Compressed Air Energy Efficiency

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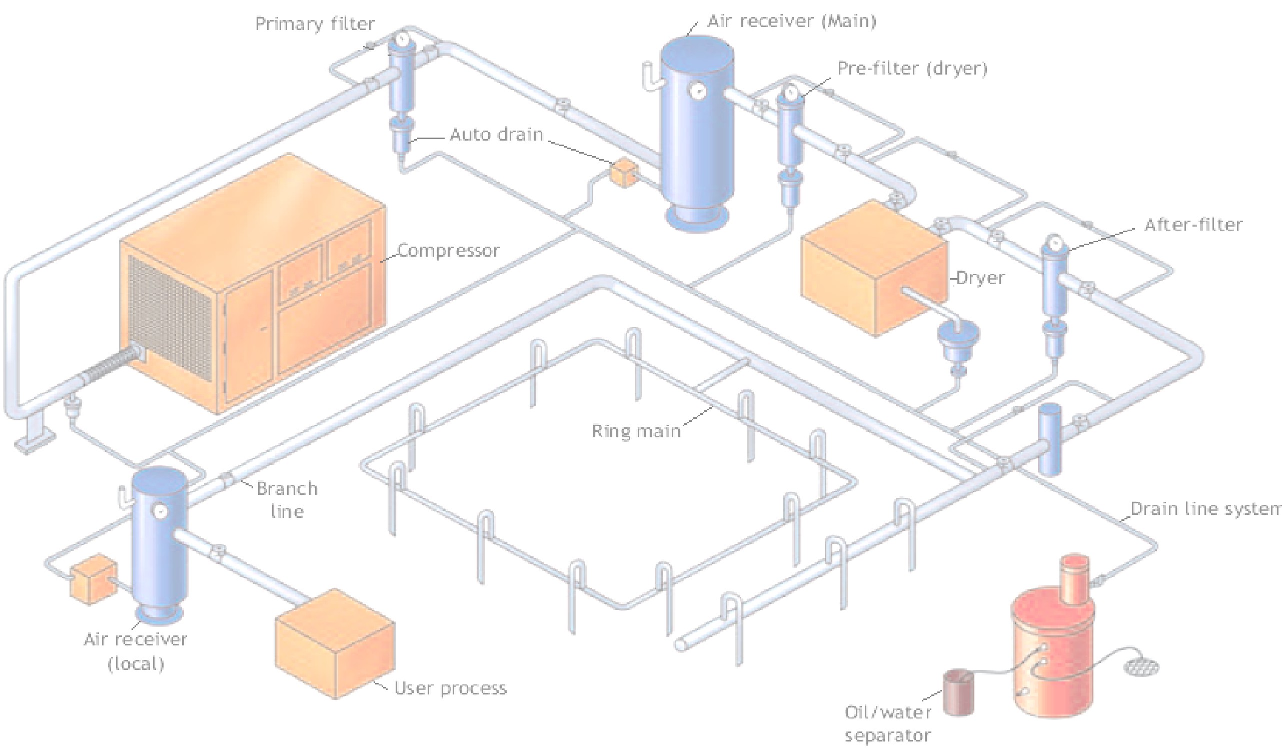
COMPRESSED AIR ENERGY EFFICIENCY



Contents

COMPRESSED AIR ENERGY EFFICIENCY

Overview

Compressed air is widely used for industrial purposes due to its various technological advantages such as high operating speed, force, accuracy and safe handling. But despite these advantages, the compressed air systems consume considerable amounts of energy. It takes about 8 hp of electrical energy to produce 1-hp-worth of work with compressed air. Here's some food for thought:

As a good approximation, typical compressor produces:

4 cubic foot per minute (CFM) per 1 motor hp (horsepower)

Where:

1 hp = 0.746/0.9 = 0.829kW

Therefore:

1 CFM = 0.207kW

And, at $0.05/kW-hr:

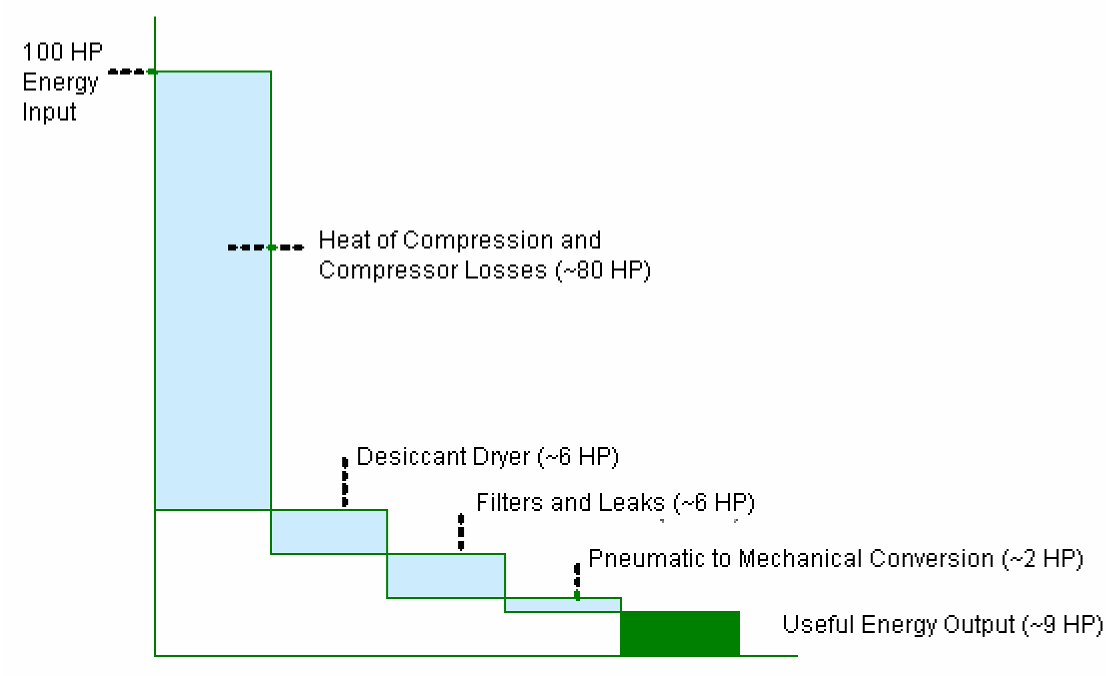
1 CFM = $0.0104/hr

Thus, 10 CFM over 8000 hours per year costs:

1 x 8000 x 0.0104 = $83.20

Approximately 90% of the energy to produce and distribute compressed air is lost.

Figure below illustrates the typical losses associated with producing and distributing compressed air. Assuming 100 HP energy input, approximately 91 HP ends up as losses, and only 9 HP as useful work.



Compressed Air Energy Input and Useful Energy Output

Certainly compressed air is the most expensive energy utility – the figure above shows that approximately 10% percentage of useful energy only reaches the point of final use. Always question if compressed air is the most appropriate power source for an end use application. In many cases, you would be better off to use a direct drive electric tool instead of a compressed air driven one.

Most facilities can easily save 10-20% of their compressed air energy costs through routine maintenance such as the fixing of air leaks, lowering air pressure, and replacing clogged filters. Even higher savings numbers can be gained by choosing better compressor control, adding storage receiver capacity, and upgrading air dryers and filters. This course explains how the selection, control and maintenance of compressed air plant can improve energy efficiency and reduce running costs.

SECTION #1 COMPRESSOR TYPES AND CONTROLS

Every compressed-air system begins with a compressor - the source of air flow for all the downstream equipment and processes. The main parameters of any air compressor are capacity, pressure, horsepower, and duty cycle. It is important to remember that *capacity* does the work; *pressure* affects the rate at which work is done. Both are independent – i.e. adjusting an air compressor's discharge pressure does not change the compressor's capacity.

There are two basic compressor types:

1. Positive displacement, which includes reciprocating and rotary air compressors, and

2. Dynamic, which includes centrifugal and axial air compressors

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Reciprocating Air Compressors

Reciprocating air compressors are positive displacement machines, which function by increasing the pressure of the air using a piston within a cylinder. There are three basic selection decisions that must be made about reciprocating compressors:

1. Single- or double-acting operation

2. Single- or multi-stage configuration

3. Air or water cooling option

In a ***single-acting*** compressor, the piston only compresses air in one direction of its stroke. In a ***double-acting*** model, the piston compresses air with both directions of its stroke. Obviously, because both strokes perform work, a double-acting compressor is more efficient (in moving a volume of air per input hp) than a comparable-size single-acting unit. However, they also are heavy and bulky, making them relatively expensive to install. They generally have more-significant unbalanced forces, which combines with their size to require a special foundation and support.

A ***single-stage*** unit compresses air from inlet to discharge pressure in one operation. Usually single stage operation is in pressure ranges of 95 psi or less. A ***multi-stage*** unit compresses from inlet to discharge pressure in two or more operations. Multiple stage units are theoretically more efficient. They can cool down the air between stages reducing the work required to compress the air. Usually two-stage operation is in pressure ranges of 100 – 175 psig and three-stage reciprocating units are generally used for pressures above 250 psig.

Air-cooled compressors, as the name implies, are cooled by ambient air. The compressor cylinders head are finned to provide increased cooling and heat transfer. Air-cooled units are generally designed for <u>50% to 75% duty cycles</u>*, depending on the particular units and their application. In ***water-cooled*** compressors, integral water jackets surround the cylinders and heads. Heat transfer through the water is much more efficient than air.

Duty Cycle

Duty cycle is the percentage of time, the compressor motor is generally running under loaded conditions. In an application, at 50% duty cycle, and at 4 cfm/ hp, a 32.65 cfm application will require a compressor capacity of 16.32 hp and NOT 8.16 hp…..[32.65 cfm ÷ 4 cfm/hp] ÷ 50% duty cycle = 16.32 hp. For a reciprocating compressor to be categorized as *continuous duty*, it is generally agreed that it must be <u>double acting and water cooled</u>.

Two primary control system types are available in reciprocating compressors: On/off control and load/unload control. Reciprocating compressors are designed as two-step (start/stop or load/unload), three- step (0%, 50%, 100%) or five-step (0%, 25%, 50%, 75%, 100%) control. These control schemes generally exhibit an almost direct relationship between motor power consumption and loaded capacity. Generally speaking reciprocating air compressors have better unloading characteristics than screw compressors, and are more suited to single compressor installations, with fluctuating air demand.

Most air-compressor manufacturers promote the two-stage – single acting compressor as the optimum machine for producing 100-psi class air - the base pressure level in most industrial plants. These compressors are available with oil-lubricated and oil-free cylinders.

Rotary Air Compressors

Rotary air compressors are positive displacement compressors and are most commonly used in sizes ranging from about 5 to 900 HP. Depending on the air purity requirements, rotary screw compressors are available as lubricated or dry (oil free) types.

1. **Oil - cooled rotary helical screw compressors** - This type of unit provides non-pulsating air in range of 22 to 3,100 cfm. Two-stage rotary-screw compressors are frequently used in the 150- to 400-psia pressure range and offer advantages associated with lower compression ratio per stage. Reduced pressure differential across the rotors minimizes blow-by and significantly reduces thrust-bearing loads. (Obviously two-stage units require two air ends, which increase the initial cost.)

   The unique characteristic of this compressor is that it is cooled by oil. Oil injected into the air stream absorbs the heat of compression while it is being generated. The heated oil then is taken to an air- or water-cooled heat exchanger for cooling. Because the cooling takes place right inside the compressor, there are no hot spots inside the airend, no matter what the load on the compressor. In other words, oil-cooled rotary-screw compressors can run at full load and full pressure -twenty-four hours a day, seven days a week.

   Compared to other types of continuous-duty air compressors, oil-cooled rotary-screw compressors offer a number of advantages:

   - Oil cooling holds internal temperatures to an optimum level. As a result, discharge air is relatively cool -no more than about 180°F higher than ambient.
   - Discharge air is clean - free from burned oil or carbon.

- The rotary design lends itself to higher speeds, particularly in the larger sizes. Consequently, larger flow capacity is available from compressors with physically smaller envelopes - providing significant savings on floor space and foundation requirements.

- Because of their compact size and inherent quiet-running characteristics, it is relatively easy to suppress noise. Electric-motor-driven models are commercially available rated from 75 to 85 dB at one meter.

- Most models have fewer moving parts, and those parts run under more ideal conditions - resulting in lower temperatures and less vibration.

- Fewer parts make it easier to stock them for the rotary designs, and the machines are easier to work on.

In summary, oil-cooled rotary-screw compressors offer users a continuous-duty source of compressed air in a neat, compact package that has low initial cost, maximum flexibility of installation, and easy maintenance. This type of compressor is best suited as a base load machine.

2. **Non-lubricated rotary screw and lobe compressors** - Also referred to as "clearance-type" compressors because the internal parts do not contact each other. These require no lubrication in the compression chamber and the cooling is accomplished through the cylinder walls via water jackets.

   The lobes or screws do not drive one another either; they are driven by some type of gear arrangement instead. This drive system also acts as a timing gear to maintain the rotor or lobe profile relationship accurately. Lubricant for the drive train must be confined to the bearing and gear area - and not allowed to get into the compression chamber.

   In this basic design, there is a constant leakage rate for any fixed set of conditions. The critical internal clearances are between end covers and the rotor, between the rotor lobes, and between the rotor OD and the cylinder ID. These gaps, combined with no injected oil to help with sealing, are the main reasons why two stages are required for these units to produce acceptable efficiencies in 100-psi class applications.

Oil-free rotary helical screw compressors are available in volume range from 400 to 12,000 cfm and oil-free rotary lobe compressor is available from 100 to 500 cfm.

3. **Sliding vane rotary compressors** – Sliding-vane compressors function by trapping a charge of intake air between the vanes. As the eccentric rotor turns, the vanes are forced into the rotor slots, shrinking the size of the cell holding the trapped air. The air is compressed to full discharge pressure when it reaches the outlet port. The heat of compression is removed by cooling oil sprayed right into the air while it is being compressed. The same oil helps with sealing the vane tips.

   Air volumes range up to approximately 3,000 cfm. Such compressors can be oil-injected, oil- flooded or oil-free types. This type of compressor has low operating cost, no pulsation, and is free from vibration. This permits installing the compressor directly on the simplest foundation.

Four primary control system types are available in rotary screw compressors: variable speed drive, load/unload, adjustable rotor length and throttling (listed in order of most efficient to least efficient at part load).

Rotary positive displacement compressors are smaller and quieter than reciprocating compressors. They also have smaller footprints than equal size reciprocating models, and may be installed directly on the factory floor. They also do not produce the pulsations typically found in reciprocating compressors due to continuous flow. The biggest advantage of screw compressors over small air cooled reciprocating units is that they can run at full load continuously where the reciprocating compressors must be used at 60% duty cycle or below.

Two-stage rotary compressors are more efficient than single-stage reciprocating, but not as efficient as two-stage, double-acting reciprocating units. Another drawback of rotary units is that their efficiency quickly decreases at part load. They may not be the most efficient choice compared to start/stop reciprocating compressors.

Centrifugal Air Compressors

Centrifugal compressors are **dynamic** compressors which raise the pressure of air by imparting velocity energy, using a rotating impeller, and converting it to pressure energy. Approximately one-half of the pressure energy is developed in the impeller with the other half achieved by converting the velocity energy to pressure energy as the air speed is reduced in a diffuser and volute.

Centrifugal compressors are generally used in applications requiring a large volume of air flow but usually at relatively lower pressures. They are only the real option over 600 hp. Centrifugal compressors are **oil-free** by design (0 ppm oil carryover).

When a centrifugal compressor needs to provide flow less than 80 percent capacity it will "blow-off" or vent the compressed air directly to the atmosphere or the surroundings. Running a centrifugal compressor in "blow-off" mode wastes a lot of energy. *For this reason, centrifugal compressors should be* ***base-load*** *compressors that operate at near 100 percent capacity at all times. A reciprocating compressor or screw compressor with efficient unloading such as rotor shortening or a variable speed drive should be used as a* ***trim compressor*** *to meet the remaining load.*

Centrifugal compressors control output with inlet valves or inlet guide vanes similar to the throttling inlet valve control on a rotary screw compressor. This is not efficient. Surge can occur if a centrifugal compressor is throttled down below 75-80 percent maximum capacity. Surge is a phenomenon associated with the reversal of airflow back into the compressor when it cannot maintain a steady flow of air and can cause excessive vibration and mechanical damage in a short period of time.

COMPRESSOR CONTROLS

Compressor controls are very important factors affecting system performance and energy efficiency.

Key Terms:

Full Load: the air demand exactly matches the total available capacity of the compressor. All compressors run **most efficient** at full load.

Part Load: the air demand is less then the total available capacity of the compressor. All compressors run **less efficient** at part load.

Not all compressors are created equal and the efficiency loss at part load is related to the type of control system it utilizes. The objective of any control strategy is to shut off unneeded compressors or delay bringing on additional compressors until needed. In addition, good control systems maintain lower average pressure without going below minimum system requirements and are designed to match the compressor output with the system demand.

There are at least seven common types of compressor control modes:

1. Start/stop
2. Load/unload
3. Inlet modulation
4. Auto-dual
5. Variable displacement
6. Variable speed
7. System controls

Start / Stop Control

Start/stop control is frequently used by small reciprocating compressors. In this type of control, the compressor turns itself off and draws no power as long as the discharge pressure remains above a specified level. This control strategy is the most energy-efficient mode since a compressor operating in this mode only produces air while running at 100% capacity and never idles; performance approaches the “ideal”. Most rotary compressors are unable to run in start/stop mode.

Pros –

- The air compressor runs only fully loaded

Cons –

- Most AC electric motors can survive only a finite number of starts (usually 4 to 6 per hour) over a given time frame, primarily due to heat build up. This limits the application of automatic start-stop controls - particularly for motors larger than 10 to 25 hp.

- The compressor must run above minimum system pressure to hold that pressure. Care should be taken in sizing storage receivers and maintaining wide working pressure bands to keep motor starts within allowable limits. Large receivers are required for efficient operation.

- The system must have adequate air-storage capacity to perform satisfactorily.

Load\unload Control

With load/unload control, the compressor runs fully loaded, producing compressed air at maximum efficiency until the discharge pressure reaches the upper activation pressure setting, which causes the compressor to unload. When unloaded, the compressor no longer adds compressed air to the system, but the motor continues to run. There will be small loss of energy each time the outlet blows down, because any compressed air preceding the check valve will be vented to attain a lower pressure.

Reciprocating compressors control air output by unloading cylinders. The most common is the two-step control which holds the compressor inlet either fully open or fully shut. Over the complete operational band, the compressor runs fully loaded (or at full flow) from the preset minimum pressure (or *load* point) to the preset maximum pressure (or *no-load* point). At the latter, the control shuts off air flow completely. The unit then runs at no flow and full idle until system pressure falls back to the load point. The control then goes immediately to full-flow capacity. A pressure switch typically actuates the two-step control, which can be either the primary control or part of a dual-control system on virtually every type of air compressor. (Some reciprocating compressors can be fitted with 3- and 5-step controls.)

Pros –

- Reciprocating compressors are typically efficient at part-load operation because the pistons operate against very little air-pressure resistance in this mode and therefore, very little energy is wasted. A fully unloaded reciprocating air compressor uses approximately 10% of its full load energy.

Cons-

- Load/unload method is not very efficient for screw compressors, which will consume approximately 20 to 25 percent of full-load horsepower while delivering no useful work. The energy savings will be less if the discharge pressure drops to 30 – 40 psig, which is usually required to maintain oil circulation.

- Adequate air storage is necessary to allow enough idle time over the operational pressure band to generate any significant energy savings.

Throttling or Modulation Control

Throttling restricts the inlet opening to admit only the amount of air demand by the system. The inlet valve modulates continuously and responds immediately in to any change in the sensed system pressure. In effect, flow capacity is controlled by restricting air intake. The control holds a constant system pressure with minimal valve movement at any given steady system demand. Throttling mode is not desirable if extended low load periods are expected.

Pros-

- Smooth, non-cycling control of system pressure is easier on the power train and most other components.
- Relatively efficient at loads from 60 to 100%.
- Will not short cycle, regardless of storage capacity and or piping.
- Simple to operate and maintain.

Cons-

- Relatively inefficient at loads below 60%.
- Backpressure must be overcome in order to reach full capacity.
- Instant response may make the machine back down and unload, even when flow is needed for the base load.
- Sensitivity and rapid reaction make correct piping and backpressure control necessary for optimum operation.

Variable Displacement Control

Variable displacement controls for rotary screw compressor match output to demand by varying the effective length of rotor compression volume. The inlet pressure remains the same throughout the turn down, and the compression ratio stays relatively stable. This method of reducing flow without increasing compression ratios has a power advantage over modulating and/or 2-step controls in the operating range from 50% to full load.

The two most common of these unloading controls are the spiral-cut high lead valve and the poppet valve. Both methods open and close selected ports in the compressor cylinder, thus changing the seal-off points. These ports are located at the start of the compression cycle where pressure is very low. Opening them even a small amount prevents compression from occurring until the rotor tip passes the cylinder bore casing that separates the ports. This effectively reduces the trapped volume of air to be compressed and consequently the horsepower needed to compress it.

Pros –

- Very efficient part-load performance from 50% to 100%.
- Maintains set pressure at minimum system pressure.
- Very responsive.

Cons -

- At higher loads, some units lose efficiency due to increased leakage.
- The mechanism is complex.
- Still must run 2-step or modulation in lower operating range.

Auto – Dual Control

To avoid wasting pressurized air in centrifugal machines due to bypassing, auto- dual controls can be used to sense the point of maximum turndown and then close the inlet valve and off-load the machine. This reduces considerably the power being consumed.

Auto-dual control is a combination of modulation and load/unload control in which the compressor operates in modulation control down to a specified pressure and switches to load/unload control below this pressure. One disadvantage of auto-dual control is that the pressure differential is increased to about 7 psi during light load running.

Variable Speed Drive Control

A Variable Speed Drive (VSD) Air Compressor is an air compressor that takes advantage of variable speed drive technology. This type of compressor uses a special drive to control the speed (RPM) of the unit, which in turn saves energy compared to a fixed speed equivalent provided the air demand fluctuates.

The most common form of VSD technology in the Air Compressor Industry is a variable frequency drive, which converts the incoming AC power to DC & then back to a quasi-sinusoidal AC power using an inverter switching circuit. The benefits of this technology included reducing power cost, reducing power surges (from starting AC motors), and delivering a more constant pressure. Another inherent advantage of VSD is the ability to start and stop as often as desired. Unlike fixed drives, VSD systems "soft start" and incur the lowest required inrush current. Whereas a 100-hp fixed drive is limited to two or three starts and stops per hour because required inrush current heats up the motor windings, the VSD has no limit. Power companies may penalize users for even one high-inrush spike on the demand chart. The down side of this technology is the heavy expense associated with the drive, and the sensitivity of these drives - specifically to heat and moisture.

VSD is generally fitted to oil-injected screw and centrifugal machines. Speed reduction is not an option for reciprocating compressors because this may affect its lubrication system.

Note that a compressor that runs at full load will consume more energy if a variable speed drive is fitted. Ideally, when there are multiple air compressors at a facility, one or more fixed speed compressors should supply the base load and a VSD compressor should be used to supply the fluctuating or trim load.

Applicability of Air Compressor Unloading Controls

Table below shows the applicability of air compressor unloading controls.

Type of control	Reciprocating (single-acting)	Reciprocating (double-acting)	Lubricant-cooled rotary screw	Oil-free rotary screw	Centrifugal
Automatic start-stop	Yes	Yes	Yes	Yes	Yes
Two step	Yes	Yes	Yes	Yes	Yes (dual)
Three and five step	No	Yes	No	No	No
Throttled inlet	No	No	Yes	No	Yes
Variable displacement	No	No	Yes	No	N/A
Variable speed	No	No	Yes	No	No

Operating Cost Comparison of Different Control Modes

The compressor control mode can have a big effect on operating costs. In modulating mode the compressor would use 90% of full load power. For load/unload with minimal air storage (1 US Gal per cfm), the compressor would use about 92% of full power. By increasing the air storage to 10 US Gal per cfm, the load/unload compressor will use about 77% of full power. With variable speed drive control, the same size compressor will use about 66% of full power.

Table below shows the operating costs for a 100 HP compressor running at 65% average load.

Approximate Annual Cost for a 100 HP Compressor at Different Control Modes

% Load	Modulating	Load/Unload with 1 gal/cfm Receiver	Load/Unload with 10 gal/cfm Receiver	Variable Speed Drive
100	$36,130	$36,130	$36,130	$36,850
75	$33,420	$34,680	$29,350	$27,090
65	$32,330	$33,240	$27,820	$23,480
50	$30,710	$31,070	$24,200	$18,060
25	$28,000	$24,930	$16,800	$9,030
10	$26,370	$16,620	$11,740	$3,610

Based on 10 cents per kWh and 4,250 hours per year

Multiple Compressor System Controls

Local compressor controls independently balance the compressor output with the system demand and are always included in the compressor package.

The operating goals for plants having multiple compressors to feed a single air system are different. The primary goals are to automatically maintain the lowest and most constant pressure, through all flow conditions, while ensuring all running compressors except one are either running at full load or off. The remaining compressor (trim unit) should be the one most capable of running efficiently at partial loads.

To achieve the stated goals, systems with multiple compressors require more advanced controls or control strategies (cascaded pressure bands, network or system master controls) to coordinate compressor operation and air delivery to the system.

Cascaded Pressure Band Control

Cascade controller ensures only the number of air compressors required to satisfy demand are running. As compressed air demand rises and line pressure begins to fall, compressors with lower pressure bands come into operation increasing system output. When compressed air demand falls and line pressure rises, only the compressors with higher pressure bands will operate. This sounds fine in theory, but in practice the cascade concept controllers have a number of inherent drawbacks.

Drawbacks

1. The cascaded control method results in higher than necessary system pressures during partial loads which causes higher than required energy consumption. As the number of coordinated compressors increases, it becomes more and more difficult to achieve accurate compressor control without exceeding the pressure rating of the connected compressors at low loads or experiencing low system pressure at high loads.

2. A conventional cascade arrangement generally does not allow for 'fine tuning' of either compressor utilization and / or air system pressure and demand changes; such as shift pattern changes, weekend shutdowns or low demand periods.

3. With the cascaded pressure switch method of selection, compressors are managed in a very basic and routine. This restricts the user’s ability to select the optimum compressors to match air demand at a given point in time resulting in increased system energy costs.

Network Control

Network control uses the optional feature of the local compressor control to communicate with other compressors to form a chain of communication that makes decisions to stop/start, load/unload, modulate, and vary speed. One compressor generally assumes the primary lead with the others being secondary to the instructions from this compressor. Network control can accommodate many compressors while maintaining system pressure within a single lower pressure band for all flow conditions. The limitation is that these types of controls usually interconnect compressors of the same manufacturer.

System Master Controls (also called automatic sequencers)

Similar to network controls these externally installed controls interface with the local compressor controller to ensure system pressure remains within a single more efficient lower pressure band. Most system master controls can accommodate different manufacturers and types of compressors in the same system.

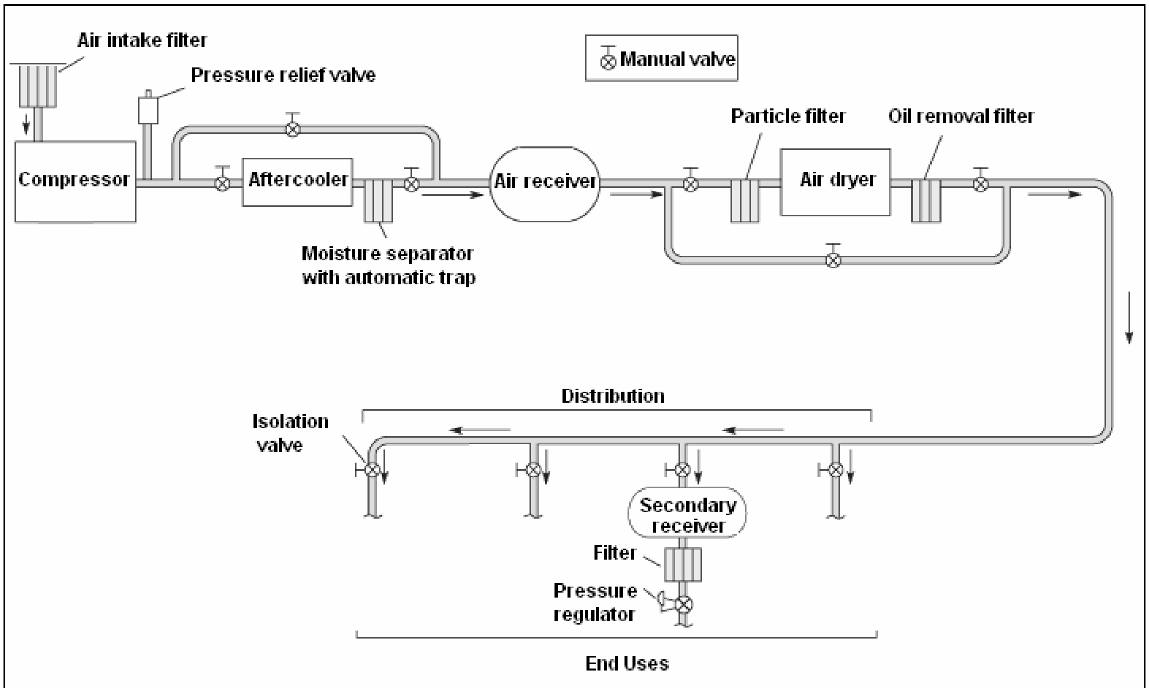
A PLC-based automatic sequencer allows for as many as eight compressors to communicate with one another and operate as a team as it follows a programmed schedule. The sequencers monitor and match compressor supply to demand. For example, it can select which compressors to use, shutting down those not necessary to plant operations, even choosing backup units as needed. An automatic sequencer can ensure a stable system pressure, allowing your entire operation to run as efficiently as possible, saving both time and money. PLC-based modular control systems can allow your plant operations engineers to monitor and perform diagnostic checks on your compressed air systems remotely, helping to predict and prevent systems malfunctions that could result in engineered-air downtime. These control systems should be easy to operate, resulting in less training time.

SECTION # 2 COMPRESSED AIR SYSTEMS AUXILIARY COMPONENTS

Compressed air auxiliary equipment includes compressor after-coolers, filters, separators, dryers, heat recovery equipment, lubricators, pressure regulators, air receivers, condensate drains, and automatic drains. They are devices associated with the air compressor and help to condition compressed air to the required specifications.

When selecting the compressors and sub-components of a compressed air system, keep in mind that life-cycle energy costs for a compressed air system are the greatest costs - and it's important to select components that maximize efficient use of compressed air.

The setup of a typical compressed air supply system is shown in Figure below. All these components can affect compressor efficiency.



Compressed Air System Diagram

Motors or Engines

Electric motors are the most common prime movers for all types of air compressors. Normally, most compressors are equipped with the standard O.D.P. Open Drip Proof motors. These motors require a roof covering to protect from weather exposure and should be located in an area away from sand blasting or similar activities.

NEMA 1 - An Electrical Enclosure suitable for indoor use. This is the standard for most units.

NEMA 4 - An Electrical Enclosure suitable for outdoor weather.

NEMA 7, 8, 9 - Any of a series of special “Hazardous Exposure” Electrical Enclosures. Used in the presence of Flammable gases or vapors.

Federal requirements have been increasing the minimum efficiency requirements for these motors, and they now routinely have efficiencies between 85 and 95 percent. However, as the efficiency is increased, the starting torque is decreased. Also, the operating speed of the motor must increase. Other common prime movers are diesel engines and natural gas engines. The benefit of engine driven air compressors is their higher efficiency when throttled for part load applications.

Primary and Secondary Air Receivers

A receiver tank is a vessel that store air needed to meet peak demand events with minimal effect on changes pressure. Air receiver tank serves various functions:

1. Damping pulsations caused by reciprocating compressors.

2. Supplying peak demands from stored air without needing to run an extra compressor.

3. Reducing load/unload or start/stop cycle frequencies to help screw compressors run more efficiently and reduce motor starts. Most screw compressors have internal protection that prevents more than 4 to 6 starts per hour.

4. To allow better compressor control and more stable system pressures.

5. Separates moisture and oil vapor, allowing the moisture carried over from the after-coolers to precipitate.

There are two types of storage – primary storage and secondary storage.

Primary storage is located close to the compressors and it reacts to any system event. Secondary storage, located close to an end use, minimizes the effect that a local high-volume, low time-duration event has on the upstream system.

A typical rule of thumb is to - *"size your primary air receiver tank at about one gallon capacity for every CFM of air compressor output".* For example if your compressor delivers 1000 CFM, then your receiver tank should be 1,000 gallons capacity. Other factors come into play when sizing are the type of air compressor, method of capacity control and compressor starting delays.

The location of the primary receiver can have a significant effect on the air dryer. Receivers located downstream of the air dryer can store large quantities of dry air for use in feeding peak demands. If there is a sudden demand in excess of the compressor capacity, the stored air can flow directly from the receiver to help maintain adequate pressure. If, on the other hand, the primary receiver is located on the upstream side of the dryer the combined flow from the compressor and the receiver must flow through the dryer. This can cause flows that exceed the dryer capacity. For this reason the primary receiver should be located downstream of the dryer and filters.

Secondary Receivers

Facilities having large fluctuations in air demand, or having insufficient air pressure (usually at the end of the line), should evaluate the need for one or more secondary air receivers strategically located in the air distribution system. Secondary receivers would be located very close to the point of air use at a piece of equipment that uses a large volume of air on an intermittent basis. Intermittent is the key word here. If you had a piece of equipment using a large volume of air on a constant basis, a secondary receiver won't do anything to help your system.

Typically, a receiver of about 110 US gallons will store 1 cubic foot of compressed air per psi. Required receiver size for any application is simply the cubic feet required multiplied by 110, and then divided by the pressure range.

Example -

A sand blasting operation requires 100 cfm of compressed air @ 80 psi for 1 minute every 10 minutes. The system pressure is 100 psi. Estimate the size of secondary receiver required to meet this transient load.

- Cubic feet required = 100 cfm x 1 minute = 100 cubic feet
- Pressure (psi) range = 100 – 80 = 20 psi
- Storage receiver required = 100 cubic feet × 110/20 psi= 550 gallons (US)

This receiver could be filled over 10 minutes at a rate of 10 cfm which would reduce the previous system pressure differential by a factor of 100. The inlet shall be restricted by an orifice or needle valve so that the storage tank can be refilled at a reasonable lower flow rate and won't affect other local pressure sensitive end uses.

Both primary and secondary storage also can help align supply with demand by minimizing the effects that air users have on the system. Air receiver tanks greater than 6" shall be constructed in accordance with ASME Boiler and Pressure Vessel Code Section VIII.

Intercoolers and After-coolers

Intercoolers

The function of an intercooler is to remove heat of compression between the stages of compression. When the air is compressed, it also sees a rise in temperature. This rise in temperature causes an increase in volume. An intercooler is used to cool the air so the volume is decreased and more air can be packed in. This improves efficiency of compression.

After-cooler

An after-cooler is a heat exchanger immediately downstream from the compressor. It removes the heat of compression from both the compressor lubricant and the discharge air. The proper operation of the air cooler is important because the moisture content of the air directly relates to discharge temperature. An after-cooler discharging saturated compressed air at 100°F will pass along 67 gallons of water per 1,000 standard cubic feet per minute ever 24 hours. If moisture is left in the compressed air system, this can condense in the piping, pneumatic tools and instruments, causing premature damage or failure.

The after-cooler shall be located between the compressor and air dryer as close to the compressor as possible. These coolers should be cleaned periodically to maximize the heat transfer capability for energy efficiency. Temperatures in excess of 38°C [100ºF] will generally overload air dryers and cause moisture problems. The compressors can also be classified according to the type of cooling. They are

Air cooled compressors - These compressors use the fan for forced cooling of the compressors. Due to the low cooling efficiency, this type of cooling is mostly used for low capacity compressors having intermittent usage.

Water cooled compressors - For heavy duty or continuous applications water cooling system is adopted, as the efficiency of cooling is high.

Circulating Water Cooling System

Air compressors that operate continuously generate substantial amounts of heat from the heat of compression. This heat needs to be removed both from the air after-cooler and from the oil cooler.

An adequate water flow through the intercooler, cylinder jacket, and after cooler is required for cooling the compressor, cooling the compressed air, and for moisture removal. A water flow sensing control (flow switch) is needed which verifies that sufficient cooling water is flowing before the compressor is allowed to start. Water for the after-cooler for liquid seal rotary compressors should be piped in series with the compressor. Water flow, prior to startup, for rotary screw compressors and rotary lobe compressors is not required.

Piping shall be designed to conform to the manufacturer’s recommendations. A strainer or filter should be used in the piping system to reduce fouling of the cooler system components.

Typical heat dissipation from intercoolers, cylinder jackets, and after-coolers is listed below:

Heat dissipation (Btu/minute/BHP total)

	Single Stage	Two Stage
Intercooler	None	20
Cylinder Jacket	15	5
After-cooler	26	17

The amount of cooling water required for intercoolers, cylinder jackets, and after-coolers may be determined as follows:

GPM = BHP * Heat dissipation / (T-rise * 8.33)

Where:

- GPM = gallons of water flow per minute.
- BHP = air compressor brake horsepower.
- Heat dissipation = value from table above

- T-rise = degrees F, water temperature rise.

To keep condensation from forming in the cylinder inlet ports, a differential of 15°F should be maintained between the temperature of the cooling water entering and the air temperature leaving the after-cooler]. This can be accomplished by circulating water through the intercooler first, and then piping the same water through the cylinder jackets. An alternate method is to reduce the water supply to the cylinder jackets. The compressor manufacturer should be consulted to verify the cooling water requirements for cooling compressor cylinder jackets.

Piping

Piping delivers compressed air from the compressor room to end use equipment and processes.

Most compressed air systems use galvanized, black steel or stainless steel piping - schedule 80 for sizes 2 inches and smaller and schedule 40 for sizes over 2 inches. Schedule 40 is suitable for pressures in the 175 psig range.

Copper compressed air piping or tubing shall be Type K or Type L.

Fiberglass reinforced plastic (FRP) may also be used within the following limitations:

- 150 psig maximum pressure, up to 200°F.
- 75 psig maximum pressure, up to 250°F.

PVC piping is relatively inexpensive, easy to install, lightweight, and corrosion resistant. However, PVC has one major drawback, it is brittle. An inadvertent impact could cause the piping to shatter, endangering surrounding personnel. PVC should is not recommended for above ground exposed piping.

Pipe fittings shall be galvanized or black steel or stainless steel, to match piping used. When copper pipe or tubing is used, brazed joints shall be used for connections. Brazing filler metals with melting temperatures between 1,000°F and 1,600°F shall be used. Soldered joints should not be used.

Sizing of compressed air piping is based on the allowable velocity of compressed air in the pipeline, keeping a check on the pressure drop. In compressed line if the pressure drop is high, the operating pressure at the generation end has to be increased to match with the requirement. This will result in increased power consumption of the compressor. The recommended velocity for interconnecting piping and main headers is 20 fps or less.

Valves

Although valves are used primarily for isolating a branch or section of the distribution network, they are also used for flow or pressure control.

Ball valves are recommended because they cause almost zero pressure drop when fully open. This is because the throat diameter of the valve is equal to the pipe bore. The quick action handle clearly indicates if the valve is open or closed. However, their purchase price is higher than some alternatives (e.g. gate valves).

Gate valves are often used due to their low purchase price. But, because their throat diameter is smaller than the pipe bore, they present a constriction and cause pressure drop. In addition, when set fully open, the sealing surfaces can erode over time, making it impossible to obtain an airtight seal. Gate valves are often left partially open due to the number of turns required to go from fully closed to fully open. The glands are often a source of leaks.

Some other valves such as **diaphragm** and **globe valves** cause large pressure drop and are not recommended for compressed air systems.

Separators and Drains

Water separators are devices that remove entrained liquids from the air. They are installed following after-coolers and are needed at all separators, filters, dryers and receivers. Poorly designed or maintained drains tend to waste significant compressed air.

There are four main methods to drain condensate:

1. **Zero air loss traps with reservoirs:** The most common type of zero air-loss traps is a float or level sensor that operates a ball valve through a linkage to expel the condensate in the reservoir to the low-level point. These are most efficient design as only condensate is expelled and are normally easy to test and maintain.

2. **Electrically operated solenoid valves:** The solenoid operated drain valve opens for a specified time based on a preset adjustable interval. The valve will operate even if little or no condensate is present, resulting in the loss of valuable compressed air. The solenoid-operated drain valve wastes energy –

   - If the period during which the valve is open may not be long enough for adequate drainage of accumulated condensate.

   - If set to drain worst case moisture loading this drain style will waste air during periods of lower moisture demand.

3. **Float operated mechanical drains:** Float-type traps do not waste air when operating properly, but they often require a great deal of maintenance and are prone to blockage from sediment in the condensate. Inverted bucket traps may require less maintenance but will waste compressed air if the condensate rate is inadequate to maintain the liquid level (or prime) in the trap.

4. **Manual drains:** Manual valves used to discharge condensate are often located at points where moisture problems are experienced. As these valves are not automatic, in many instances, manual valves are left partially cracked open allowing compressed air to constantly escape. This type of drainage should be avoided.

Air Dryers

Compressed air leaving the compressor after-cooler and moisture separator is normally warmer than the ambient air and fully saturated with moisture. As the air cools, the moisture will condense in the compressed air lines. Excessive entrained moisture can result in undesired effects like pipe corrosion and contamination at point of end use. For this reason some sort of air dryer is normally required.

Different types of compressed air dryers have different operating characteristics and degrees of dew point suppression (dew point is the temperature where moisture condenses in air). Air dryers can broadly be categorized into one of three types: refrigerated, desiccant and membrane type.

1. **Refrigerated dryers** remove moisture by cooling the air below the dew point using a cooling coil that condenses moisture out of the air. Refrigerated air dryers can produce air with a dew point as low as 33-39°F. Refrigerated dryers cannot operate below this range because the condensing water will freeze on the cooling coil of the dryer. By adjusting the refrigeration unit operating parameters, these units can produce pressure dew points of 50°F. Higher dew points are available in either direct refrigeration or chiller-type design.

2. **Desiccant dryers** dry air through the process of moisture adsorption with a desiccant material. If your processes require a 33°F or lower pressure dew point then a desiccant dryer would be appropriate. Desiccant dryers are typically designed to produce dry air with a dew point of –40°F and are capable of supplying air down to dew points of –150°F.

   Desiccant dryers require regeneration, which requires inlet air to be automatically cycled between two desiccant towers: the one absorbing moisture from the inlet air while the other is being regenerated to maintain its adsoption capabilities. Typical desiccant materials are silica gel, molecular sieve of crystalline metal alumiosilicates and activated alumina. The activated alumina is the most commonly used media.

   There are two methods of regeneration:

   <u>Heatless desiccant dryers</u>

Heatless desiccant dryers typically use two identical drying towers, each containing a desiccant bed. While one tower is drying the compressed air the other tower is regenerating by purging a certain portion of the dried air coming from the active desiccant dryer through it. The purge air requirements can range from 10-18% of the total compressed airflow, making them very inefficient compared to other dryers.

Heated desiccant dryers

Heat regenerative dryers utilize heat from an external source (either electric or steam) in conjunction with purge air to regenerate the off-stream tower. By reducing the amount of purge air, the heat regenerative dryer operating costs are lower. High regenerative temperatures are however damaging to equipment and desiccant, so any savings in operating costs can be outweighed by the costs of maintenance and downtime.

3. **Membrane Dryers**

These units use a semi-permeable membrane to separate water vapor from the air stream. They have no moving parts. The units use about 20% or the nameplate rating to sweep the membrane. This sweep air is lost to the air system. These dryers exhibit variable dew point output depending on the flow of air and the temperature.

Filters

All air compressors are sensitive to dust and airborne vapors. These contaminants build up in rotating parts and can induce excessive wear and mechanical unbalance, thereby damaging the compressor.

An important filter within the system is the intake filter for the compressor. This filter removes dust and other particulates from the intake air feeding the compressor.

Compressed air filters downstream of the air compressor are generally required to remove contaminants, such as particulates, condensate, and lubricant. Numerous choices for filtering exist depending on the cleanliness of the air required.

Types of Filter

The selection of the filter type is based on whether the air compressor is lubricated or non-lubricated.

1. Viscous impingement filters have an efficiency of 85 to 90 percent of particle sizes larger than 10 microns. This type of filter is acceptable for lubricated reciprocating compressors operating under normal conditions.

2. Oil bath filters have an efficiency of 96 to 98 percent of particles sized larger than 10 microns. This type of filter is more expensive, and for the most part no longer recommended by compressor manufacturers, but may be considered for lubricated reciprocating compressors operating under heavy dust conditions.

3. Dry filters have an efficiency of 99% of particles larger than 10 microns. Because of their high filtration efficiency, these filters are the best selection for rotary and reciprocating compressors. They must be used for non-lubricated compressors and whenever air must be kept oil-free.

4. Two-stage dry filters provide 99 % efficiency of particles larger than 0.3 micron and are recommended for centrifugal units.

5. Coalescing filters are used to remove lubricant and moisture. Coalescing filters combine aerosols into larger droplets within the filter. These droplets eventually achieve sufficient size to fall out of the filter into the bottom of the device for draining. An example of coalescing filters is glass fiber media. This material is neither absorbent, nor adsorbent. It will retain its dry proper-ties throughout its useful life, which may be compromised by oils and particulate matter.

With all types of filters, a means of monitoring the air pressure drop through the element must be provided, which indicates element contamination.

Energy Efficiency

From an energy efficiency perspective, air filter types should be chosen carefully as there is an energy penalty for over filtering. A given filter pressure differential increases to the square of the increase in flow though it. This filter differential increases the compressor energy required to produce a fixed downstream pressure.

About 1% in higher energy costs results from every 2 psi in filter differential. If a given filter capacity is doubled the pressure loss across it will reduce by a factor of 4, for a 75% savings.

To save energy, where possible, minimize the filter pressure drop by using low differential mist eliminator style filters, oversized filters, or by using filters installed in parallel.

Filter Maintenance

Maintenance of filters is critical when operating an efficient system. A clogged filter increases the flow resistance causing increased pressure drops and consuming additional energy.

It is important to monitor pressure drop across filters to determine whether a filter element is in need of being replaced. Pressure gages or sensors should be placed upstream and downstream of the filters to determine when a filter element requires cleaning or replacement. The pressure drop of 6-10 psi indicates the need for a filter element to be replaced or cleaned.

SECTION # 3 CHOOSING THE RIGHT COMPRESSOR PACKAGE

Compressor package selection is arguably the most important component in a compressed air system and can dramatically influence equipment, maintenance, and energy costs of the system.

The choice of a compressor package is based upon several key factors. These include but not limited to the total expected demand in CFM on a routine daily basis, the duty cycle of the load demand verses the designed duty cycle of the unit, required system pressure, brake horsepower (bhp) per 100 cubic feet per minute (cfm), unloaded horsepower, expected compressor life, specific air treatment requirements and expected operation and maintenance costs. Emphasis should be on life cycle cost.

Maximum compressed air consumption

As a first step towards compressor selection, you should know the quantity of compressed air required for his plant. To estimate this quantity, the following should be determined.

Maximum compressed air consumption is the total quantity of compressed air required by all the pneumatic equipments connected in the plant, operating in full load condition. To estimate this compressed air consumption, the air consumption per unit depending on the equipment connected should be considered. Wherever process air is used due consideration has to be given.

Compressors capacity is rated in CFM (cubic feet per minute). There is no universal standard for rating air compressors, air equipment and tools. Common terms are:

1. CFM - CFM (Cubic Feet per Minute) is the imperial method of describing the volume flow rate of compressed air. It must be defined further to take account of pressure, temperature and relative humidity - see below.

2. ICFM - ICFM (Inlet CFM) rating is used to measure air flow in CFM (ft^3/min) as it enters the air compressor intake.

3. SCFM – Standard Cubic Feet per Minute (SCFM) is a volumetric flow-rate corrected to standard-density conditions. SCFM is volumetric flow-rate at a standardized pressure, temperature, and relative humidity. American Society of Mechanical Engineers (ASME) standards define standard conditions at 14.7 psia, 68°F and 36% relative humidity. This converts to a density of 0.075 lbs /cu- ft for air.

4. ACFM – Actual Cubic Feet per Minute (ACFM) is the volume of gas flowing anywhere in a system independent of its density. If the system were moving air at exactly the "standard" condition, then ACFM would equal SCFM. Unfortunately, this usually is not the case as the most important change between these two definitions is the pressure. To move air, a positive pressure or a vacuum must be created. When positive pressure is applied to a standard cubic foot of air, it gets smaller. When a vacuum is applied to a standard cubic foot of air, it expands. The volume of air after it is pressurized or rarefied is referred to as its actual volume.

5. FAD - FAD (Free Air Delivery) is the actual quantity of compressed air as measured at the discharge of the compressor. The units are measured according the ambient inlet standard conditions ISO 1217. ISO 1217 - standard reference ambient conditions - temperature 20^{o}C, pressure 1 bar abs, relative humidity 0%, cooling air/water 20^{o}C, and working pressure at outlet 7 bar absolute.

To size a compressor the capacity must be stated as the volume it will occupy at the compressor's suction. This volume is normally referred to as inlet cubic feet per minute (ICFM). If the term ACFM is used, it must be made clear the volume is measured at suction pressure and temperature and not some other conditions.

The confusion surrounding the measuring of a volume of gas is due to the fact that gasses are compressible. This simply means that a given number of gas molecules may occupy a vastly different volume depending on its pressure and temperature. A 60 gallon vessel contains significantly less gas at 50 psig than at 200 psig even though the size of the vessel remains constant. Specifying a capacity of 15 CFM does little except create confusion unless a reference pressure and temperature are also specified or implied.

When you express your "demand" in SCFM, you are saying that you want this compressor to deliver this CFM even at your worst case conditions. If you have a "demand" of 500 SCFM and you pick a unit from the manufacturer's literature that indicates a "capacity" of 500 ACFM, you will not get the amount of air that you require during times when your inlet conditions vary from the standard conditions. Corrections must be made to assure that the unit furnished will provide the proper amount of air for the process to function properly.

How to Convert ACFM to SCFM

SCFM = ACFM x (Actual Inlet Pressure /14.5) X (520 / (Actual Inlet Temperature + 460) X RH% correction (.995 to .97)

Example: 500 ACFM compressor at 14.3 psia and summer conditions 95°F and 60% RH

SCFM = 500 x (14.3/14.5) x (520/(95+460) x .97 = 448 scfm

Utilization Factor (Use Factor or Load Factor)

Use factor or load factor is the ratio of actual air consumption in a plant to the maximum continuous air consumption.

$$\text{Load Factor} = \frac{\text{Actual air consumption in 24 hours}}{\text{Maximum continuous air consumption in 24 hours}}$$

Load factor plays a vital role in estimating the total compressed air requirements at the design stage. In any industry, where a large number of pneumatic tools / application are involved, all may not be operating simultaneously. In these cases, the use factor is of immense help to the factory manager, to determine the approximate average compressed air consumption. This use factor can be determined with the help of work-study or it is best estimated by evaluating experiences of similar plants. In cases where use factor for each class of machines cannot be estimated, it is a common practice to use an overall factor of 50%, of the total consumption of all machines in the plant.

Average Consumption of Air

Average consumption of air can be estimated, considering both the maximum air consumption and the utilization factor. *A provision of 5% of the total average capacity requirement should be made to cover leakage losses.* Also a provision of 10% of the total average capacity requirement should be given for unaccounted usage and future expansion. The effect of altitude should also be given due consideration during the estimation. The estimated air consumption should be 115% of the calculated average air consumption.

Pressure (psi or pounds per square inch)

System pressure depends on user requirements, controls, and safety valves.

An unregulated compressor will keep increasing pressure until a failure occurs.

When plant capacity demand exceeds system capacity (CFM), compressor discharge pressure will drop.

The Pressure - Capacity relationship is expressed as:

$P1 \times V1 = P2 \times V2$

Where:

- P1= Initial pressure
- V1= Initial capacity
- P2= Final pressure
- V2= Final capacity.

A CFM rating at 40 psig will always be a higher value than at 100 psig or 175 PSI.

It is important to note that a trending decrease in plant air pressure typically indicates a requirement for more capacity (CFM), not for more pressure. Increasing or decreasing the existing compressor discharge pressure will normally have negligible effect on the compressor capacity.

Air Quality and Lubrication System

When selecting a compressor consideration needs to be given to the level of air quality required. If lubricant-free air is required, this can be achieved with either lubricant-free compressors, or with lubricant-injected compressors that have additional separation and filtration equipment. Lubricant-free compressors usually cost more to install and have higher maintenance costs.

Note that the oil less non-lubricated compressors have the advantage of not contaminating air, however, the oil in an oil-flooded compressor acts as a sealant between the male and female rotors and carries away heat from the inside of the compressor. This makes the oil-flooded compressor more efficient.

System design will be in accordance with the manufacturer’s recommendations. Lubricant type will depend on the compressor application:

1. Gravity, splash, or pressure petroleum oil should be specified where oil contamination of the compressed air at the point of use is not a problem.

2. Synthetic liquid lubricants should be used where there is a danger of fire, where the carbonaceous deposits must be reduced, or where lubricant is provided for extended maintenance periods.

3. Solid lubricants, such as carbon or Teflon piston rings, should be used for oil-free reciprocating compressed air applications.

Number

A key issue in planning a compressor installation is whether there should be a central compressor plant or a number of separate compressors near to the main points of use. An economic evaluation is necessary to determine which one is most cost-effective. Seasonal or operational load variations must also be considered. The efficiency of larger compressors is generally higher than that of smaller units, but use of smaller air-cooled units permits savings on water, water piping, and system losses. Multiple units with interconnecting piping give flexibility for maintenance shut-down of one compressor. A smaller air compressor to handle requirements for weekends, holidays, and other low usage times is generally economical. Follow the table below for advantages and disadvantages of centralized v/s decentralized systems.

Centralized Systems	**Decentralized Systems**
Advantages	**Advantages**
Capital cost per unit output generally falls with increased capacity of centralized plant.	Low capital cost, savings made on minimizing the distribution systems.
Tends to be better engineered, operating at higher efficiencies (where load factors are high) and more durable.	Systems can be zoned to more closely match the demand patterns.
Some systems will naturally require centralized plant, e.g. to meet very large volumes.	Can be readily altered and extended.
Possibly a higher efficiency, and thus lower running costs due to large units.	Output and/or pressure can be varied to suit each particular plant section.
Energy performance for processes with diverse patterns of use is usually better.	Pipe sizes and lengths can be reduced, thus minimizing leakage and cost.

Heat recovery potential may be greater due to larger centralized plant, particularly if hot water is required.	Compressors and/or associated equipment can be shut down during periods of low demand or for maintenance, with only a localized effect.
Greater security of supply due to built-in standby of multiple compressors	Heat recovery may be simplified due to individual compressors being close to heat use.
Condensate collection simplified by grouping to one system.	
Disadvantages	**Disadvantages**
Space requirements of centralized plant and distribution systems are significant.	Equipment tends to be less robust with shorter operational life.
Leakage losses will be greater due to larger distribution network.	Quality of control, maintenance and air quality may be inferior to central plant systems.
Capital costs of distribution systems are high.	Smaller machines tend to be less efficient.

Multi- Compressors (Pattern of demand)

The demand for compressed air varies widely from factory to factory depending on what the compressed air is used for. Demand patterns can be relatively constant, stepped or widely fluctuating.

As compressors are most efficient when operating at full load, it is more efficient to use a combination of compressors and controls (including variable speed technology) to meet the varying demand than to use one large compressor running at part load for most of the time.

While selecting the compressors, the following points should be kept in view:

1. As a first step, identify the base load and fluctuating load.
2. For base (steady) loads select centrifugal compressors (best for very high capacity) or screw compressors.
3. For fluctuating loads select screw compressor with built-in VFD (the best option) or reciprocating compressors.

The trim compressors do not have to be the same size and capacity as the base load compressor. As an example a trim compressor may have to respond to +30% of the base load compressors capacity only. Air compressor motor loading will become clear, only after the completion of your load measurements. If your base load compressor is a 1,000cfm unit, your trim compressor may only need to be sized at 250cfm.

Multi - Staging

Multistage compression can be used to reduce power losses associated with the air temperature rise during compression. Also, compression efficiency will be increased using multi-staging. The air compressor unit, however, will increase in cost and will be a more complicated machine. Before selecting compressor staging, an economic evaluation should be performed with consideration given to the required air pressure levels and the intended compressor use. When using multistage compression, intercoolers should always be considered to improve the efficiency of the air compressor unit.

Operating Costs

Energy costs account for 80 percent of total purchasing and operating costs over the life of compressed air system. A typical compressor installation of 1000 cfm will have a capital cost of $80000, and when running at full load (200 kW) continuously, it would cost as much in energy consumption in the first seven months of use as it did to buy.

As a result, there has been a push towards implementing equipment, controls, and maintenance systems, which allow owners of compressed air systems to get the greatest value from their energy investment dollars while maintaining the functional integrity of the system. It is recommended to make purchase decisions on the overall expected lifetime operating costs, and NOT just on the initial cost of the equipment.

If your compressed air system runs at a fairly constant load near 100% of capacity, a variable speed drive will not improve your total energy consumption and may end up costing you more in energy costs because of some additional losses of variable speed drives. It is important to understand that the savings from a variable speed drive accrue when the compressors are partially loaded. With the decrease in cost in variable speed drives for electric motors, these devices have recently become very common when handling fluctuating air demand.

Total life cycle cost and benefits must be weighed carefully before selecting the most cost-effective option, not only for the compressed air supply system but also for the end uses.

Air Compressor Efficiencies

The energy efficiency of an air compressor can be defined as the **ratio of compressed air output to input power.** In normal operation, this ratio varies over time in response to varying loads and other factors, such as discharge pressure and the temperature of the inlet air. The most accurate method of determining average compressor efficiency is to directly measure the input power and compressed air output over time.

The input power can be measured using power recorders or clip on clamp meters. Compressed air output can be measured by inline or non-intrusive flow meters. In-line flow measurement is fairly common in very large plants and in applications where a plant purchases compressed air from a supplier. Non-intrusive flow meters are somewhat expensive and their accuracy is dependent on proper placement and other factors.

Efficiency usually is expressed as brake horsepower per 100 cfm of delivered air

Depending on the type of compressor used, most compressors are typically rated to deliver **four** SCFM per horsepower (rule of thumb). The industry norm for comparison of compressor efficiency is given in terms of bhp/100cfm (brake horse power per 100 cubic feet per minute) at a compressor discharge pressure of 100 psig.

Compressor Specifications

To ensure energy efficient compressors are purchased for a given duty, a specification should be written against which qualified suppliers can offer a proposal. Suppliers should be advised that bids will be analyzed from an energy efficiency and lifetime cost of ownership viewpoint.

A specification should always include the following:

1. Background information about the site.
2. The scope of supply
3. The duty in terms of mean, peak and minimum demand
4. The range of site ambient air temperature and pressures expected.
5. The mean site ambient air temperatures and pressures expected

6. The maximum site cooling air or water temperatures expected

7. The height above seal level of eh site

8. The standby strategy

9. The minimum pressures required at the usage points

10. The air quality require at the usage points

11. The ancillary equipment needed (starters, isolators, local and remote controls, annunciators and other items)

12. The noise level required of all items

13. The hours to run each week and the number of weeks per year

Suppliers should be asked to provide the following information with the proposal:

1. The type of machine and configuration offered

2. The unit size in terms of rated output

3. The conditions of air temperature, air pressure, relative humidity and cooling temperature under which machine is rated

4. The output of the machine in SCFM (given the mean site ambient air pressure and temperature)

5. The number of units offered

6. Is the compressor water-cooled or air-cooled?

   - If water-cooled, what is the volume and pressure of the cooling water required and what is the water quality specification?

   - If air-cooled, what is the cooling air volume and pressure capacity of the compressor cooling fan?

7. The air treatment system offered (i.e. type of dryer, number and type of filters)

8. The required delivery pressure at the compressor discharge, taking treatment system losses into account
9. For oil-injected compressors, what type of condensate separation equipment will be required?
10. The power consumed at the compressor shaft at the required delivery pressure
11. The method of control
12. The part load power consumptions
13. If a variable speed machine, what is the total input power and FAD at the stated delivery pressure at 75%, 50% and 25% speed? What is the minimum flow and number of starts per hour allowed?
14. What motor speed has been assumed for the performance data? Is it typical of normal operating conditions?
15. What are the recommended lubricants?
16. The cooling system power including all pumps, fans and heaters
17. The actual power of each drive motor and the total package electrical input power
18. Type of test employed
19. Tolerances on flow, power and specific power at full and part load
20. Full maintenance costs over five/ten years
21. What are the conditions of ensuring warranty validity?
22. Itemized prices

Compressor Evaluation Summary

The table below shows the advantages / disadvantages of different compressor types:

Compressor/ control type	**Advantages**	**Disadvantages**	**Good application**	**Poor application**
Reciprocating/ unloading	Very good part-load efficiency	Maintenance intensive	Trim compressor	Sterile air
Rotary screw (oil)/ inlet throttling	Minimum maintenance cost, low initial cost	Poor unloading efficiency	Base or continuous loading	Trim compressor
Rotary screw (oil)/ rotor shortening or load-unload	Low maintenance cost		Trim compressor	Sterile air
Rotary screw (oil)/ variable speed drive	Very good part-load efficiency	High initial cost	Trim compressor or fluctuating load	Full or base loaded
Rotary screw (oil-less)/load-unload	Very good part-load efficiency, oil-free	Not as efficient as oil-flooded	Trim compressor, sterile air	Industrial uses
Centrifugal/ inlet throttling	Inherently oil-less, low maintenance costs	Poor part-load efficiency	Base or continuous load	Trim compressor or part-loaded

Summarizing…..

The type of compressor you choose will depend on your system pressure, capacity, quality requirements and the shape of the demand pattern.

1. Systems with steady, high demands might opt for a series of centrifugal machines.
2. Systems with fluctuating loads might opt for reciprocating or screw compressors with variable drive.
3. Systems requiring extremely high quality air should opt for oil-free, non lubricated compressors

SECTION # 4 COMPRESSED AIR SYSTEM ASSESSMENTS

Evaluating your compressed air system is the first step in improving its energy efficiency performance. Facilities may undertake compressed air system assessments using in house expertise or, possibly, through a qualified consultant or contractor. The common steps in establishing a compressed air system improvement program include:

Gathering Equipment Data

A first step in the process is gathering equipment data. This can be found by recording nameplate data, service records, operating manuals and purchase orders. This inventory should include recording the nameplate information and setpoints for all of the equipment in the compressed air system including the air compressor(s), after-coolers, air dryers, receivers, filters, and controllers. A sketch should be made of your compressed air production and distribution system layout noting the pipe sizes, air take off points, and valves. The type and characteristics of machinery or tools along the route of the compressed air system should be recorded.

Establishing a Baseline

Establish the baseline performance. A compressed air load profile indicates how demand for air and compressor energy consumption changes over time. A facility with short periods of heavy demand may benefit from implementing storage options, whereas a facility with a varying load profile will likely benefit from advanced control strategies.

The following measurements, assessments and calculations are normally included in a system assessment:

1. Air pressure measurements over time.

2. System pressure differentials at various locations between the compressor discharge and the important end uses.

3. Compressor Amps or kW vs. time. (Note: only properly qualified personnel should undertake electrical measurements.)

4. System flow (either calculated or directly measured) preferably over time. This can be easily calculated using loaded vs. total run time for compressors with hour meters.

5. Ambient and compressed air temperatures.

6. Calculated operating costs for electricity, (water or chilled water), maintenance and taxes based on the gathered data.

7. System leak identification and measurement.

8. End use equipment pressure drops or differentials.

9. Identification of inappropriate uses of compressed air

10. Assessment of air filtration systems for pressure drops and effectiveness.

11. Evaluation of air storage receivers.

12. Assessment of air dryers (required dew points, energy consumption and pressure drops).

Analyzing Performance Data and Establishing Performance Levels

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Once measurements are taken and performance standards established, the data can be analyzed to determine if the system is meeting the facility's needs. The analysis will point to areas of deficiency and identify potential opportunities for improvement.

Areas to evaluate include:

1. Compressor type, size and condition - The compressors are evaluated for appropriateness of the intended use as well as overall condition. Compressor efficiency can be estimated from manufacturer specifications that are corrected to site conditions. The compressor installation is also evaluated for location, air intake, ventilation, and heat recovery.

2. Primary and Secondary Receivers - The effectiveness of the receiver tank should be evaluated for location and size. For the most part, the air compressors should be able to supply the plant's air needs, except for short periods of high demand that can be supplied by one or more receivers. Secondary air receivers to control demand events should also be investigated.

3. Compressor Controls - Check for appropriate pressure set points. In the case of multiple compressors, the pressure bands to trigger the start or stop of a compressor need to be adjusted.

4. Filters should be examined for cleanliness and appropriateness for the application. Pressure drops across the filters should be evaluated to estimate energy losses attributable to the filter. Check the appropriateness of maintenance schedules for changing the filters. Consider purchasing higher performance filters.

5. After-cooler and moisture separator efficiency and cooling effectiveness can be measured and feasible modifications or alternative systems recommended.

6. Dryer appropriateness needs to be assessed based on the facility's end use for compressed air. It is important to note the dryer size, pressure drops, overall dryer efficiency, and consider dryer modifications based on the volume and quality of air requirements.

7. Automatic Drains - The location, condition, and effectiveness of all drains needs to be evaluated and energy efficient alternatives recommended where appropriate.

Other areas to consider are:

- System pressure stability (is the plant having pressure problems)

- System specific power (how many kW does it take to produce 100 cfm) or how many dollars does it cost per 100 cfm
- Dewpoint stability (is there water in the air)
- Peak, minimum and average flows (can the production system to adequately supply these flows)
- Peak, minimum and average compressor room temperatures (can the compressors and dryers operate adequately in these conditions)
- Maintenance and operating costs per year and per hour of operation (is it costing more to maintain a compressor than to purchase a new unit)

Devising a Plan

Once peak and average flows are known and performance levels established it is possible to calculate energy savings numbers based on various alternatives.

Some things that could be considered include:

1. Identification of equipment that can be shut down
2. Selection and use of compressor and flow controllers
3. Opportunities to downsize or purchase new equipment where appropriate
4. Evaluation to minimize compressed air equipment operating hours
5. Proper selection of air compressors (number of stages, type of air compressor, and control modes)

Points to Consider for Compressed Air Audit

An audit should examine air production and air use. It should also investigate the manner in which it goes from supply to each end use. The cost side of an audit should measure the output of the system, and calculate the energy consumed and annual cost.

The auditor should also address system issues. System issues involve the entire system, not just individual parts. The issues most often addressed are:

- Level of air treatment (and efficiency)
- Leaks
- Pressure levels
- Controls
- Heat recovery

On the demand side, issues most often addressed are:

- Distribution system
- Load profile
- End-use equipment

On the supply side, issues most often addressed are:

- Compressor package
- Filters
- After-coolers
- Dryers
- Automatic drains
- Air receivers

- Storage

On the supply side, the efficiency of the receiver package as well as the individual components of air treatment should be examined for efficiency, expected life, type, and application.

When hiring energy auditor to undertake a compressed air assessment or to plan a new system expansion. Here’s a list of questions to think about in helping you make your decision:

1. What’s the track record and knowledge level of the firm and individual who will undertake the work?

2. How well does the service provider understand energy efficiency and economic tradeoffs?

3. How familiar is the service provider with all aspects and types of compressed air, including air supply, and air demand?

4. How well does the service provider understand my industry and the products we manufacture or process?

5. How objective will the report or advice be? (e.g., are they just trying to sell us more equipment or services, or is the work being done impartially and independently?)

6. How responsive is the service provider? (Availability to do the testing to minimize impacts to the facility and/or undertake the testing during nights/weekends)

7. How responsive is the service provider to health and safety practices and procedures?

SECTION # 5 ENERGY EFFICIENCY MEASURES

The adage, “If you can’t measure it, you can’t manage it,” applies to establishing your baseline. While temperature and dew point are useful air system measurements, the key metrics are pressure, rate of air flow and electrical consumption. This trio helps to determine the cost, monitor system operation and establish a baseline for evaluating future modifications.

This section describes time-proven measures to improve the energy efficiency of compressed air systems, including:

- Compressor sizing
- Reduce system pressure to a minimum
- Minimize system pressure drop
- Improve compressor efficiency
- Control of compressors
- Use cooler inlet air
- Detect and repair leaks
- Use heat recovery
- Reduce inappropriate and unnecessary uses, and
- Improve routine maintenance
- Optimization of air production equipment

Appropriately Size your Compressor

The total energy consumption of the compressed air system depends on correct type of size of compressors. If the installed compressor capacity is much higher than the compressed air requirement of the plant the compressor often falls in unloading mode.

The loading/unloading of the compressor is done based on the receiver pressure. If the compressor air delivery is more than the requirement of the plant, the system pressure increases. Once the system pressure reaches set pressure the compressor gets unloaded.

The compressor remains in the unload mode till the system pressure drops due to compressed air consumption in the plant and reaches the set load pressure.

The unload power consumption of the compressors is significant. In case of reciprocating compressors the unload power consumption is in the range of 15-20% of load power consumption. For screw compressors the unload power is still higher, which would be in the range of 30-35% of load power consumption.

Reduce System Pressure to a Minimum

When designing and operating a system it is important to correctly evaluate the amount of pressure required. Air must be delivered to the point of use at the desired pressure and in the right condition. Too low a pressure will impair tool efficiencies and affect process time. Too high a pressure may damage equipment, and will promote leaks and increase operating costs.

Many industrial plants run at unnecessarily high pressure, which wastes energy and increases running costs. For example, some systems operate at an elevated pressure of 100 psi at full load when the machinery and tools can operate efficiently at a lower air pressure of 90 – 70 psi. The extra 10 – 30 psi would be responsible for approximately 5% -15% of the plant’s increased energy costs.

Most often, the compressor discharge pressure is set artificially high for variety of reasons, such as:

1. **To compensate for high pressure drops** –

   In some cases, the elevated compressor discharge pressure is set to account for the higher pressure drop through the components of the air treatment and distribution system. *In a well-designed compressed air system, the pressure at the end use should be at least 90 percent of compressor discharge pressure.* Virtually every component of the compressed air system downstream of the compressor can be a source of pressure drop, such as dryers and filters on the supply side and undersized distribution piping, equipment hoses, disconnect couplings, filters, regulators, or lubricators on the demand side. If you find pressure at the end use significantly below 90 percent of compressor discharge, work upstream one component at a time to identify where the major pressure drops are occurring. When specifying or replacing this equipment, always ask manufacturers to provide information on pressure drop at the maximum anticipated flow rate and select equipment that minimizes pressure drop. And be sure to clean or replace filter elements regularly.

2. **To compensate for intermittent consumers –**

   Sometimes, elevated pressures are maintained to compensate for unacceptable pressure drops that would otherwise occur due to large, intermittent compressed air consumers on the same distribution system. One example is an air-driven agitator used for a settling pond. Three minutes of agitation at 150 cfm are required every 30 minutes to promote digestion. The pressure drop, measured for the cycle is from 90 psig to 40 psig.

   A fix for this problem is adding secondary receiver at or near the point of use to smooth out system wide pressure fluctuations. For the example above, the volume of receiver should be at least:

   The total air, based on the inlet pressure, required for agitation is 450 ft^3 (3 minutes × 150 cfm).

   Therefore, the volume of the receiver should be at least:

   $VR = (450)\ (14.7)/\ (40) = 165\ ft^3$

Energy Loss at Elevated Pressures

Compressing air to a high pressure and then regulating down to site equipment is wasteful for three reasons.

1. Higher the pressure, higher is the power consumption. It takes more compressor energy to pump air to higher pressure. A rule of thumb for systems operating at about 100 psig range states: "For every 10 psig increase of pressure in a plant system, energy consumption will increase by approximately 5% at full output flow".

2. Higher the pressure, higher is the air consumption. The high pressure system will use more air. If there is no resulting increase in productivity, air is wasted. Increased air consumption caused by higher than needed pressure is called "artificial demand".

3. Higher the pressure, higher is the air leakage. At 80 psi, about 21.4 cfm will flow through a leak with a diameter of 1/8 inch. At 100 psi that flow would increase by over 20 percent to 26 cfm wasting thousands of dollars annually.

Reducing system pressure to the minimum that is absolutely necessary should be the first step in system optimization. Often it's just a matter of simply readjusting the compressor control setpoints to a lower level but as a caution, this should be done carefully and in small steps so as not to affect sensitive plant equipment.

How to Calculate Costs due to High Supply Pressure?

All air tools are rated for their flow and optimum pressure. The air wastage can be calculated by using the pressure ratio (absolute), and then multiplying by the rated air flow i.e. if consumption at 3 barg is 8 cfm at 7 barg this will be:

$$[(7 + 1) / (3 + 1)] * 8 = 16 \text{ cfm}$$

This can then be substituted into the annual wastage formula to calculate savings.

Example

10 air tools rated @ 4 barg consumes 15 cfm each. How much air will be used if the pressure is 7 barg?

Solution

The air consumption of each drill at 4 barg is 15 cfm.

At 7 barg each tool will be consuming:

(8 / 5) x 15 = 24 cfm

So by using a lower pressure there is a potential saving of 24 – 15 = 9 cfm per tool.

Segregate HP & LP Compressed Air System

Higher the pressure, higher is the power consumption. While calculating the average compressed air consumption of the plant, the total requirement of low pressure (30 to 50 psig) and high pressure (above 50 psig) compressed air has to be estimated. If any, say LP or HP air constitutes more than 30 % of the average compressed air consumption then separate compressed air system has to be installed. The advantage of segregating HP & LP compressed air user has many advantages. These are:

- Reduces the leakages proportionally, as the leakage levels are high at higher pressures.
- Reduces the overall operating cost. Say a 20 % reduction in pressure results in 20% reduction in power consumption of the compressors. Moreover, the wear & tear of the compressors are less at low pressures.
- Increases the life of instrument valves, as higher pressure tends to damage the joints, packing etc., frequently
- Reduces the investment on pressure reducing valves at design stage itself.

Minimize System Pressure Drop

The compressor must produce air at a pressure high enough to overcome pressure losses in the supply system and still meet the minimum operating pressure of the end use equipment. As a result, it is not uncommon for a compressor to be delivering air at a pressure of 115 psig while the pressure at the point of end use is only 90 psig. This pressure drop of 25 psig through the system represents wasted energy and money. Note that every 2 psig of pressure drop represents a 1% increase in compressor energy costs.

In a properly designed and installed system, pressure drop should be less than 10% of the compressor's discharge pressure, measured from the point of discharge to the point of end use. Thus at a discharge pressure of 100 psig, the pressure drop should be less than 10 psig.

In compressed air systems, the typical pressure drops are:

1) Filter pressure drop @ 0.5 to 1.5 psig

2) Dryer pressure drop @ 2 to 3 psig

3) Piping pressure drop @ 3 to 4 psig

We will address two main areas where the pressure loss is maximum.

Pipe-work

Pressure drop in a pipe work is due to airflow resistance caused by pipe friction and various components within the system (e.g. valves, bends). If the pipe is too small for the volume of flow the velocity of the air will be very high and there will be a big loss in power. Energy is also lost when there is a change in flow direction i.e. elbows, junctions and shut off valves. Simple pipe systems will minimize pressure drop. Key points are listed below:

1. Straighten the path. Compressor location should be selected, so as to minimize the length of piping between the air compressor and the largest user of compressed air user. In systems with a large distribution network, it is preferable to have compressor centrally located.

2. Horizontal lengths of distribution piping should be sloped slightly downwards, with provision for moisture drainage.

3. Use larger diameter pipes to take advantage of lower pressure differential. When designing piping, it is often a good practice to add 30% to the expected air flow (to add for future potential system expansion), and then select the pipe diameter having the lowest pressure drop.

4. To minimize energy loss from pressure differential and to help stabilize the end of line air pressures, the distribution system should be sized for no more than 2-3 psi pressure differential.

5. Excessive velocity can be a root cause of backpressure, erratic control signals, turbulence and turbulence-driven pressure drop. The British Compressed Air Society suggests that a velocity of 20 fps or less prevents carrying moisture and debris past drain legs and into controls. A velocity greater than 30 fps is sufficient to transport any water and debris in the air stream. Thus, the recommended design pipeline velocity for interconnecting piping and main headers is 20 fps or less and never to exceed 30 fps. For short branch lines less than 50ft the velocity can be up to 50 fps.

   <u>Equation</u>

   First, look at the velocity at maximum anticipated flow conditions using the following equation:

   $V = 3.056 * Q/D^2$ -------------------- (Eq. 1)

   Where

   - V = air velocity (ft /sec)
   - Q = volumetric flow rate (cfm)
   - D = conduit inside diameter (inches)

   Although this method of determining the minimum pipe size on the basis of air velocity is easy, you also must consider that the compressed air volume is expressed in cubic feet per minute of free air, which is the air volume at ambient atmospheric conditions, not the compressed volume.

To adjust the inlet air volumetric flow rate to actual pipeline conditions, you'll need to divide the volume of free air by the compression ratio (CR) using the following equation:

CR = (P + Pa) / Pa --------------------- (Eq. 2)

Where

- P = line pressure (psig)
- Pa = average atmospheric pressure at your elevation (psi)

Note that at higher elevations, the average atmospheric pressure drops and the compression ratio rises.

For example, at a 7,000-ft. elevation, has an average atmospheric pressure of about 11 psi. At 100 psig, the compression ratio is equal to 10 (i.e. 111/11).

To determine the pipeline velocity at conditions, merely divide the velocity given in Equation 1 by the compression ratio given in Equation 2. After selecting the minimum pipe size on the basis of velocity, check any long runs for excessive pressure drop using an appropriate drop chart. For example, a velocity of 25 fps in black iron pipe represents about 0.25 psi loss per 100 ft. of run. Although this is a little above the recommended minimum of 20 fps and, depending on the layout, would probably be acceptable from a turbulence standpoint, a high total frictional loss might dictate using a larger pipe.

6. If possible it is good practice to loop the distribution piping in order to allow for air to travel in multiple directions. Where possible the piping system should be arranged as a closed loop or "ring main" to allow for more uniform air distribution to consumption points and to equalize pressure in the piping. Separate services requiring high air consumption and at long distances from the compressor unit should be supplied by separate main airlines.

7. Pipes should be installed parallel with the lines of the building, with main and branch headers sloping down toward a dead end. Branch headers from compressed air mains should be taken off at the top to avoid picking up moisture. Traps should be installed in airlines at all low points and dead ends to remove condensed moisture.

8. Replace tee connections with directional angle entry connections. Use larger size couplings: at the same flow, a 3/8 inch quick coupler has one-sixth the pressure differential of a 1/4 inch connector.

9. Consider choosing a piping material with a lower coefficient of friction such as copper or extruded aluminum for lower pressure loss.

10. Specify pressure regulators, lubricators, hoses, and connections with the lowest pressure differential and the best performance characteristics. Size components for the actual flow rates, and not the average flow rates.

Filtration

Filtration is an essential part of the conditioning in a compressed air system. If not protected from water, particles and degraded compressor oils, machines will quickly breakdown. To keep pressure drop as small as possible:

Look for the right size filter unit: As with pipe work, fitting a smaller filter is a false economy, as it will give higher initial pressure drop and also block more quickly because the surface area of the element is smaller. About 1% in higher energy costs results from every 2 psi in filter differential. If a given filter capacity is doubled the pressure loss across it will reduce by a factor of 4, for a 75% savings.

Provide the right level of filtration: A very fine filter will have a greater resistance to flow than a coarse filter. Most air tools for example will only require filtration to around 40 micron. It makes sense therefore not to use a 5 micron or even a 0.01 micron filter in this application. Where applications needing higher grade filtration exist, place the higher grade filters as close to the application as possible. Do not filter the whole of the air line or branch line to this standard.

Replace filters periodically - use pressure drop indicators: Blocked filters will increase the pressure drop and reduce the flow rate. Filter differential should be carefully monitored and filter elements replaced in accordance with manufacturers' specifications or when pressure differential causes excessive energy consumption. Accurate pressure differential gauges should be used to monitor pressure differential.

Use Cooler Inlet Air

Cool air intake leads to a more efficient compression. An increase in delivery of approximately 1 percent is gained for every 5°F reduction of intake temperature. The colder the incoming air the more the air that can be packed in for each revolution of the air compressor.

Table below shows the effect of inlet or initial temperature on air compressor delivery.

Effect of Intake Temperature on Air Compressor Delivery

Deg F	Relative Air Delivery
-10	1.155
0	1.130
10	1.104
20	1.083
30	1.061
40	1.040
50	1.020
60	1.000
70	0.980
80	0.961
90	0.944
100	0.928

Deg F	Relative Air Delivery
110	0.912
120	0.896
130	0.880

The table indicates that with respect to 60°F intake temperature any lower average air temperature means higher delivery rate of the compressor or more cfm / kW. Higher temperature than 60°F reduces the delivery capacity of the compressor by the factor shown corresponding to yellow band.

Location

Compressors shall be located in clean, well lighted, and ventilated areas of sufficient size to permit easy access for cleaning, inspection, and any necessary dismantling. Adequate aisle space is needed between items of equipment for normal maintenance as well as for equipment removal and replacement.

Where practicable, an outside air intake should be located on the coolest side of the building at least 6 feet above the ground or roof. For reciprocating units, the intake should be located at least 3 feet from any wall to minimize the pulsating effect on the structure and an intake filter silencer or an intake pulsation damper should be provided. A compressor intake must not be located in an enclosed courtyard.

Detect and Repair Leaks

One of the most fundamental ways to improve compressor system efficiency is by reducing leakage. Even small air leaks will reduce the performance of your air system and dramatically drive up your operating costs. For example, a 1/16" leak at 100 psig will lose approximately 70,000 CFM of air in a single week! A plant with several small leaks could add up to thousands of dollars lost in electrical cost alone over a year's time.

While every effort should be made to keep a compressed air system leak-tight, all systems will have some leakage. An efficient system will only have less than 10 percent going to leakage.

Estimating Total Air Leaks

A good first step in addressing air leakage in a plant is to do a low load test during a non-production time. This might be fairly easy if there is an existing accurate flow meter already installed in the system or if the air compressors have capacity gauges. If not, there are two straightforward ways to do this, but both methods must be done while production is shut down.

<u>Compressors having load/unload controls</u>

If the plant compressors operate in load/unload mode, allow the compressor to bring the system up to the pressure setpoint. Then allow the compressed air system to run through several cycles (more cycles will give you greater accuracy) as the pressure drops due to leakage and the compressor kicks on or loads up to bring pressure back to the setpoint. On each cycle, record the amount of time that the compressor is on – load (running time). A leak estimate can be made by the ratio of the on-load time to the total time of the test.

$$\text{Leakage (\%)} = \frac{T \times 100}{(T + t)}$$

Where:

- T = on-load time (minutes)
- t = off-load time (minutes)

For example if a 100 HP compressor rated at 400 cfm is loaded for 2 minutes and unloaded for 3 minutes, the leak load can be estimated by taking the loaded time and dividing the total loaded plus unloaded time, or for this example 2/5 = 0.4. This indicates the compressor is loaded 40% of the time. The leak load would then be 40% of 400 cfm or 160 cfm. If another compressor was loaded during this time its capacity would be added to this calculated value. Generally the output capacity of any compressor operating around 100 psi would be about 4 times the compressor nameplate horsepower rating.

<u>Compressors with different capacity control</u>

For systems that have other types of capacity control, leakage can be estimated by noting the time it takes for system pressure to drop from its setpoint to one-half of setpoint pressure with the compressor shut off and no production activity. The leakage rate (L) in cfm is then determined by

$$\text{Leakage (CFM free air)} = \frac{V \times (P_1 - P_2) \times 14.7 \times 1.25}{t}$$

Where:

- V = the system volume in cubic feet
- P_1 = the operating pressure in psig
- P_2 = the pressure after time t (in minutes) and should be a point equals to about one-half the operating pressure P1
- t = time in minutes, it takes for the system to drop to one-half the operating pressure P1
- The 1.25 multiplier corrects leakage to normal operating pressure, allowing for reduced leakage with falling pressure.

By comparing this leakage rate to the total volume of compressed air delivered, you can estimate the fraction of compressed air costs that are wasted by leaks. The air lost is proportional to the size of the orifice and a function of the air compressor supply pressure. The following table illustrates the amount of air lost through different orifice sizes.

Leakage rates (cfm) for different supply pressures and approximately equivalent orifice sizes

Pressure (psig)	**Orifice Diameter (inches)**					
	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

Source - http://www.oit.doe.gov/bestpractices/compressed_air

Estimating Cost of Leakage - Example

An energy audit on a factory reveals compressed air leakage through a hole of 1/8" at 100 psig. Estimate the approximate cost of leakage in dollars, assuming 3000 hours operation and electricity rate of $0.05 per kWh. The supplier data states the compressor output of 5 CFM per bhp and motor efficiency of 0.90.

Solution

Using the table above, the leakage amounts to about 25 cfm for 1/8" hole and 100 psig.

The annual power consumption for a compressor of given "cfm rating" can be found out from equation:

$$\text{kWh/yr} = \text{cfm} \times \frac{1 \text{ bhp}}{5 \text{ cfm}} \times \frac{0.746}{\text{bhp}} \times \frac{1}{\text{motor efficiency}} \times \frac{\text{hrs of operation}}{\text{year}}$$

Or

Power consumption = 25 x (1/5) x (0.746 / 0.90) x 3000 = 12433 kWh / yr

The annual cost:

Cost = kWh/yr x electricity cost

Cost = 12433 x 0.05 = $ 621 / yr

How to Track Down Air Leaks

The next step obviously is to find and eliminate the leaks. Where to look for leaks?

Experience has shown that air leaks occur most often at joints and connections. The most common problem areas are couplings, hoses, tubes, fittings, pipe joints, quick disconnects, FRLs (filter, regulator, and lubricator), condensate traps, valves, flanges, packing's, instrumentations, tools, thread seal-ants, and point-of-use devices.

There are two common methods that can be used for the detection of leaks. The more sophisticated technique utilizes an ultrasonic acoustic detector. These devices employ directional microphones, and amplifiers, to locate high frequency sounds associated with air leaks. The operator is directed to the leak location with either a visual display or thru earphones. The ultrasonic acoustic detector is fast, accurate and able to detect very small leaks, but relatively expensive to justify on small compressed air systems.

The second simpler method is to apply a soapy water solution to suspected leak locations with a brush. Then observe formation of air bubbles to pinpoint leaks. Although this method is cheap and reliable, it can be time consuming when looking for generalized leaks in a system.

The best time to find air leaks is when the plant is not operating, usually at night or on weekends. Walk the length or perimeter of the compressed air distribution system. Stop every so often and listen for air leaks. Look for damaged fittings or cracked hoses. Write down and sketch the location of the air leaks. Use tags to mark the location of air leaks for repairs. Repeat the process periodically as part of your maintenance routine.

Once you've found a leak, eliminating it is often just a matter of tightening the connection, but sometimes it will be necessary to open a joint, clean the threads, and apply proper thread sealant. In some cases you may find that you need to remove and replace faulty equipment. There are two basic types of leak repair programs, the "leak tag" and the "seek and repair" program. Seek and repair program is the simplest. As it states, you simply find the leak and repair it immediately. With the leak tag program, the leak is identified with a tag and logged for repair at a later time. This is often a two-part tag; one part stays on the leak and the other part is turned into the maintenance department, identifying the location, size and description of the leak to be repaired.

Caution: Always use appropriate vision and hearing protective equipment, and follow proper safety procedures when detecting air leaks or when working at elevated heights.

Once you've completed your leak hunt and eliminated as many leaks as possible, reevaluate the leakage rate to determine the impact you've had on the system and to estimate the resulting savings. Also, be sure to re-measure the system pressure during normal plant operation—you may find that you are now able to further reduce the compressor discharge setpoint and gain additional savings.

Some other recommended elements:

In addition to being a source of wasted energy, leaks can also contribute to other operating losses. There is strong cause and effect relationship between the number and magnitude of air leaks with the overall compressed air system pressure. For example, lower air pressure can affect air tools and equipment by reducing the mechanical output and decreasing the resulting productivity of the process.

1. Don't generate at a higher pressure than necessary - the higher the pressure, the more air that will escape through a given-size hole.

2. Don't keep your whole system pressurized during non-productive hours just because a few items of machinery require a constant supply of compressed air.

3. Do isolate parts of the system that require air at different times. Isolation valves can be operated manually or automatically using simple control devices like time switches or interlocks, or they can be controlled using your centralized energy management system, if you have one.

4. Inspect your compressed air system regularly. These inspections are an ideal opportunity to find and repair leaks.

5. Welded joints should be used instead of screwed joints as far as possible.

6. Install ball valves at the user ends, to facilitate easy opening & closing of valves.

7. Initiate a system to replace the flexible rubber hoses, joints, packings, etc., in regular intervals (Say once in 3 months).

8. Isolate non-operating equipment with a valve in the distribution system. The solenoid valve helps in cutting the compressed air supply to the individual shop when there is no activity. This minimizes the leakage loss and pressure drop to a considerable level, as most of the work shops do not operate continuously. Hence, it is recommended to install individual shop wise solenoid control valves for the compressed air line at design itself, so as to minimize the compressed air consumption during non-active periods.

Heat Recovery

For both screw and reciprocating compressors, approximately 60% to 90% of the energy of compression is available as heat, and only the remaining 10% to 40% is contained in the compressed air. This waste heat may be used to offset space heating requirements in the facility or to supply heat to a process. The heat energy recovered from the compressor can be used for space heating during the heating season. The amount of heat energy that can be recovered is dependent on the size of the compressor and the use factor. For this measure to be economically viable, the warm air should not have to be sent very far; that is the compressor should be located near the heat that is to be used.

Heating Air

Packaged rotary screw compressors are ideal candidates for heat recovery for space heating. Generally, ambient air is heated by passing it across the compressor's after-cooler and lubricant cooler. As packaged compressors are enclosed in cabinets, and generally come equipped with heat exchangers and fans, only ducting and HVAC fans need to be installed to extract heat. The ducting can include a vent that is controlled by a thermostat. The vent could direct heated air to the outside during warmer parts of the year.

It is not uncommon to be able to heat air to 15 to 25°C above the cooling air inlet temperature with 80-90 percent heat recovery efficiency. It is important to realize in using this heat that any heat recovery ventilation duct must not restrict the compressor cooling air flow. Booster fans are usually required if extensive ductwork is installed.

Heating Water

With an appropriate heat exchanger, waste heat can be extracted from the lubricant coolers in packaged water cooled, reciprocating or rotary screw compressors. Some manufacturers offer this as optional equipment. This can be used to produce hot water for use in central heating or boiler systems, industrial cleaning processes, plating operations, heat pumps, laundries, or any other application where hot water is required. Heat exchangers also offer an opportunity to produce both hot air and hot water, and allow the operator some ability to vary the hot air/hot water ratio. As many water cooled compressors are large (>100 HP), heat recovery for space heating can be an attractive opportunity.

Implement More Efficient Compressor Control

Proper control and monitoring aligns air supply with demand. The correct control system must be able to handle a compressed air system that is almost always dynamic. If your production process or operating schedule changes, verify your baseline numbers again to ensure the change hasn't degraded your system dynamic.

With regard to compressor control the following points should be considered:

1. Control of an individual compressor requires consideration of demand variation and control of air users to minimize their effect on the system.

2. Operate a minimum number of compressors necessary to base load (operate at full capacity), and use only one trim compressor to track the overall varying load. If you have multiple compressors of the same type, use sequencing controls to run all but one at full capacity. These sequencers not only control trim compressor turndown, but also will start and stop compressors according to system demand.

3. For systems with multiple compressor types, it may be beneficial to separate the control for each type. Sophisticated sequencing controllers (cascaded pressure bands, network or system master controls) now available can control more than one compressor type. When using these control schemes, don't ignore compressor type. *For example, rotary compressors with modulating, or load/unload, capacity control should be run fully loaded; variable-speed rotary compressors should be used only for trim; and centrifugal units have relatively efficient but limited, reduced capacity modulation.*

4. Remember to consider the element of time when designing or tuning a compressor control system. Compressors require time to start up and be brought up to speed. This may require extra storage receiver capacity.

5. Cascaded pressure bands single pressure bands need to be adjusted from time to time.

6. The ‘‘trim compressor’’ should be the one most capable of running efficiently at partial loads.

Install Storage Capacity

Receivers can help compressed air systems operate more efficiently and can help stabilize system pressures.

The following points should also be considered:

1. Where practical, locate the receivers as close to the air compressors as possible.

2. For most facilities with load/unload rotary screw compressors, install air receiver capacity of 10 US gallons per cfm of compressor capacity.

3. When receivers are exposed to subfreezing temperatures, precautions need to be taken to prevent freezing in the condensate drains. In some cases receivers rated for lower temperatures are required.

4. Select a slightly larger receiver than what may be currently required. This will generally result in improvements to stabilizing system pressure and also respond to intermittent demands.

5. In cases where the air needs to be dried, it is sometimes beneficial to install two receivers -- one before and one after the dryer.

Optimize Air Dryers

Air dryers can consume significant compressed air or electrical power and often have limited turndown capabilities. It is possible the existing air dryer could be upgraded or replaced with good savings results. Consider the following points with regard to dryers:

1. For new purchases of refrigerated air dryers always consider the energy savings cycling style.

2. Avoid drying the air to a dew point level that is lower than what is needed for a specific application.

3. Use energy saving dew point controllers for all types of regenerative desiccant dryers.

Reduce System Drainage

Condensate drains are a common point of compressed air loss. Consider airless drains as replacements for timer drains or manual drains that are partially cracked open. The following points should be considered:

1. Where possible, procure condensate drains having a gauge glass. This will provide a visual indicator if the trap malfunctions.

2. Regularly test automatic drain traps for proper operation.
3. Piping should be sloped slightly downwards and away from the compressors.
4. Locate drains at the bottom of main headers in order to allow condensate to collect and flow by gravity.
5. Avoid using open manual drain valves.

Substitution of compressed air

Compressed air is highly energy intensive and costly. So the users have to think of the possibilities of replacing compressed air, with an equivalent source at design stage. The possible areas for substituting compressed air are:

1. Cooling and cleaning – compressed air for cooling/cleaning is a common practice in manufacturing industry. Compressed air for cooling purposes can be replaced with a blower cooling. This not only saves power, but also effects a good cooling. Wherever separate system for cleaning cannot be justified, transvector nozzles (work on the venture principle) can be installed for the compressed air cleaning hoses, to minimize the compressed air consumption.
2. Vacuum generation with a venture, eductor or ejector is wasteful. Vacuum pump is better choice.
3. Sparging – aerating, agitating plating tanks, oxygenating, or percolating liquid with compressed are. Low pressure blowers or fans should be used instead.
4. Agitation - Normally compressed air is utilized for agitation purposes in ETP tanks, Pretreatment tanks, etc., For agitation purposes, the quantity of air required is important than the pressure (required only to push through the water column through - a max of 10m height). This can be replaced with a Roots Blower.

5. Aspirating – using compressed air to induce the flow or another gas. Low pressure blowers or fans should be used instead.

6. Atomizing – Using compressed air to deliver a liquid to a process as an aerosol. Low pressure blowers or fans should be used instead.

7. Dilute pneumatic conveying – using compressed air to transport fine powders in a diluted format. Low pressure blower or fans should be used instead.

8. Install electrical tools as much as possible, instead of pneumatic tools. Previously, the electrical tools had some design problems like overweight, overheating, frequent armature failures, etc. Now these have been taken care of and a new generation of high frequency electrical tools is available. The replacement of pneumatic tools with electrical tools will result in a power savings of 30%. This is mainly due to the high cost and energy intensiveness of compressed air. Moreover, pneumatic tools are highly leak-prone. This results in unnecessary wastage of compressed air.

Although upfront capital investment will be necessary to eliminate some inappropriate applications, performing them with compressed air is so inefficient that the required investment will usually be repaid quickly. Eliminating inappropriate uses will reduce compressed air consumption and may allow you to shut down one or more compressors entirely. This can save capital in the future as well—should expanded production require additional compressor capacity, you'll have it ready and waiting.

Use Root Blower

If the pressure for a particular area of application is less than 30 psig, a blower is usually more cost-effective than compressing air at 100 psig and then regulating it down to a much lower level.

Compressed air for pneumatic tools

In some applications electric motor is much more efficient than the compressed air. For example, a 1.17 rated horsepower air operated mixer uses 45 cfm at 80 pounds-per-square-inch (psi) and operates 40 hours per week. The cost of the compressed air to operate this motor over a year is $1,292. A comparably sized electric motor of Energy Policy Act (EPACT) efficiency, rated for hazardous locations, is around $350. The cost to operate the EPACT motor under the same conditions is less than $100 per year. Including installation, payback is under one year.

Use Pressure Regulators

Artificial demand is created when an end use is supplied air pressure higher than required for the application. If an application requires 50 psi but is supplied 90 psi, excess compressed air is used. Use pressure regulators at the end use to minimize artificial demand.

Optimizing Motor Loading

The motor loading of an air compressor is a direct function of the air demand placed on the compressor by the plants pneumatic equipment.

Determining if your motors are properly loaded enables you to make informed decisions about when to replace motors and which replacements to choose.

Most electric motors are designed to run at 50% to 100% of rated load. Maximum efficiency is usually near 75% of rated load. A motor’s efficiency tends to drop significantly below about 50% load. Another negative by product of lightly loaded air compressor motors, is the adverse effect on the line power factor. Power factor is the fraction of power actually delivered in relation to the power that would be delivered by the same voltage and current without the phase shift. Low power factor imply excess current in the system. The energy associated with the excess current is alternately stored in the motor windings’ magnetic field and regenerated back to the line with each AC cycle. This exchange is called reactive power. Though reactive power is theoretically not lost, the distribution system must be sized to accommodate it, which is a cost factor. To reduce these costs, capacitors are used to “correct” low power factor.

There are two methods to determine motor loading: 1) Phase current methodology and 2) Slip technique

The phase current methodology is quite straight-forward:-

1. Using the manufacturers data calculate the full load power input to the motor.

2. Using the digital ammeter determine the phase currents, and take the mean value of the three readings.

3. Calculate the dynamic power input to the motor.

4. Divide the dynamic power input by the full load power input, and multiply by 100.

Actual Motor Loading

Calculate motor loading using equation:-

$$\text{KW Input Power} = \frac{[V \times I] \times [\cos \Phi] \times [\sqrt{3}]}{1000} \text{ -- (eq. 1)}$$

Where

- V and I are the mean values of the phase voltage and current readings.

- The Cos term is the power factor, if the compressor is not metered for power factor, then use the overall cos Ø value for the plant.

Power Input at Full Rated Load

$$\text{KW Input Power At Full Rated Load} = [\text{hp}] \times \left[\frac{0.7457}{\eta\ \%} \right] \text{ ----------------- (eq. 2)}$$

Where

- η% is manufacturer's motor efficiency at full rated load

- hp is manufacturers nameplate rated horsepower

Motor Load Factor

Motor load factor can be calculated as:

$$\text{Load} = \frac{\text{kW input power}}{\text{kW at full rated load}} \times 100\% \qquad \text{------------------ (eq. 3)}$$

Example

Suppose we have a 100hp motor operating at the mean phase current of approximately 45 Amps for 30% of the time.

Use Equation 1 to calculate the load power:

$$\text{Actual Input KW} = \frac{[440 \times 45] \times [0.87] \times [1.732]}{1000} = 29.8\text{KW}$$

Use Equation 2 to calculate the full load input power:

$$\text{Full Load Input Power} = [100] \times \left[\frac{0.7457}{0.94}\right] = 79.3\text{KW}$$

Use Equation 3 to calculate Load%:

$$\text{Actual Motor Load} = \left[\frac{29.8}{79.3}\right] \times 100 = 37.6\%$$

We observe in this example that the motor is operating approximately at 37% loading.

Ideally, the air demand should be such that the compressor motor is always operating at its most efficient, however, this is rarely the case, and you need to determine when, and how for long the compressor motor is operating at less than 50% load.

Method #2 Rotational or Slip Method

The Rotational or Slip Method of determining, air compressor motor loading, is a relatively old method of doing things. The methodology is beautiful, no electrical or physical connection to, or contact with the air compressor under test is required. The only test instrument required is a stroboscope. However, this method is not as accurate as the previously describe method, using measured values of phase voltage and current.

The technique is interesting and is described below.

All electric motors rely upon "slip" to develop power, or useful torque. The amount of rotational slip, and power developed, being proportional to the motor loading. When a motor is running with an "open rotor" i.e. no mechanical load connected, then the "open rotor" synchronous speed will be a function of the number of poles the motor has.

Number Of Poles	**Synchronous Speed (rpm)**
2	3,600
4	1,800
6	1,200
8	900
10	720
12	600

The "pure slip" method as described by equation 1.

$$\text{Load \%} = \frac{[\text{Slip}]}{[S1-S2]} \times 100 \quad \text{------------------------------} \quad (\text{eq. } 1)$$

Where

- Slip = Measured speed in rpm
- S1 = Synchronous speed in rpm
- S2 = Nameplate full load speed rpm

Example

Suppose we have a 100hp motor with a synchronous speed of 1800rpm, a nameplate full load rotor speed of 1750rpm, and a measured rotor speed of 1760 rpm. What will be the motor load in hp?

Solution

$$\frac{[1800-1760]}{[1800-1750]}\times 100 = 80\% \text{ of } 100\text{hp} = 80\text{hp}$$

With = +/- 10% tolerance.

Purchase a More Efficient Compressor

A good energy management strategy may be to purchase a new more efficient compressor as a replacement for an older existing unit. Often the existing unit can be retired to standby duty, providing backup capacity for increased system reliability. Consider the following when purchasing a compressor:

1. Purchase the most energy efficient compressors, including ones equipped with premium efficiency motors.

2. In situations involving multiple compressors, operate the base load units at maximum capacity rather than partially loaded.

3. Consider purchasing and operating at least one Variable Speed Drive compressor to supply variations in flow above the base load.

4. Purchase of a two stage compressor might provide better system efficiency if used as a base compressor.

Improve Routing Maintenance

Good maintenance will ensure optimum efficiency of the compressor system. Plant operators should follow the maintenance schedule as set out in the operator’s manual, and keep maintenance records. This may involve regular visual inspection of thermometers and gauges, e.g. high discharge temperature may be a result of faults on coolers, valves, pistons and rotors.

Some maintenance tips (aimed at rotary screw compressor systems) for you to consider are listed below:

Frequency or Running Hours	**Action**
Daily	After normal start procedure, check control panel and gauges. Using a log book, record pressures, cooling water temperature. Check for abnormalities compared to previous days' operations.
Weekly	Inspect for Air Leaks (fittings, cracked hoses) Inspect and replace filters if necessary Check and adjust air regulators Check and adjust system pressures Check and adjust refrigerated dryer set points
Every 3,000 hours	Check and replace filter element Check/change sump-breather filter element Check/clean condensate drain valves Inspect the condition of shaft couplings and fasteners Apply the specified quantity and type of lubricating

Frequency or Running Hours	Action
	grease for motor bearings
Every 15,000 hours	Test all safety devices Inspect and clean heat exchangers Check and clean blowdown valves, check valves, interstage pipe works, isolation mounts Inspect and clean lubricant sump check valves and strainers

An easy method to check the condition of a compressor is to regularly record the time the compressor needs to run to build up pressure in the receiver, or by observing the position of the controller at a set load on a rotary compressor. This method will give a good indication on the condition of the compressor. A worn or faulty compressor will still provide compressed air, but it may require a higher controller position, or longer running periods.

From an energy efficiency perspective, in many cases, it is wise to maintain equipment more frequently than the recommended intervals. This is especially true for managing air leaks, high pressures, moisture and controls.

Computer control and monitoring systems

Sophisticated computer control and monitoring systems are also becoming more common. These computer control systems coupled with active monitoring of system pressure, temperature, and air demand can trend the operation of a compressed air system and stage multiple compressor types to best accommodate the load or demand.

Computer controls can also save maintenance costs by providing alarms for filter clogging, pressure, temperature, and demand spikes all from a central management console.

Again, caution should be used before implementing such a system. When purchasing a sophisticated control or maintenance system the user should be keenly aware of the operation of their system before purchase so as not to be caught off guard with unrealistic expectations of savings.

Replacing “manual controls” with automatic controls serving the same function will not save energy. A sick compressed air system with leakage, and poor air quality will not be cured by such a system. Most of the time these systems are best suited for complex compressed air systems with numerous compressors and fluctuating demand.

Air compressor systems will be protected against high temperature, high pressure, low oil pressure, and in the case of centrifugal compressors, excessive vibration. Protective controls will include a fault indicator and a manual reset device.

Annexure – 1

Checklist for Energy Efficiency in Compressed Air System

1. Ensure air intake to compressor is not warm and humid by locating compressors in well ventilated area or by drawing cold air from outside. Every 4°C rise in air inlet temperature will increase power consumption by 1 percent.

2. Clean air-inlet filters regularly. Compressor efficiency will be reduced by 2 percent for every 250 mm WC pressure drop across the filter.

3. Keep compressor valves in good condition by removing and inspecting once every six months. Worn-out valves can reduce compressor efficiency by as much as 50 percent.

4. Install manometers across the filter and monitor the pressure drop as a guide to replacement of element.

5. Minimize low-load compressor operation; if air demand is less than 50 percent of compressor capacity, consider change over to a smaller compressor or reduce compressor speed appropriately (by reducing motor pulley size) in case of belt driven compressors.

6. Consider the use of regenerative air dryers, which uses the heat of compressed air to remove moisture.

7. Fouled inter-coolers reduce compressor efficiency and cause more water condensation in air receivers and distribution lines resulting in increased corrosion. Periodic cleaning of intercoolers must be ensured.

8. Compressor free air delivery test (FAD) must be done periodically to check the present operating capacity against its design capacity and corrective steps must be taken if required.

9. If more than one compressor is feeding to a common header, compressors must be operated in such a way that only one small compressor should handle the load variations whereas other compressors will operate at full load.

10. The possibility of heat recovery from hot compressed air to generate hot air or water for process application must be economically analyzed in case of large compressors.

11. Consideration should be given to two-stage or multistage compressor as it consumes less power for the same air output than a single stage compressor.

12. If pressure requirements for processes are widely different (e.g. 3 bar to 7 bar), it is advisable to have two separate compressed air systems.

13. Reduce compressor delivery pressure, wherever possible, to save energy. Provide extra air receivers at points of high cyclic-air demand which permits operation without extra compressor capacity.

14. Keep the minimum possible range between load and unload pressure settings.

15. Automatic timer controlled drain traps wastes compressed air every time the valve opens. So frequency of drainage should be optimized.

16. Check air compressor logs regularly for abnormal readings, especially motor current cooling water flow and temperature, inter-stage and discharge pressures and temperatures and compressor load-cycle.

17. Install equipment interlocked solenoid cut-off valves in the air system so that air supply to a machine can be switched off when not in use.

18. Compressed air piping layout should be made preferably as a ring main to provide desired pressures for all users.

19. Misuse of compressed air such as for body cleaning, agitation, general floor cleaning, and other similar applications must be discouraged in order to save compressed air and energy.

20. Pneumatic equipment should not be operated above the recommended operating pressure as this not only wastes energy but can also lead to excessive wear of equipment's components which leads to further energy wastage.

21. Pneumatic transport can be replaced by mechanical system as the former consumed about 8 times more energy. Highest possibility of energy savings is by reducing compressed air use.

22. Pneumatic tools such as drill and grinders consume about 20 times more energy than motor driven tools. Wherever possible, they should be replaced with electrically operated tools.

23. Where possible welding is a good practice and should be preferred over threaded connections.

24. On account of high pressure drop, ball or plug valves are preferable over globe valves in compressed air lines.

Annexure – 2

Engineering Equations

A good process and systems engineer understands the engineering formulae and technical relationship between compressor motor power-draw and process variables.

Reciprocating/ Rotary Screw Compressor Formulas

Basic Law

One of the most significant gas laws - Marriott and Gay-Lussac law states:
P x V = a x T

Where:

- P : absolute pressure (Pa)
- V : volume (ft^3, m^3)
- T: absolute temperature (K)
- a : constant

This relation is used within the compressor: constant air volume is pumped from the compressor chamber, and the volume decreases. This decrease causes an increase in both the pressure and the temperature of the air

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Air flow Calculation

Flow is equivalent to the quantity of compressed air conveyed in a given section per unit of time.

Q = A1 x V1 = A2 x V2

Where

- Q: flow (cfm)

- A: flow section (ft²)
- V: speed (ft/min)

The international system of flow is cubic meters / second (m^3/s), but l/s, m^3/h or cfm is more common in industry.

Flow at Standard temperature and pressure (STP):

$$Q_s = \frac{1545 \times w \times T_S}{60 \times 144 \times MW \times P_S}$$

Where

- Qs - Volumetric flow rate at Standard Condition.
- w -: Mass flow rate, lb/hr.
- MW - Molecular weight.
- Ts - Absolute Temperature at Standard Condition, °R.
- Ps - Pressure at Standard Condition, psia.

There are three standards available to enter Flow@STP.

1. API Standard: 14.7 psia, 60 °F, 0% relative humidity
2. ASME Standard: 14.7 psia, 68 °F, 36% relative humidity
3. CAGI Standard: 14.7 psia, 60 °F, 36% relative humidity

Compressor Capacity (ICFM)

$$Q_1 = (1545 \times w \times Z_1 \times T_1) / (60 \times 144 \times MW \times P_1)$$

Where

- Q_1 - Compressor capacity at inlet T and P, cubic ft/min (ICFM)
- Z_1 - Compressibility factor of gas at inlet
- T_1 - Inlet temperature, ^oR
- P_1 - Suction pressure, psia

Inlet Gas Density:

$$\rho_1 = \frac{MW \times 529.7 \times P_1}{386.7 \times T_1 \times 14.7}$$

Where

- p_1 - Inlet gas density, lb/cubic ft

Outlet Gas Density:

$$\rho_2 = \frac{MW \times 529.7 \times P_2}{386.7 \times T_2 \times 14.7}$$

Where

- p_2 - Outlet gas density, lb/cubic ft
- P_2 - Discharge Pressure, psia.
- T2 - Discharge Temperature, °R

Adiabatic Head:

For **BHP, FREE AIR** and **FLOW** calculation:

$$H_{ad} = Z_{av} \times R \times T_1 \times (r^{\frac{k-1}{k}} - 1) / (\frac{k-1}{k})$$

Where

- Had - Adiabatic head, ft-lb/lb
- Zav - Average compressibility factor
- R - Gas constant, 1545/MW
- T_1 - Inlet air temperature, °R
- r - Compression ratio (P2/P1), unit less
- k - Adiabatic exponent

Discharge Temperature:

$$T_2 = T_1 \times r^{\frac{k-1}{k}}$$

Gas Horsepower:

<u>If mass flow rate is available:</u>

$$GHP = \frac{w \times H_p}{60 \times 33000 \times E_p}$$

Where

- GHP - Gas horsepower, hp

- Hp - Adiabatic head, ft-lb/lb
- Ep - Adiabatic efficiency

If capacity is available:

$$GHP = \frac{Q_1 \times P_1 \times \frac{Z_{av}}{Z_1} \times \frac{r^{\frac{n-1}{n}} - 1}{\frac{n-1}{n}}}{229 \times E_p}$$

Where

- Q_1 - Capacity (ICFM), cubic ft/min
- Zav - Average compressibility factor
- Z1 - Compressibility factor of gas at inlet

Brake Horsepower:

BHP = GHP (1 + %Mechanical Losses)

Annual electricity costs

Annual electricity costs = (compressor BHP) x (0.746 kW/HP) x (motor efficiency) x (Annual hours of operation) x (Electricity cost in $/kWh)

Note - 0.746 in the formula converts horsepower into kilowatts.

Annexure - 3

Evaluating Compressed Air Costs

Compressed air is one of the most expensive sources of energy in a plant. The overall efficiency of a typical compressed air system can be as low as 10-15%. For example, to operate a 1 hp air motor at 100 psig, approximately 7-8 hp of electrical power is supplied to the air compressor.

To calculate the cost of compressed air in your facility, use the formula shown below:

$$\text{kWh/yr} = \frac{\text{bhp} \times (0.746 \text{ kW / bhp}) \times \text{hrs of operation} \times (\%\text{ time}) \times (\%\text{ full load bhp})}{\text{Motor efficiency}}$$

Where

- bhp - Compressor shaft horsepower (frequently higher than the motor nameplate horsepower—check equipment specification)
- Percent time—percentage of time running at this operating level
- Percent full-load bhp—bhp as percentage of full-load bhp at this operating level
- Motor efficiency—motor efficiency at this operating level

The annual cost of electricity used to power a compressed air system can be found out from equation:

$$\$ \text{ per year} = \frac{\text{kWh}}{\text{yr}} \times \text{ cost of electricity in } (\$ / \text{kWh})$$

Example

A facility operates a 100 hp air compressor 4,160 hours annually. It runs fully loaded, at 94.5 percent efficiency, 85 percent of the time. It runs unloaded—at 25 percent of full load—at 90 percent efficiency, 15 percent of the time. The electric rate is $0.06 per kWh, including energy and demand costs. The cost per year to power the air compressor will be as follows.

Cost when fully loaded = 100 HP x 0.746 x 4,160 hr x $0.06/kWh x 0.85 x 1.0 / 0.945 = $16,748

Cost when unloaded = 100 hp x 0.746 x 4,160 hr x $0.06/kWh x 0.15 x 0.25 / 0.90 = $776

The total annual energy cost to operate the air compressor is $16,748 + $776 = $17,524.

Alternate Method #1

Compressed air is one of the most expensive uses of energy: depending on the type of compressor used, compressors are typically rated to deliver **four to five** SCFM per horsepower (rule of thumb).

The annual power consumption for a compressor of given "cfm rating" can be found out from equation:

$$\text{kWh/yr} = \text{cfm} \times \frac{1\ \text{bhp}}{5\ \text{cfm}} \times \frac{0.746}{\text{bhp}} \times \frac{1}{\text{motor efficiency}} \times \frac{\text{hrs of operation}}{\text{year}}$$

The annual cost of electricity:

$$\$\ \text{per year} = \frac{\text{kWh}}{\text{yr}} \times \text{cost of electricity in } (\$ / \text{kWh})$$

<u>Method # 2</u>

The most accurate method of determining instantaneous compressor power is to directly measure the input power using digital ammeter, voltmeter and power factor with clamp meter.

$$\text{kW Input Power} = \frac{[V \times I] \times [\cos \Phi] \times [\sqrt{3}]}{1000}$$

Where

- V and I are the mean values of the phase voltage and current readings.

- The cos Ø term is the power factor, if the compressor is not metered for power factor, then use the overall cos Ø value for the plant.

This method provides instantaneous power consumption. The calculation can be repeated at every 15 minutes to create log for about a weak. The power profile than can be worked out. Alternatively a power input recorder can provide the average power consumption.

The annual power consumption:

$$\text{kWh/yr} = \text{kW Input Power} \times \frac{\text{hrs of operation}}{\text{year}}$$

The annual cost of electricity:

$$\text{\$ per year} = \frac{\text{kWh}}{\text{yr}} \times \text{cost of electricity in (\$ / kWh)}$$

Example

A power profile of compressor use shows an average current of 230 amperes, voltage of 460 volts and power factor of 0.85. What will be the annual power consumption, if the power cost is $0.05 per kWh and the annual operation hours are 4160?

The power input can be calculated as:

$$\frac{(230) \times (460) \times (1.732) \times (0.85) \times 4{,}160 \times \$0.05}{1{,}000}$$

= $ 32,398 per year

Remember, over the life of a compressor, energy costs will be five to 10 times the compressor's purchase cost. Energy savings can rapidly recover the extra capital required to purchase an energy-efficient air compressor motor.