.0 percent because of the additional unregulated air being consumed. The combined effect results in a total increase in energy consumption of about 1.6 to 2 percent for every 2 psi increase in discharge pressure for a system in the 100 psig range with 30-50 percent unregulated usage.

In industrial compressed air systems, the supply side generally is considered to be the air compressors, dryers and related filters, and a primary air receiver. There are two differing points of view on the location of a primary air receiver in a plant air system. If the receiver is located soon after the compressor discharge and the compressor(s) is a piston type, the receiver acts as a dampener for pressure pulsations. If the receiver is located before the compressed air dryer, the receiver will provide additional radiant cooling and drop out some of the condensate and entrained oil, benefiting the dryer. However, the receiver will be filled with saturated air and if there is a sudden demand which exceeds the capacity rating of the compressor and matching dryer, the dryer can be overloaded, resulting in a higher pressure dew point.

If the air receiver is located after the compressed air dryer, some of the above advantages are lost but the receiver is filled with compressed air which has been dried and a sudden demand in excess of the compressor and dryer capacity rating will be met with dried air. The dryer is not overloaded, since it is seeing only the output of the compressor, so the pressure dew point is not affected. In either case, it should be recognized that the compressed air dryer and associated filters will add pressure drop, which must be taken into account when determining the compressor discharge pressure to achieve the desired pressure leaving the primary air receiver to the system.

The size of an air receiver can be calculated as follows:

$$V = T \times \frac{C \times P_a}{P_1 - P_2}$$

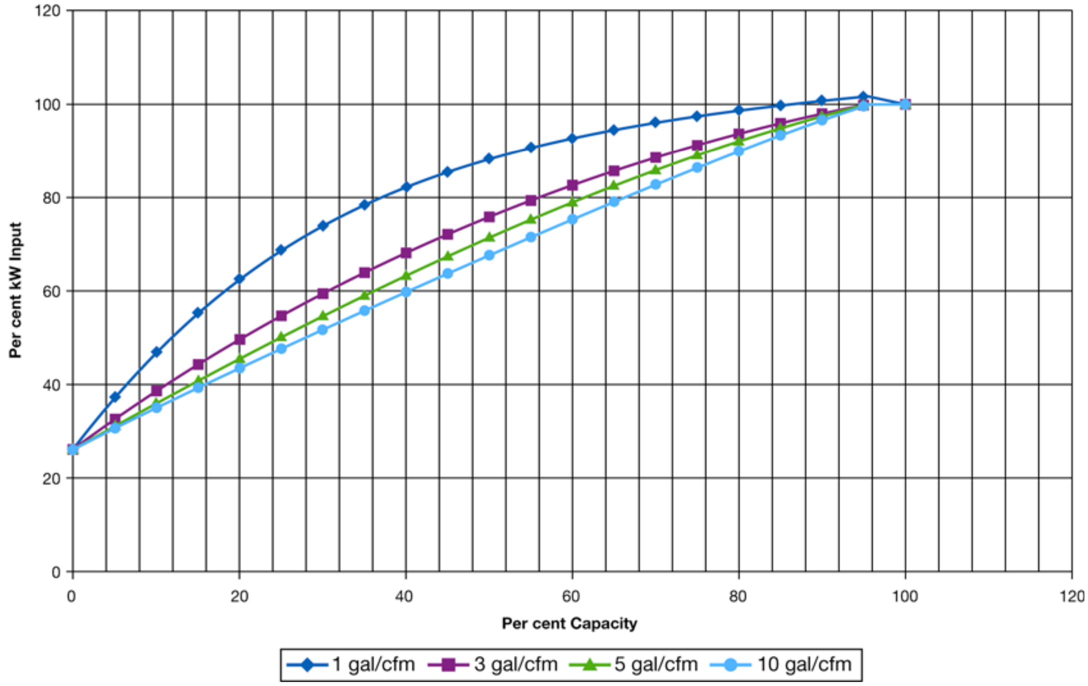
Where:
V = Receiver volume, $ft.^3$
T = time allowed (minutes) for pressure drop to occur
C = Air demand, cfm of free air
P_a = Absolute atmosphere pressure, psia
P_1 = Initial receiver pressure, psig
P_2 = Final receiver pressure, psig

The formula assumes the receiver volume to be at ambient temperature and that no air is being supplied to the air receiver by the compressor(s). If the compressor(s) is running while air is being drawn from the receiver, the formula should be modified so that C is replaced by C – S, where S is the surplus compressor capacity, cfm of free air. The initial formula also can be used with a known receiver size, to determine the time to restore the air receiver pressure. In this case, C is replaced by S, which is the compressor capacity, cfm of free air.
In the past, mainly with reciprocating compressors, rules of thumb for sizing a primary air receiver, have been from 1 gallon per cfm to 3 gallons per cfm of compressor capacity. This is no longer regarded as good practice and the recommended primary receiver size will vary with the type of compressor and capacity control used.

Some lubricant-injected rotary screw compressors are sold with load/unload capacity control, which is claimed to be the most efficient. This also can be misleading, since an adequate receiver volume is essential to obtain any real savings in energy.

One solution sometimes proposed is to eliminate modulation and have the compressors operate in a load/unload mode. Certain factors must be recognized before making such a change. The standard full capacity, full load pressure, often has the compressor running at around 110 percent of motor nameplate rating, or using 10 percent of the available 15 percent continuous overload service factor. The remaining 5 percent is meant to cover tolerances and items such as increased pressure drop through the air/lubricant separator before it is required to be changed.

If the discharge pressure is allowed to rise by an additional 10 psi without the capacity being reduced by inlet valve modulation, the bhp will increase by 5 percent and the motor could be overloaded. A reduction in discharge pressure may be necessary to operate in this mode.



Effect of Receiver Capacity on Lubricant-Injected Rotary Compressor with Load/Unload Capacity

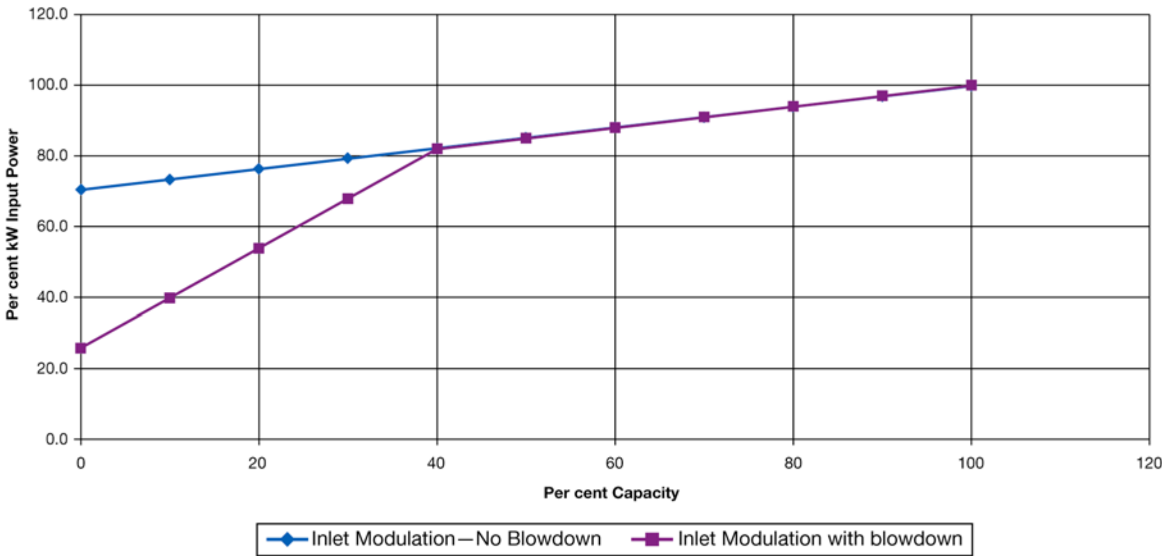
In addition, for lubricant-injected rotary screw compressors it is falsely assumed that a straight line, from full load bhp to unloaded bhp, represent the actual power requirement in this mode of operation. This presumes, for example, that if the average capacity is 50 percent, the compressor would run fully loaded 50 percent of the time and fully unloaded 50 percent of the time. Unfortunately, the compressor is not fully unloaded 50 percent of the time.

When the compressor discharge pressure reaches the unload set point, the inlet valve is closed to reduce the mass flow through the compressor. Simultaneously, the lubricant sump/separator vessel pressure begins to be relieved gradually. Typically, this takes about 40 seconds to prevent foaming of the lubricant with the potential of excessive lubricant carry-over. The rate at which blow-down occurs gradually slows as the pressure is reduced. The fully unloaded power is not realized until the pressure in the lubricant sump/separator is fully relieved. In addition, a period of about 3 seconds is required to re-pressurize the air/lubricant sump/separator vessel when the system calls for the compressor to re-load.

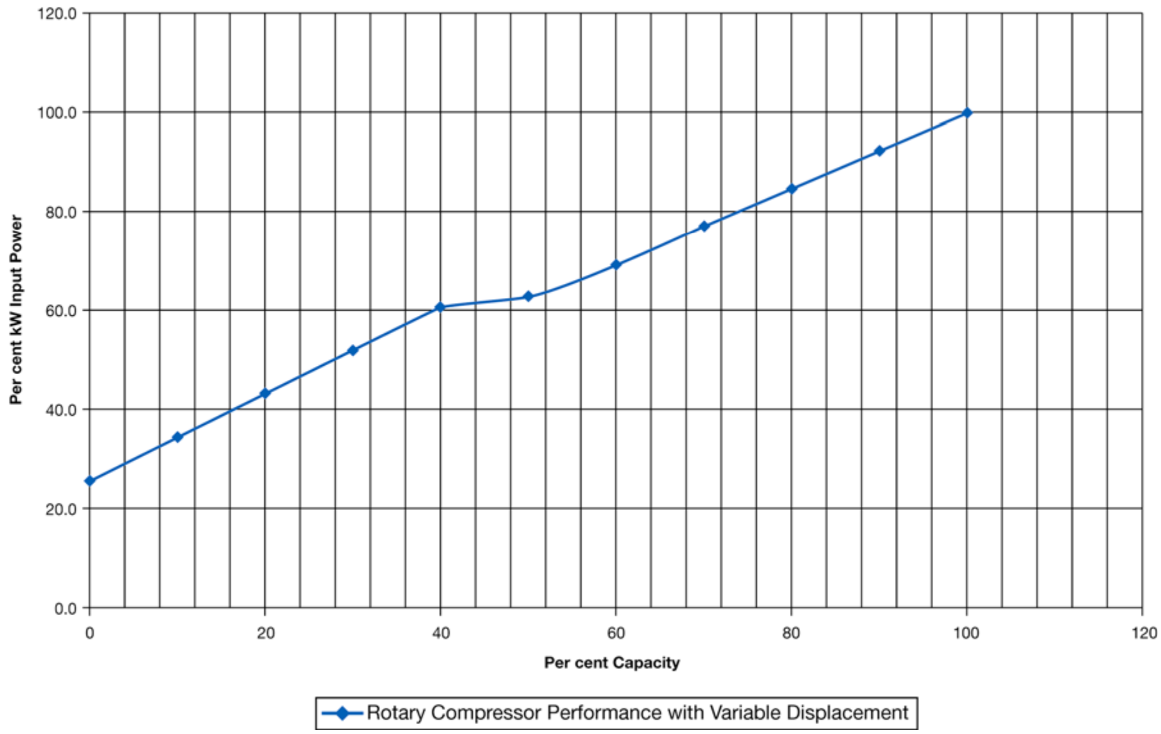
In many cases, the system pressure will fall and the compressor will re-load before the fully unloaded power is realized. To overcome this, an adequately sized air receiver and/or system volume is essential. Taking the above factors into account, Figure F6-1 shows the effect of different sizes of air receivers/system volume. It will be seen that some rules of thumb established many years ago for reciprocating air compressors are not adequate for a lubricant-injected rotary screw compressor.

Most lubricant-injected rotary screw compressors are equipped with capacity control by inlet valve modulation designed to match the output from the air compressor with the demand from the points-of-use. On this basis, it has been stated that an air receiver is not needed. At best, this is misleading. An air receiver near the discharge of a rotary screw compressor will shield the compressor control system from pressure fluctuations from the demand side downstream of the receiver, and can allow the compressor to be unloaded for a longer period of time, during periods of light demand. The addition of an over-run timer (Automatic Dual Control) can stop the compressor if it runs unloaded for a pre-set time, savings additional energy.

The top line on Figure F6-2 shows what would happen if inlet valve modulation was used without unloading the compressor.



Lubricant-Injected Rotary Compressor with Inlet Valve Modulation



Lubricant-Injected Rotary Compressor with Variable Displacement

Approximately 70 percent of full load power would still be used when modulation had reduced compressor output to zero. The second line on that graph shows inlet valve throttling to 40 percent capacity and unloading at that point. Figure F6-3 shows variable displacement (slide/turn/spiral/poppet valve) capacity reduction to 50 percent capacity followed by throttling to 40 percent capacity and unloading at that point.

Variable speed may be achieved by variable frequency ac drives, or by switched reluctance dc drives. Each of these has its specific electrical characteristics, including inverter and other losses.

Air end displacement is directly proportional to rotor speed but air end efficiency depends upon male rotor tip speed. Some variable speed drive (VSD) package designs involve full capacity operation above the optimum rotor tip speed, at reduced air end efficiency and increased input power, when compared with a constant speed compressor of the same capacity, operating at or near its optimum rotor tip speed. Efficiency with VSD generally is improved at reduced capacities. The best energy savings are realized in applications where the running hours are long, with a high proportion in the mid to low capacity range. Some designs stop the compressor when a lower speed of around 20 percent is reached, while others may unload at 40-50 percent, with an unloaded power of 10-15 percent. The appropriate amount of storage volume should be considered for each of these scenarios.

Field conversion of an existing compressor to variable speed drive must consider the electric motor, the proposed male rotor tip speed at 100 percent capacity and the reduction of air end efficiency at reduced speeds and capacity.

It should be noted that in systems with multiple compressors and sequencing controls, it is possible to have most of the compressors running fully loaded on base load with only one compressor modulating (operating as the “trim” compressor), providing the most efficient mode for the system. It also is not necessary to have the air receiver/system storage capacity based upon the total capacity of all the compressors, provided they are not all on the same load and unload pressure settings.

A primary air receiver allows the compressor(s) to operate in a given discharge pressure range (usually 10 psi) from load to unload. Multiple compressors also can be sequenced as needed and with all but one operating in the most efficient, fully loaded mode.

Secondary Air Receivers

In many industrial plants, there will be one or more applications with an intermittent demand of relatively high volume. This can cause severe dynamic pressure fluctuations in the whole system, with some essential points-of-use being starved, impacting the quality of the end product. Usually, this can be relieved by the correct sizing and location of a secondary air receiver close to the point of high intermittent demand. Such demand often is of short duration and the time between the demand events is such that there is ample time to replenish the secondary receiver pressure without adding compressor capacity. A check valve before the secondary air receiver will prevent back flow to the rest of the system and ensure that the required volume is stored to meet the anticipated event(s).

Correctly sized and located air receivers can provide major advantages in a compressed air system and require little maintenance. They should meet ASME unfired pressure vessel requirements and have appropriate pressure relief valves. An automatic drain device, with manual bypass for service, also should be included. Condensate removed should be decontaminated to meet appropriate Federal, State and Local Codes.

Additional pressure/flow controls can be added after the primary receiver to maintain a reduced and relatively constant system pressure and at points-of-use, while allowing the compressor controls to function in the most efficient control mode and discharge pressure range, with significant energy savings.