# Industrial Systems Optimization –

# Experts Training Manual

## Compressed Air Systems Optimization

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# LEARNING OBJECTIVES

## General learning objectives for Compressed Air System Management workshop

The goal of the workshop is to provide participants with an understanding of general elements regarding compressed air systems so they will be able to:

- Identify all components of the system and understand their function.
- Identify the sources of demand and evaluate their appropriateness.
- Quantify the impact of variations in the compressed air supply on plant production.
- Establish a baseline of system performance.
- Calculate energy costs associated with various components of the system.
- Calculate lifecycle costs of components.
- Take measurements to document the dynamics of the system.
- Recommend changes that will improve system performance and reduce operating costs.

In order to accomplish these tasks, the participants will learn:

- How to calculate energy costs.
- How to estimate the cost impact of poor air quality on production.
- Definitions and gas laws associated with compressed air systems.
- General operating principles and types of positive displacement and dynamic compressors.
- Proper application of positive displacement and dynamic compressors.
- Compressed air quality requirements.
- Compressed air treatment equipment and its application.
- How to produce simple block diagrams.
- How to develop a pressure profile.
- How to identify the sources of demand, including; 1) system dynamics, 2) critical pressure applications, 3) critical flow applications, and 4) leak load.
- How to balance the supply side to the demand side with proper control strategies.
- How to determine proper storage requirements.
- How to collect the data required to understand the system and to make recommendation

# 1. Introduction to Compressed Air Systems


Compressed air has been used in industry for well over a century. Only in recent years are industrial facilities learning to generate and use compressed air efficiently. Compressed air was once considered to be either free or part of the ongoing operating expense of the company. With increased worldwide energy costs, these facilities are realizing that compressed air is not free and that the cost of compressed air contributes to the per-unit cost of their products. Beginning a facility program to better manage an industrial compressed air system is difficult because there is very little written information about managing the system as a whole. One can learn about compressors or dryers or filters easily enough. Learning about the dynamics of the entire system is much more difficult.

This training course will provide enough material to begin to understand how a compressed air system works.  The real learning, however, will begin in the field.

## 1.1 Compressed Air Systems

Compressed air has three primary uses.  Compressed air is used as a *power source*, *part of a process, or for control*.

For *power* applications, air is compressed to a pressure above the ambient, or atmospheric, pressure.  The air is then transported through a distribution system (piping) and provided to a point of use where some of the energy required to compress the air can be recovered to do work. This work may be to move the rod of an air cylinder or to move paint from the nozzle of a spray gun to a product or any other application of force or movement.

In *process applications*, the air is part of the process. Air may be used to aid combustion, aerate a liquid or catalyze a chemical reaction. In process applications compressed air does not produce a movement or apply a force.

*Control applications* use pneumatics in much the same way an electrical circuit might be used. The compressed air starts, stops, regulates or otherwise controls the operation of a machine or process.  In many plants, the compressed air is used for two or all three of these functions.

Often compressed air is, in fact, a utility much like water and electricity. In other applications, compressed air is an integral part of the production process. For applications impacting productivity, product quality, scrap rates, and rework cost; compressed air is a critical process variable that should be monitored and controlled.

A compressed air system is comprised of both supply side components and demand side components.

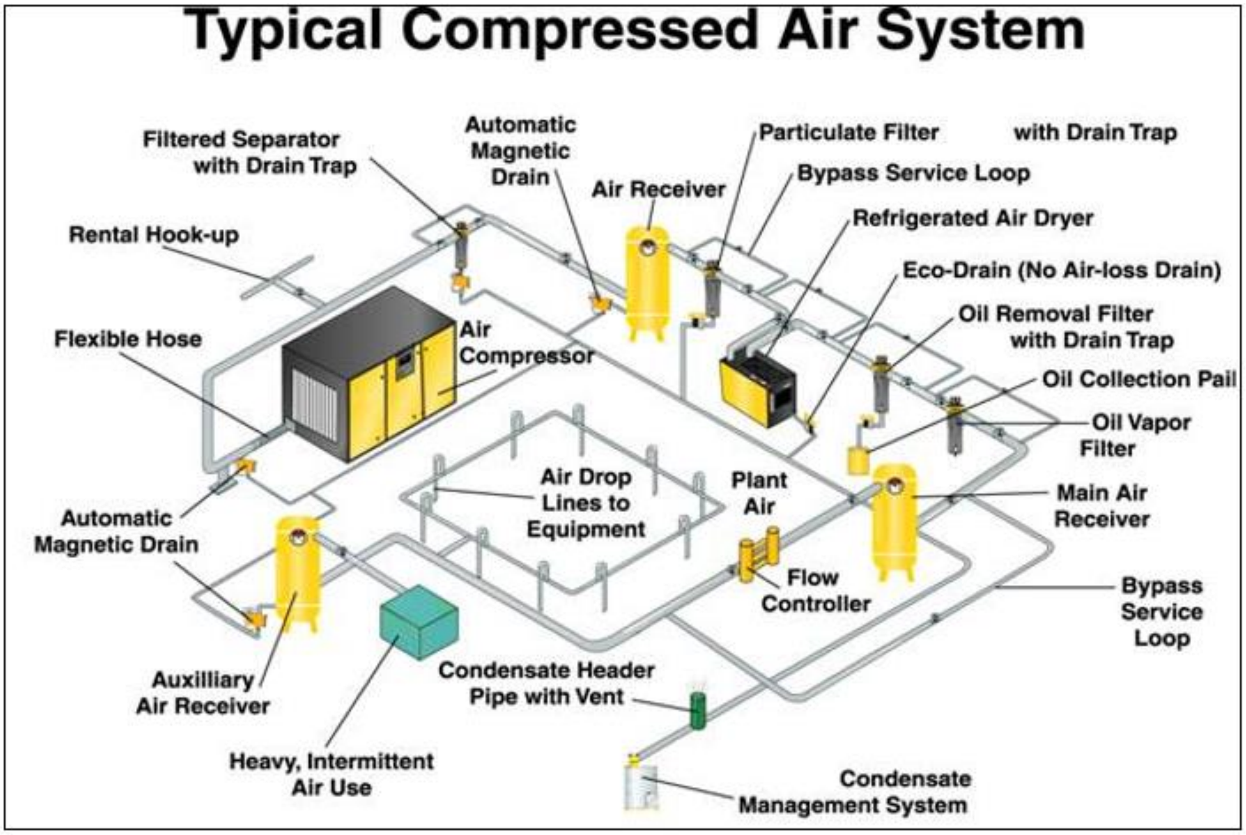



**Figure 1-1: Typical Compressed Air System**

The supply side includes all of the equipment required to generate and treat the compressed air. Typically, this will include the compressor, a compressed air receiver for storage, a dryer to remove moisture and filters to remove oil and particulates. In some applications, there may also be an pressure/flow controller that separates the supply side from the demand side.

The demand side of the system includes all compressed air consumers. Demands include productive end-use applications and various forms of compressed air waste.

The transmission system includes distribution (piping) system and other system components that move the compressed air energy from where it is generated to where it is used. The transmission system may include additional air dryers, filters, receiver tanks, control valves, etc. The transmission system may also include instruments to measure performance, e.g. flow, pressure, temperature, dew point; along with data collection and performance reporting systems.

A properly designed compressed air system will have:

- The supply side optimized to provide that air with the lowest possible kilowatt input.
- The demand side optimized to use the least amount of air at the lowest possible pressure.

- The distribution system will transmit pneumatic energy from the supply side generation to end use demands with minimal energy (pressure) loss.

## 1.2 Managing Compressed Air Systems

To optimize performance and energy efficiency it is necessary to:

- understand how the compressed air system supports productive air demands,
- eliminate waste,
- apply compressed air energy storage, and
- optimize compressor control.

To accomplish these objectives requires knowledge about the compressed air system, the production process, and how the supply and demand sides of the system interact. Unfortunately, in many instances little is known about the system because performance has not been measured and is not well defined. System management is guided by what is believed to have worked properly in the past and is often based on anecdotal information and traditional methods of operation.

Tradition is something that is done, even though the original reason might be forgotten. Traditions in compressed air system management must be questioned and examined.

Examples of traditional compressed air system management include:

- Plant production is the number one priority.
- The plant compressed air supply must always be maintained.
- Over supply of compressed air is acceptable, under supply is not acceptable.
- Minimum pressure must be maintained. Higher pressure is acceptable.

A newer understanding of compressed air system management would set these rules:

- Plant productivity is the number one priority.
- The plant air demand must always be supplied.
- The compressed air supply must be in balance with demand. Both over supply and under supply are unacceptable.
- Compressed air pressure must be stable. Pressures higher than required are unacceptable as are pressures lower than required.

Traditionally, compressed air system demand has been supplied without concern for controlling the cost. There has been no effort to balance the air supply to the demand. If an air application requires 6.5 bar, maintaining 9 bar in the system was not uncommon. This is inefficient and wastes tremendous amounts of energy. Over-pressuring the system can fulfil the requirement to deliver power to compressed air demands; however, this tradition also wastes 30% to 40% of the energy that is applied to the system.

Not only is an overpowered air system inefficient, but also this method of management usually provides inconsistent compressed air pressure and quality. This inconsistent performance directly affects production and results in production losses, wasted time adjusting equipment, and scrap product. An overpowered air system will have significant pressure changes depending on the time of day, production volume, or processes that are running. As the overall system pressure changes every air demand is affected. Changes in system pressure will also affect filter, and dryer performance, resulting in poor or inconsistent air.

## 1.3 Systems Approach

The **Systems Approach** is a fully integrated approach to compressed air system management with focus on total system performance rather than individual component efficiency.

- Understand compressed air use as it supports critical plant production functions.
- Correct existing poor performing applications, and those that cause upset to system operation.
- Eliminate wasteful practices.
- Maintain an energy balance between the compressed air supply and productive compressed air demands.
- Optimize energy efficiency with application of compressed air energy storage and air compressor control.

## 1.4 Compressed Air Economics – Reducing Life Cycle Costs

Industrial compressed air systems represent a significant capital investment and large operating expense. The obvious operating costs are energy and maintenance cost. Somewhat less obvious is the air system's performance impact the production process and plant productivity. There are four components which contribute to the total cost of ownership of a compressed air system.

- Capital investment
- Maintenance cost
- Energy cost
- Air system performance and lost plant productivity

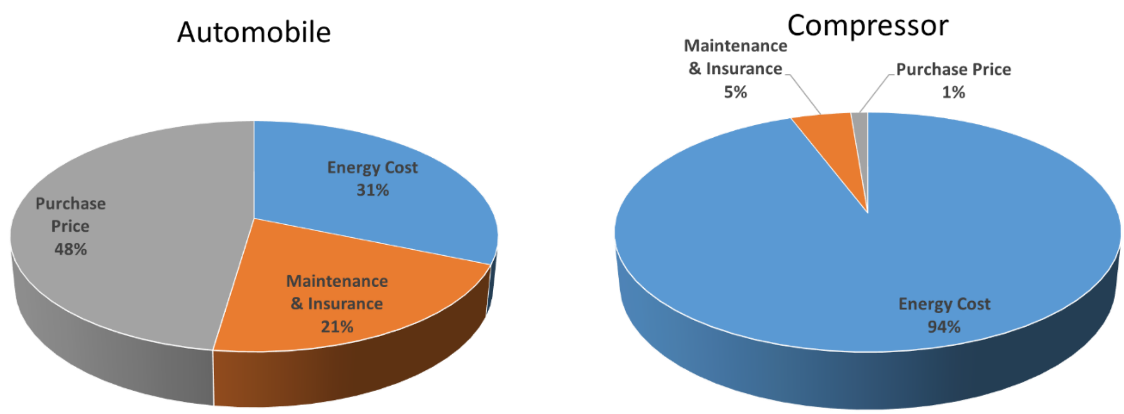



**Figure 1-2: Comparing the 5 year life cycle costs of an automobile and a compressor**

Equipment and maintenance represent only a small portion of the overall cost of owning and operating a compressed air system. Over a 10 year period, the energy required to operate the compressor is usually 75%, or more, of the annual cost of compressed air (source: Compressed Air Challenge®). If the compressed air system is unreliable and performs poorly; the impact on lost production, product quality, scrap rates, and rework costs can be the single greatest cost.

Compressed air is one of the most expensive sources of energy in a plant. A survey by the U.S. Department of Energy showed that for a typical industrial facility, approximately 10% of the electricity consumed is for generating compressed air. For example, to operate a 1 kW air motor at 7 bar, approximately 7-8 kW of electrical power is supplied to the air compressor.

For some facilities, compressed air generation may account for 30% or more of the electricity consumed. Compressed air is an on-site generated utility.

Too many decisions regarding compressed air systems are made on a first-cost basis, or with an "if it ain't broke, don't fix it" attitude. To achieve optimum compressed air system economics, compressed air system users should select equipment based on life-cycle economics, properly size components, turn off unneeded compressors, use appropriate control and storage strategies, and operate and maintain the equipment for peak performance.

## a. Compressed Air Energy Conversion

If compressed air could be used at the temperature at which the air left the discharge port and if the piping system were perfectly insulated and all of the energy that went into compressing the air were available at the point of use, the compressed air system would be perfectly efficient. Remember that isentropic compression, by definition, is a reversible process and all of the energy input should be available to perform work. Unfortunately, that is not the case.

When using an air compressor to convert electrical energy to pneumatic energy, there are efficiency losses. Two significant efficiency losses are 5% motor and drive loss; plus, the largest efficiency loss 80%; is due to heat of compression. As a result, the pneumatic energy delivered is only 15% of the air compressor's electrical energy use.

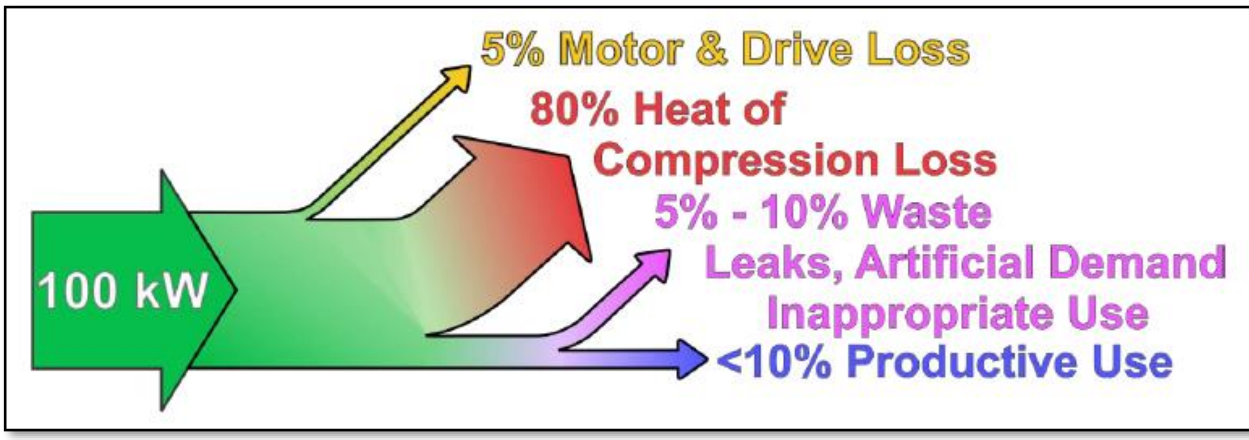



**Figure 1-3: Electrical to Pneumatic Energy Conversion**

Put another way, a pneumatic motor that can deliver 1 kW of work output requires about 6.67 kW of compressor power. This is a theoretical calculation because other system losses are not taken into account.

In most systems a portion of the compressed air energy generated at the air compressor is wasted. There are various forms of compressed air waste.

- Leakage of compressed air – 20% to 30% of total system capacity
- Artificial demand – 15% to 25% increased demand from excessive system pressure
- Inappropriate use of compressed air – 15% to 25%

Overall compressed air efficiency of the system, is typically less than 10%

## b. Reducing Energy Consumption

Optimizing the energy consumption of the entire compressed air system is important because so much of the energy used to compress air cannot be recovered to perform work. This includes supply, distribution and end use (demand).

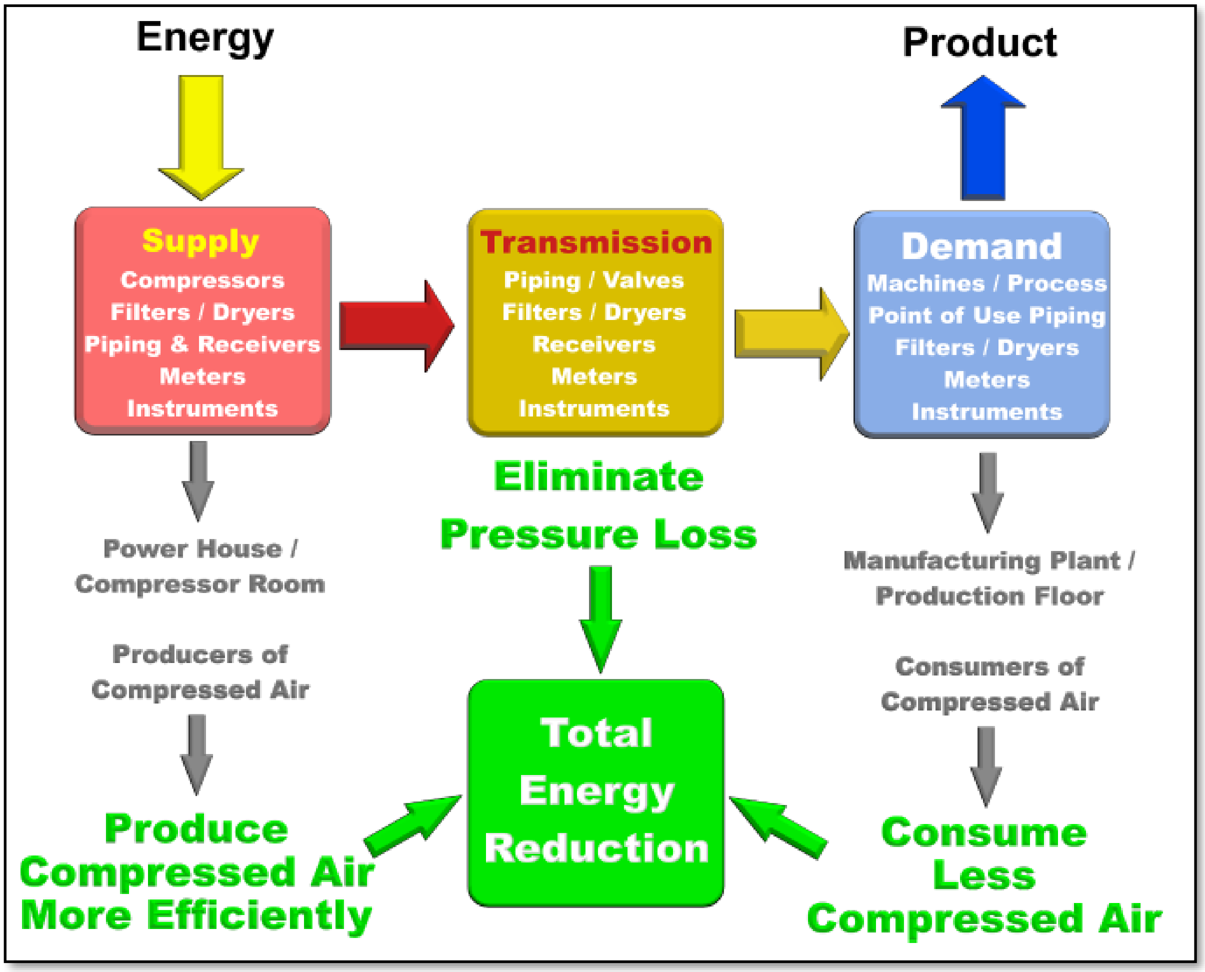



**Figure 1-4: Energy Reduction using the Systems Approach**

The systems approach directs less focus on individual component efficiency, with greater focus on total system efficiency. Three areas of opportunity for energy reduction are shown in Figure 1-4; Compress air more efficiently, reduce pressure loss, and consume less compressed air in the production process.

- The compression of air is a fundamentally inefficient process due to heat of compression. The ability to produce compressed air more efficiently is limited.
- Reducing pressure saves energy at the compressor (about 6% per bar). In addition, leaks and unregulated uses consume less air at lower pressure.

- The greatest energy reduction comes from avoiding the compression of air; by eliminating waste, and consuming less compressed air in the manufacturing process.

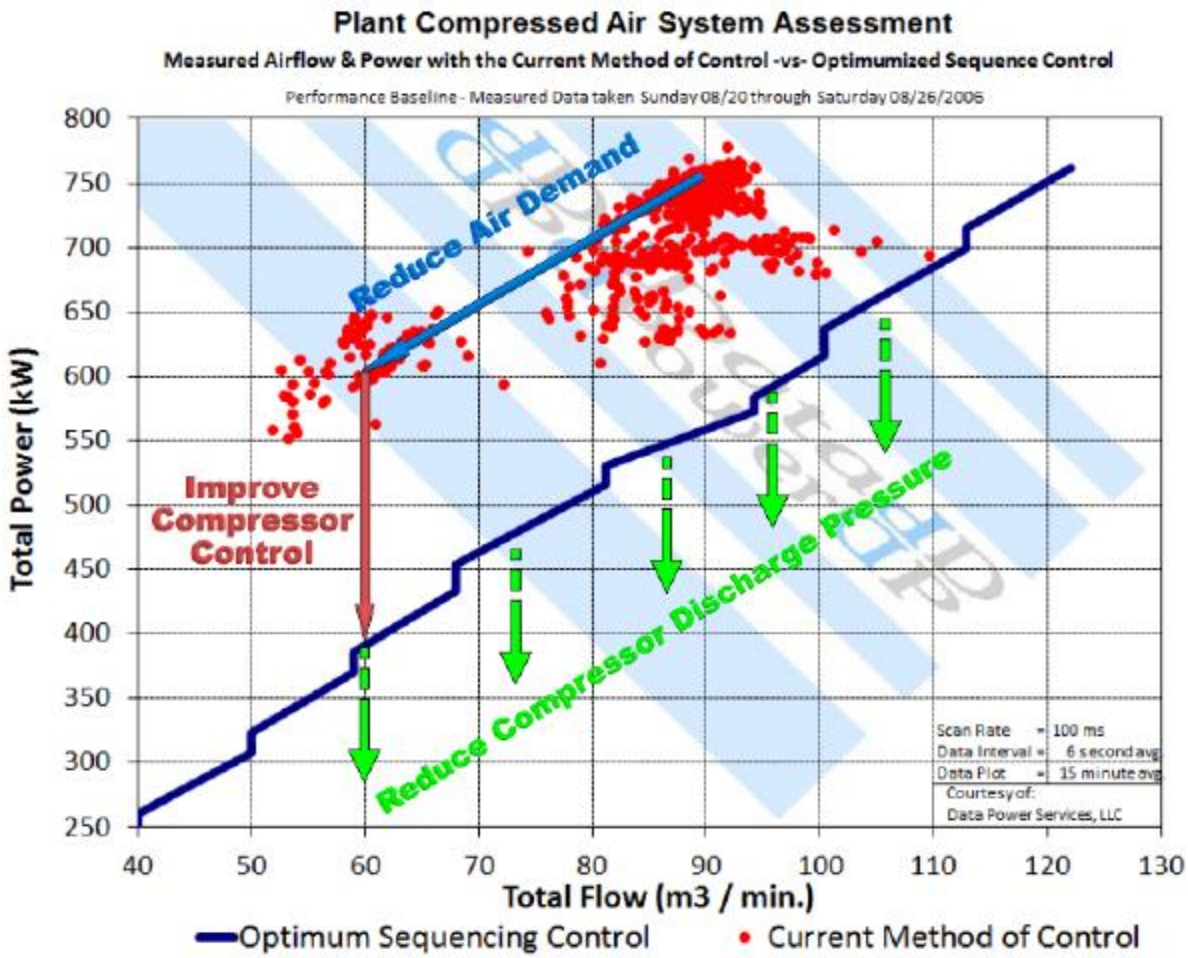



**Figure 1-5: Reducing System Energy Use**

The data chart in Figure 1-5 is a scatter plot of measured compressor power versus measured compressed airflow for a system with a total of eight air compressors operating to supply the plant's compressed air needs. The red dots are 15 minute averages for measured power and airflow rate. The dark blue line is the rated air compressor performance with an optimal control scenario where each compressor operates at full load capacity while only one compressor is operating as a trim capacity air compressor.

## c. Opportunity to Generate Air More Efficiently

The present compressor control function for the system in Figure 1-5 allows multiple compressors to operate at part load capacity resulting in the measured data indicating a greater power input at a given airflow rate as compared to the optimum sequence control. Air compressors are most efficient when operating at 100% full load capacity.

The optimum control strategy operates all running compressors at 100% capacity with the exception of one compressor that is operating as a trim capacity air compressor. Other compressors in the system should be turned off. Optimizing the operation of the multiple compressors supplying the system would reduce total power to the system at any given flow rate. The red dots of power / flow performance would move downward closer to the optimum blue line.

Most multiple compressor installations are poorly controlled. Investigation often reveals that multiple compressors are operating at part load capacity. The lack of a central controller and/or improperly adjusted individual compressor controls can account for significantly more power being used than is actually needed. In many systems, 20% or more energy reduction can be accomplished by simple adjustments to compressor controls. Additionally, properly adjusted and properly operating compressors may have greatly reduced maintenance expenses.

### d. Opportunity to Reduce Compressor Discharge Pressure

As the discharge pressure of a positive displacement compressor is reduced the power consumption of the compressor is reduced by about 6% for every 1 bar pressure (1% per 2 psig) reduction. In the data chart reduced discharge pressure would be represented by the entire blue line of optimum compressor performance moving straight down. At any given airflow rate, the compressor power consumption is lower when the compressor’s discharge pressure is reduced.

### e. Opportunity to Reduce the System’s Compressed Air Demand

The data plot shows that during times of reduced compressed air demand. The data shows at 90 $m^3$ / minute power is about 750 kW. During times when the compressed air flow is less, (about) 60 $m^3$ / minute the total system power usage is reduced by about 150 kW (from 750 kW to 600 kW).

## 1.5 The Business Case for Compressed Air System Management

Manufacturing facilities operate to produce products and generate profit for the company. Controlling and reducing cost is important to a company’s bottom line profitability. Compressed air cost is frequently overlooked. If production is interrupted by inadequate compressed air supply, cost is of little concern. The focus is to improve the reliability of production and cost concerns for compressed air are of secondary importance.

Decisions about investment and operation of the plant's compressed air system should receive the same management review as given to other investments in plant equipment and facilities.

Compressed air systems represent a significant investment in equipment, compressors, dryers, filters, etc. There is also significant infrastructure capital investment associated with compressed air systems, including, distribution piping, electrical supply to compressors, and cooling; ventilation or water cooling systems. With all of these costs considered, still as we discussed earlier life cycle cost analysis will reveal that the most significant component to compressed air system cost is the energy used to operate the air compressors.

Reliability and productivity of the system cannot be overlooked. Un-reliable compressed air systems that interrupt production, and cause lost productivity can also have significant impact on a plant's bottom line profitability.

The solution is application of the systems approach to create a properly designed and well managed compressed air system. Optimized compressed air system performance provides both increased reliability and energy savings. The business case for projects to improve and optimize compressed air system design and operation can have attractive return on investment.

## 1.6 Key Learning Points

In summary, properly designed and well managed compressed air systems have improved reliability and reduced energy cost as compared to poorly managed systems.

1. Compressed air is a necessary utility for industrial plants.

2. When compressed air is integral to the production process, it is a process variable.

3. System management must focus on productivity (controlling cost) rather than traditional goals.

4. The Systems Approach is an integrated approach, not component efficiency.

## 1.7 Key Energy Points

5. Energy cost is over 75% of the total life cycle cost to own and operate a compressed air system.

6. Generating compressed air is an inefficient energy conversion.

7. Avoiding the compression of air provides the greatest energy savings. Eliminate compressed air waste.

8. Many systems waste 50% or more of the compressed air that is consumed.

9. Including waste most compressed air systems have overall efficiency of < 10%.

10.Three basic opportunities to save energy include:

- Generate compressed air more efficiently
- Minimize pressure loss in the system
- Reduce compressed air demand

# 2. Understanding Compressed Air

Compressed air is pressurized atmospheric air. Compressed air's most common function is to act as an energy carrier. This enables energy to be transported across distances to perform work (as the air is returned to atmospheric pressure).

Atmospheric air is a mixture of gases. The primary constituents of the mix are nitrogen and oxygen. Argon and other trace gases make up about 1% of the mixture.

**Table 2-1: Components of normal atmospheric air**

| Components | Percent volume *volume may vary |
|---|---|
| Nitrogen | 78.08 |
| Oxygen | 20.95 |
| Argon | 0.93 |
| Carbon dioxide* | 0.03 |
| Neon | 0.018 |
| Helium | 0.00052 |
| Methane | 0.00015 |
| Krypton | 0.00011 |
| Carbon monoxide* | 0.0001 |
| Nitrogen monoxide* | 0.00005 |
| Hydrogen* | 0.00005 |
| Ozone* | 0.00004 |
| Xenon | 0.000008 |
| Nitrogen dioxide | 0.0000001 |
| Iodine | $2 \times 10^{-11}$ |
| Radon | $6 \times 10^{-18}$ |

Atmospheric air is formed from molecules that are bound to each other by molecular force. The molecules are in constant motion. The total molecular mass contained in a certain quantity of gas is very low. This allows a quantity of gas to be compressed to a very small proportion of its original volume.

If a gas is contained within a space, the continuous motion and collision of the molecules against the walls of the container create a force. Pressure is defined as force over a given area. For compressed air systems, the force measure is usually expressed as kPa, bar or psi (pounds per square inch). Other force measures for gases include Torr, inches or millimeters of mercury, inches or millimeters of water, millibar and others.

At atmospheric conditions and a temperature of 0oC, there are about 3 x 1023 molecular collisions per square centimeter per second. If a quantity of atmospheric gas is contained and the temperature is increased, the velocity and energy level of the molecules will increase. With increased, velocity and energy, the number of collisions with other molecules and with the walls of the container will increase and the pressure in the container will rise. Pressure, temperature and volume are proportionally interrelated.

If temperature rises with constant volume, pressure will rise. If volume is reduced with constant temperature, pressure will rise. This is the physical law that is utilized in most industrial compressed air technology. A compressor uses mechanical energy to reduce the space within which air is contained in order to increase the pressure of the air. This compressed air is then transmitted through a distribution system to a point of use, where the air can be re-expanded and the energy recovered to perform work.

## 2.1 Basic Definitions

### a. Pressure

Pressure: A force applied to a specified area.

$$\text{Pressure} = \frac{\text{Force}}{\text{Area}} \qquad 1\,\text{Pascal} = \frac{1\,\text{newton}}{1\,\text{m}^2} \qquad 1\,\text{bar} = 100\text{kPa} = \frac{14.5\,\text{pounds}}{1\,\text{inch}^2}$$

**<u>Absolute pressure</u>:** Pressure of a gas measured from absolute zero (an absolute vacuum). Absolute pressure is used for all theoretical compression calculations, and is used more commonly in vacuum applications than in compressed air applications.

**<u>Gauge pressure</u>:** Pressure measured above the prevailing ambient pressure. This is the practical unit of measure for compressed air systems. Gauge pressure is the measure of the differential pressure between the system and the local atmosphere and is one factor in determining the amount of energy available for work in a system.

**<u>Atmospheric pressure</u>:** The weight of the column of air above a particular point on the earth. This pressure varies with altitude. Atmospheric (ambient) pressure also has small variation with weather conditions.

**Table 2-2: Atmospheric air pressure, temperature and density at various altitudes**

| Altitude – meters | Pressure – bar | Temperature -- °C | Density – kg/m³ |
|---|---|---|---|
| 0 | 1.013 | 15.0 | 1.225 |
| 100 | 1.001 | 14.4 | 1.213 |
| 200 | 0.989 | 13.7 | 1.202 |
| 300 | 0.978 | 13.1 | 1.190 |

| | | | |
|---|---|---|---|
| 400 | 0.966 | 12.4 | 1.179 |
| 500 | 0.955 | 11.8 | 1.167 |
| 600 | 0.943 | 11.1 | 1.156 |
| 800 | 0.921 | 9.8 | 1.134 |
| 1000 | 0.899 | 8.5 | 1.112 |
| 1200 | 0.877 | 7.2 | 1.090 |
| 1400 | 0.856 | 5.9 | 1.060 |
| 1600 | 0.835 | 4.6 | 1.048 |
| 1800 | 0.815 | 3.3 | 1.027 |
| 2000 | 0.795 | 2.0 | 1.007 |
| 2200 | 0.775 | 0.7 | 0.986 |
| 2400 | 0.755 | -0.6 | 0.966 |
| 2600 | 0.737 | -1.9 | 0.947 |
| 2800 | 0.719 | -3.2 | 0.928 |
| 3000 | 0.701 | -4.5 | 0.909 |
| 3200 | 0.683 | -5.8 | 0.891 |
| 3400 | 0.666 | -7.1 | 0.872 |
| 3600 | 0.649 | -8.4 | 0.854 |
| 3800 | 0.633 | -9.7 | 0.837 |
| 4000 | 0.616 | -11.0 | 0.819 |
| 5000 | 0.540 | -17.5 | 0.736 |
| 6000 | 0.472 | -24.0 | 0.660 |
| 7000 | 0.411 | -30.5 | 0.590 |
| 8000 | 0.356 | -37.0 | 0.525 |

b. Flow rate of air expressed as volume flow rate or mass or weight of airflow rate:

| FAD (Free Air Delivered) volume flow rate |
| --- |
| FAD is the volume of air delivered at the discharge of an air compressor package. The volume flow rate is expressed at the prevailing ambient conditions of temperature, pressure, and relative humidity as they exist at the compressor intake. |

Free air delivered refers to a volume flow rate of air at ambient conditions, no matter what those ambient conditions are. Changes in pressure, temperature or relative humidity (changes in mass) do not change the FAD rating. This rating is, therefore, a measure of volume independent of the weight of air being delivered. The units of measure for FAD include $m^3$/min, liter/sec, acfm (actual cubic feet per minute), and many others.

| $Nm^3$/min (Normal cubic meters / minute) |
| --- |
| Normal cubic meters / minute ($Nm^3$/min) is a weight or mass flow rate measurement. Although $Nm^3$/min and $m^3$/min sound similar, they are as different as liters and kilograms. $Nm^3$/min refers to the weight (or mass) of air that occupies one cubic meter of space under a defined (normal or standard) condition of temperature, pressure and humidity conditions. |

There are several different specifications that define normal (standard) conditions (see Table 2-3). So before making calculations, determine the particular definition of "normal or standard" conditions that should be used. The resultant mass flow rate calculated is measured in units of Normal Cubic Meters / minute ($Nm^3$/min); Normal Cubic Meters / hour ($Nm^3$/hr); Normal Liters / second, (Nl/s); or other similar units of volume expressed at the defined normal condition. In U.S. nomenclature, this measure would be equivalent to SCFM or standard cubic feet per minute.

**Table 2-3: Volume related to various conditions**

| | Temperature | Pressure | Relative Humidity | Density |
|---|---|---|---|---|
| **Volume According to ISO 1217:2009** [1] | 20 $^{o}$C = 293.15 $^{o}$K | 1.0 bar(a) | 0% | 1.188 kg/m3 |
| **Volume according to DIN 1343** [2] **(normal physical state)** | 0 $^{o}$C = 273.15 $^{o}$K | 1.01325 bar(a) | 0% | 1.294 kg/m3 |
| **Volume according to DIN/ISO 2533** [3] | 15 $^{o}$C = 288.15 $^{o}$K | 1.01325 bar(a) | 0% | 1.225 kg/m3 |
| **Volume related to atmosphere** | atmospheric temperature | atmospheric pressure | atmospheric humidity | variable |
| **Volume related to operating state** | working temperature | working pressure | variable | variable |
| **Volume According to ASME** [4] | 68 $^{o}$F = 528 $^{o}$R | 14.69 psia | 36% | 0.07492 lb/ft$^3$ |
| **Volume According to CAGI** [5] | 68 $^{o}$F = 528 $^{o}$R | 14.5 psia | 0% | 0.07515 lb/ft$^3$ |
| **Volume According to ASHRAE** [6] | 70 $^{o}$F = 530 $^{o}$R | 29.92 in Hg 14.7 psia | 0% | 0.075 lb/ft$^3$ |

----- Chart courtesy of Kaeser Kompressoren, -----

[1] International Organization for Standardization ISO/TC 118/SC6 31-Jan-2010 ISO 1217:2009 Displacement compressors – Acceptance tests

[2] Deutsches Institut Fur Normung E.V. (German National Standard) / 01-Jan-1990 DIN 1343 Reference conditions, normal conditions, normal volume; concepts and values

[3] International Organization for Standardization / 01-May-1975 ISO 2533:1975 Standard Atmosphere

[4] American Society of Mechanical Engineers

[5] Compressed Air and Gas Institute

## 2.2 Gas Laws

Boyle's Law states that the volume (V) of a gas, at a constant temperature, varies inversely with pressure (P). If the volume of a contained gas is decreased, the pressure will increase. If the volume is increased, the pressure will decrease.

**Equation 2-1: Boyles Law**

$$\frac{\mathrm{P1}}{\mathrm{P2}} = \frac{\mathrm{V2}}{\mathrm{V1}} \quad \text{or} \quad \mathrm{P1} \times \mathrm{V1} = \mathrm{P2} \times \mathrm{V2}$$

Charles' Law states that a volume of gas, at a constant pressure, varies directly with absolute temperature (T). If the pressure is held constant and the temperature of the gas is increased, the volume will increase. If the pressure is held constant and the temperature is decreased, the volume will decrease.

**Equation 2-2: Charles Law**

$$\frac{\mathrm{V1}}{\mathrm{V2}} = \frac{\mathrm{T1}}{\mathrm{T2}} \quad \text{or} \quad \frac{\mathrm{V1}}{\mathrm{T1}} = \frac{\mathrm{V2}}{\mathrm{T2}}$$

Amonton's Law states that the pressure of a gas, at a constant volume, varies directly with the absolute temperature. If the volume is held constant and the gas is heated, the pressure will increase. If the volume is held constant and the temperature is lowered, the pressure will decrease.

**Equation 2-3: Amonton's Law**

$$\frac{\mathrm{P1}}{\mathrm{T1}} = \frac{\mathrm{P2}}{\mathrm{T2}} \quad \text{or} \quad \mathrm{P1} \times \mathrm{T1} = \mathrm{P2} \times \mathrm{T2}$$

Combining these three laws results in the following equation:

**Equation 2-4: Combined Gas Laws**

$$P1 \times \left(\frac{V1}{T1}\right) = P2 \times \left(\frac{V2}{T2}\right)$$

## 2.3 Volumetric Flow Rate

By definition, when air is compressed, the volume is reduced. The original mass is contained in a smaller space. Reducing the volume occupied by 7 cubic meters of ambient air to 1 cubic meter will raise the pressure from 1 bar absolute to 7 bar absolute, or 6 bar gauge.

| **Compression Ratio** |
|---|
| Compression Ratio (CR) of air or other gas is defined by the volume reduction ratio before and after compression ($V_1$ / $V_2$). As shown in Figure 2-1 the initial volume $V_1$ is 7 $m^3$ and the final volume $V_2$ is 1 $m^3$, giving a compression ratio of 7. |

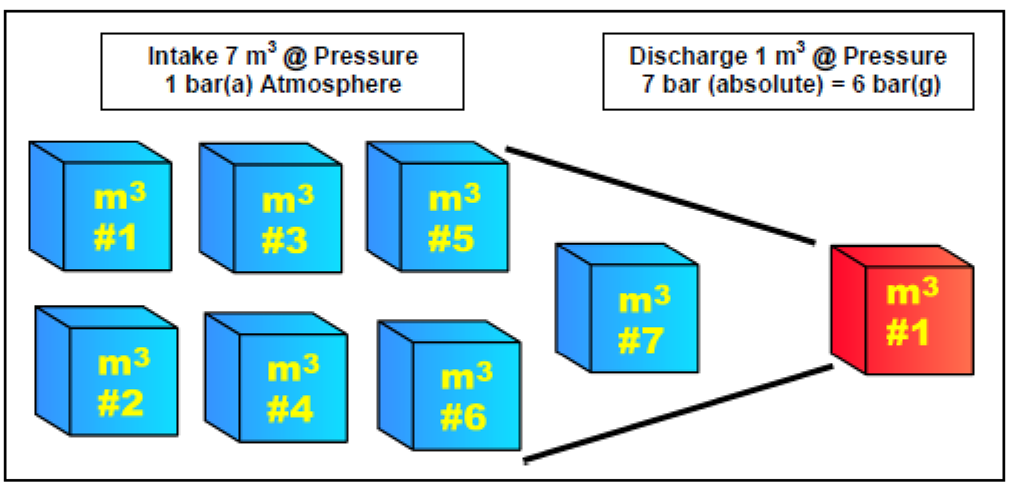



**Figure 2-1: Air Compression - volume relationship**

Compression ratio (CR) can also be calculated based on the ratio of the absolute pressure increase (assuming constant temperature).

**Equation 2-5: Combined Gas Laws**

$$\mathrm{CR} = \frac{\mathrm{V1}}{\mathrm{V2}} = \frac{\mathrm{P2}}{\mathrm{P1}} \text{ in the example above } \frac{6\ \mathrm{bar(g)} + 1\ \mathrm{bar(a)}}{1\ \mathrm{bar}} = 7$$

Using the combined gas law, it is known that compression of air will also result in an increase in the air temperature. As a result the overall Pressure Ratio (PR) is much greater than the Compression Ratio (CR).

**Equation 2-6: Overall Pressure Ratio**

From the ideal gas law:

$$\mathrm{CR} \rightarrow \frac{\mathrm{V1}}{\mathrm{V2}} = \frac{\mathrm{T1}}{\mathrm{T2}}\frac{\mathrm{P2}}{\mathrm{P1}} \quad \text{or} \quad \mathrm{CR} = \frac{\mathrm{T1}}{\mathrm{T2}}\ \mathrm{PR}$$

Note -- If all of the heat could be retained and if the compressed air could be used expansively, most of the energy required to compress the air could be recovered to do work. Unfortunately, a large amount of the work required to compress air is released as waste heat and many air tools do not use air expansively. Many air tools discharge air at about the same pressure as the inlet to the tool. If the tools could make use of the expansive nature of the compressed air, more energy could be recovered.

For most power applications, the compressed air requirements are usually expressed in volume terms that relate to a particular amount of ambient air compressed to a particular pressure. A tool with a requirement of 0.5 $m^3$/min at 7 bar needs a little more than one half cubic meter of ambient air compressed to 7 bar gauge (8 bar absolute). A little more than one half a cubic meter is required because compressors have some losses associated with their operation.

For most process applications, requirements are usually expressed in some form that represents a mass or weight measure. This may be expressed as kilograms of air or volume according to a DIN standard or some other nomenclature such as “Normal Cubic Meter”.

## 2.4 Mass Flow Rate

To convert the volume flow rating of a particular compressor to a mass measure, several things are required:

1. A definition of the standard condition to be used: (DIN 1343 in the formula below) given as:
   Temp. 0 $^{O}$C = 273.15 $^{O}$K Pressure = 1.01325 bar(a) relative humidity 0%

2. The volume flow rating of the compressor (Manufacturer's rating)

3. The maximum ambient temperature at the inlet to the compressor

4. The minimum ambient atmospheric pressure

5. The maximum ambient relative humidity

6. The water vapor saturation pressure at various temperatures

From this information, the mass flow of a particular compressor at a particular set of conditions can be calculated using the general gas equation as a basis.

**Equation 2-7: Mass flow calculated from rated volumetric flow & job-site conditions**

$$V_n = \frac{V_o \times T_N \times \left(P_A - (F_{rel} \times p_D)\right)}{P_N \times T_O}$$

*Where:*

$V_N$ = *Volume to DIN 1343*

$V_0$ = *Volume flow rating of the compressor*

$T_N$ = *Normal temperature to DIN 1343, 273.15°K*

$T_0$ = *Job-site ambient temperature at installation in °K*

$P_N$ = *Normal absolute ambient air pressure to DIN 1343, 1.01325 bar(a)*

$P_A$ = *Job-site absolute ambient air pressure at installation in bar(a)*

$F_{rel}$ = *Job-site relative humidity at installation*

$p_D$ = Saturation pressure of the water vapor contained in the air in bar,

*(Partial vapor pressure at job-site temperature & relative humidity)*

For saturation pressures, in bar, at various air temperatures; see Table 2-4. Remember, temperature must be converted to an absolute rating before using the equation. Celsius or Centigrade must be converted to Kelvin and Fahrenheit must be converted to Rankin.

**Table 2-4: Water vapor saturation pressure at various temperatures**

| $^{o}C$ | $p_D$ | $^{o}C$ | $p_D$ | $^{o}C$ | $p_D$ |
|---|---|---|---|---|---|
| -10 | 0.0026 | 10 | 0.0123 | 30 | 0.0424 |
| -9 | 0.0028 | 11 | 0.0131 | 31 | 0.0449 |
| -8 | 0.0031 | 12 | 0.0140 | 32 | 0.0473 |
| -7 | 0.0034 | 13 | 0.0150 | 33 | 0.0503 |
| -6 | 0.0037 | 14 | 0.0160 | 34 | 0.0532 |
| -5 | 0.0040 | 15 | 0.0170 | 35 | 0.0562 |
| -4 | 0.0044 | 16 | 0.0182 | 36 | 0.0594 |
| -3 | 0.0048 | 17 | 0.0194 | 37 | 0.0627 |
| -2 | 0.0052 | 18 | 0.0206 | 38 | 0.0662 |
| -1 | 0.0056 | 19 | 0.0220 | 39 | 0.0699 |
| 0 | 0.0061 | 20 | 0.0234 | 40 | 0.0738 |
| 1 | 0.0064 | 21 | 0.0245 | 41 | 0.0778 |
| 2 | 0.0071 | 22 | 0.0264 | 42 | 0.0820 |
| 3 | 0.0074 | 23 | 0.0281 | 43 | 0.0864 |
| 4 | 0.0081 | 24 | 0.0298 | 44 | 0.0910 |
| 5 | 0.0087 | 25 | 0.0317 | 45 | 0.0968 |
| 6 | 0.0094 | 26 | 0.0336 | 46 | 0.1009 |
| 7 | 0.0100 | 27 | 0.0356 | 47 | 0.1061 |
| 8 | 0.0107 | 28 | 0.0378 | 48 | 0.1116 |
| 9 | 0.0115 | 29 | 0.0400 | 49 | 0.1174 |
| | | | | 50 | 0.1234 |

If the air requirements are specified as weight of air, such as kg, then this weight of air has to be divided by the standard weight, according DIN 1343 (1.294 kg/m3). This will result in "Standard Cubic Meters" with DIN 1343 being the standard.

## 2.5 Why two different flow ratings: Volumetric and Mass Flow?

### a. Mass Flow ($Nm^3$ / minute)

The amount of pneumatic energy available to a pneumatic tool, actuator, or process end use application is a function of the mass flow of compressed air delivered. The actual volume of air delivered at the pipeline conditions of pressure, temperature, and water vapor determines the mass flow rate of air to the end use tool or process. When considering air demand and end use consumption of pneumatic energy the mass flow ($Nm^3$ / minute) is of greatest importance.

### b. Volumetric Flow ($m^3$ / minute)

Air compressor manufacturers will typically rate their compressor's performance in volumetric terms of FAD (free air delivery). You may need to observe if end use demand requires an expression of mass flow; then the compressor manufacturer should provide the mass flow delivery of their compressors, not volumetric flow.

However, the mass flow delivery of an air compressor is dependent on two factors.

• The volumetric delivered airflow of the compressor, and

• the job-site conditions of ambient pressure, temperature, and relative humidity.

Since the compressor manufacturer does not know if a compressor will be installed at sea level or 1000 meter elevation; or if the compressor will be in a hot & humid region near the equator; or a cold and dry environment at the North or South Poles; or somewhere between; they cannot provide delivered airflow performance of an air compressor in mass flow terms. That is only possible when the job-site conditions at the compressor installation are known. Even then normal seasonal changes in ambient conditions can significantly affect air compressor delivered mass flow rate.

### c. Summary; Volumetric or Mass Flow

Understanding that it is impossible for compressor manufacturers to produce performance specifications that report mass flow for every job-site on the planet, compressed air system experts, and plant personnel need to understand their roll in air compressor applications. Since the system expert and plant personnel have the necessary information about the job-site ambient conditions at the compressor installation; they need to use the manufacturer's published data and interpret how it applies to their specific compressed air system.

The actual mass flow output ($Nm^3$ / minute) delivered by the plant's air compressor(s) is the pneumatic energy available to the manufacturing end use applications. When ambient conditions result in the lowest air density at the compressor's intake, the compressor delivers the least mass flow. Therefore, compressor intake conditions of high air temperature, high relative humidity, and low intake air pressure will result in the lowest mass flow rate of compressed air delivered from the compressor(s). To support the manufacturing process with the best reliability it is essential to know that the required mass flow rate of pneumatic energy to support productive end use demands will be available at this worst case condition.

### d. Sample Calculation – Air Compressor mass flow (Nm3/minute) airflow rate calculated from manufacturer's volumetric rating.

A plant is adding the first of two new identical production lines and is told by the Original Equipment Manufacturer (OEM) that each production line requires 2.0 Nm$^3$/min airflow rate at 5.8 bar(g) for operation at full rated production capacity. The normal reference condition used by the OEM is in accordance with DIN 1343. Also the plant has an old air compressor with a manufacturer's rated performance of 4 m$^3$/min at 7 bar discharge pressure. The decision has been made to replace the aging air compressor.

The plant is considering purchase of a new 45 kW lubricant injected single stage rotary screw compressor with a manufacturer's performance rating of 8.26 m3/min at 7.5 bar(g) discharge pressure.

The plant is situated at 600m elevation with ambient condition at the job-site compressor installation of 35$^o$C, and 70% RH during the hottest summer condition.

**Is the proposed 45 kW compressor capable of supplying the compressed air system?**

First both compressors, the old unit being retired and the new compressor being considered are rated in volumetric terms of m$^3$/min. Subtracting the 4.0 m$^3$/min from the new compressor rating of 8.26 m$^3$/min leaves a difference of 4.26 m$^3$/min.

A calculation is necessary to see if the remaining capacity of the new compressor; 4.26 m$^3$/min is great enough to satisfy the air demand of the two new production lines with total air demand of 4.0 Nm$^3$/min.

$$V_n = \frac{V_o \times T_N \times \left(P_A - (F_{rel} \times p_D)\right)}{P_N \times T_O}$$

*Where:*

$V_N$ = *Volume to DIN 1343*

$V_0$ = *Volume flow rating of the compressor*

$T_N$ = *Normal temperature to DIN 1343, 273.15$^o$K*

$T_0$ = *Job-site ambient temperature at installation in °K*

$P_N$ = *Normal absolute ambient air pressure to DIN 1343, 1.01325 bar(a)*

$P_A$ = *Job-site absolute ambient air pressure at installation in bar(a)*

$F_{rel}$ = *Job-site relative humidity at installation*

$p_D$ = Saturation pressure of the water vapor contained in the air in bar,

*(Partial vapor pressure at job-site temperature & relative humidity)*

$$V_n = \frac{4.26\text{m}^3/\text{min} \times\ 273.13^{\circ}\text{K} \times \left(0.973\text{bar(a)} - (\ 0.7 \times 0.0562\text{bar(a)}\ )\right)}{1.01325\text{bar(a)} \times\ 308.15^{\circ}\text{K}}$$

$$V_n =\ 3.5\text{Nm}^3/\text{min}$$

The result of the calculation shows that the 4.26 $m^3$/min remaining volumetric airflow capacity of the proposed air compressor (after subtracting 4.0 $m^3$/min for the compressor capacity being replaced) is equivalent to 3.5 $Nm^3$/min (DIN 1343) at the given job-site condition. The system requires 4.0 $Nm^3$/min capacity to supply the two new production lines.

Is the proposed 45 kW compressor capable of supplying the compressed air system?

No, the 45 kW compressor has 0.5 $Nm^3$/min too little delivered air capacity!

## 2.6 Introduction to Pneumatic Energy Transmission

The transmission portion of a compressed air system moves pneumatic energy from the supply equipment where the compressed air is generated, to the end use demands that accomplish the production tasks. The distribution piping is the primary component of the transmission system. However, the transmission system may also include components such as filters, air receivers, control valves, or other components.

Every component of the transmission system that compressed air passes through offers some restriction creating a resistance to the flow of compressed air. In the first section we learned that a well designed and constructed system will transmit pneumatic energy from the supply side generation to end use demands with minimal energy (pressure) loss.

Pressure loss in components of the pneumatic energy transmission system is dependent on the amount of restriction through the component and the air flow rate (or fluid velocity) of compressed air passing through the restriction.

### a. Piping Restriction and pressure loss

The restriction to airflow through a length of pipe is directly proportional to the length of the pipe. If 100m of pipe has 0.2 bar pressure loss at a given airflow rate, then 200m of the same pipe passing the same airflow rate will have double the pressure loss or 0.4 bar.

**Equation 2-8: Pressure loss as a function of pipe length**

$$\frac{\Delta P2}{\Delta P1} = \frac{L2}{L1} \quad \text{or} \quad \Delta P2 = \Delta P1 \frac{L2}{L1}$$

$$\Delta P2 = 0.2 \times \frac{200\text{m}}{100\text{m}} = 0.4$$

### b. Piping Velocity and pressure loss

The velocity of air through the pipe line also impacts the amount of pressure loss in a given length of pipe. Pressure loss in a fluid system is proportional to the change in fluid velocity squared. Also consider that (at constant temperature and pressure) the change in velocity of a compressed air line is equal to the change in the compressed air flow rate. Therefore, for compressed air flowing at constant pressure and temperature, pressure loss changes as the square of the change in compressed air flow rate.

**Equation 2-9: Pressure loss as a function of compressed air flow rate**

$$\frac{\Delta P_2}{\Delta P_1} = \left(\frac{v_2}{v_1}\right)^2 \quad \textit{and} \quad \frac{v_2}{v_1} \approx \frac{Q_2}{Q_1} \quad \textit{(@ Const. temperature \& pressure)}$$

$$\textit{Therefore,} \quad \frac{\Delta P_2}{\Delta P_1} = \left(\frac{Q_2}{Q_1}\right)^2 \quad \textit{or} \quad \Delta P_2 = \Delta P_1 \left(\frac{Q_2}{Q_1}\right)^2$$

The velocity of compressed air in the distribution piping must be evaluated when designing new or reviewing an existing design of a compressed air system. Frequently designs include calculated pressure drop values based on average airflow rate and tables of flow through pipe and fittings. Peak airflow rate in compressed air piping is often much greater than the average airflow rate.

Many pneumatically powered tools and processes have great variation in their compressed airflow demand throughout time. Tools such as drills, grinders, chipper hammers, etc. rarely operate continuously, the air use starts and stops frequently and randomly. Other applications such as a dense phase transport may have an airflow requirement of 8.4 $m^3$/min (140 l/s; 300 cfm) specified by the manufacturer. However, when reviewing the operating cycle, it is discovered that during filling of the transport pressure pot (~ 30 seconds of time) there is very little compressed air use. When pressurizing the pot and transporting the material (~ 30 seconds of time) compressed air use is at its highest. In this example the actual airflow rate will be essentially zero for 30 seconds followed by the peak flow rate of 16.8 $m^3$/min for 30 seconds. What the average airflow is 8.4 m3/min, the peak flow is double that rate. Valves and piping must be designed with consideration of the peak airflow rate that will occur.

Compressed air flow creating excessive velocity will result in pressure drops that will greatly exceed the calculated pressure drop using average flow and reference tables. Excessive velocity may also result in moisture and debris being pushed past the drain legs of the distribution piping.

### c. Compressed Air Piping Design Velocity

Throughout the years there have been a number of recommendations for maximum compressed air pipeline velocity. In the past it may have been common to design mainline piping to operate at 15 meters per second velocity.

Recommended design velocity targets no more than 6 meters per second in the mainline and major branches of air distribution piping. The velocity should be designed for the typical peak airflow rate that will occur. Velocity in piping connections leading to a point of use should not exceed 15 meters per second in short runs (less than 15 meters).

**Equation 2-10: Maximum air flowrate Q (l/s) for pipe diameter d (mm)**

$$\textit{Mainline \& Branch piping } v = 6 \ {}^{m}\!/_{\sec} \Rightarrow Q = \frac{(P_L + P_A)\times d^2}{212}$$

$$\textit{Point of use piping } (\leq 15 \ m \ long) \ v = 15 \ {}^{m}\!/_{\sec} \Rightarrow Q = \frac{(P_L + P_A)\times d^2}{85}$$

$$NOTE: multiply \ Q(l/s)\times 0.06 = Q(m^3/minute)$$

*Where:*

*Q = the flow rate in liters per second*

*$P_L$ = the pipeline gauge pressure in bar (g)*

*$P_A$ = the atmospheric pressure in bar(a)*

*d = the internal diameter of the pipe in millimeters*

### d. Sample Calculation – check recommended pipe size for automated packaging machine

For example, check the manufacturer's recommendation for compressed air supply pipe size for an automatic packing machine. The manufacturer's machine specification recommends ¾ BSP connection (BS:1387 spec I.D. = 21.7 mm). Specifications also state "compressed air supply 30 cfm (14.1 l/s; 0.85 $m^3$/min) at 600 kPa (6 bar(g))" and the plant is located at sea level atmospheric pressure = 1 bar(a).

The pneumatic end use devices on the machine consist of small pneumatic cylinders, grippers, and rotary actuators. The operating cycle is short and the machine once operating runs continually. Due to the number and diversity of pneumatic end use devices and the rapid continuously repeating cycle the average airflow and peak airflow are assumed to be nearly equal.

The pipeline is the connection from the branch header to the point of use connection on the packaging machine. A design velocity of 15 m/sec is selected. Using

Equation 2-10 check maximum airflow rate for the recommended pipe size.

$$Point\ of\ use\ piping\ (\leq 15m\ long)\ v = 15\ m/sec$$

$$Q = \frac{(P_L + P_A) \times d^2}{85}$$

$$Q = \frac{(6.0 + 1.0) \times 21.7^2}{85}$$

$$Q = 38.8\ l/s$$

The calculation shows the machines air demand of 14.1 l/s is about ½ of the maximum airflow rate for the ¾ BSP pipe operating at 6 bar(g) and a design velocity of 15 m/sec.

NOTE: it is good to consider leakage that will likely occur and increase the air flow rate to the machine. Even with additional 20% leakage, the pipeline airflow capability is well within design.

Conclusion; the ¾ BSP pipe connection is suitable.

Another quick way of picking pipe size for distribution systems is to use the following chart. Keeping the acceptable pressure drop to 0.1 bar, or lower, will result in pipe sizes the keep velocities near 6 meters per second.

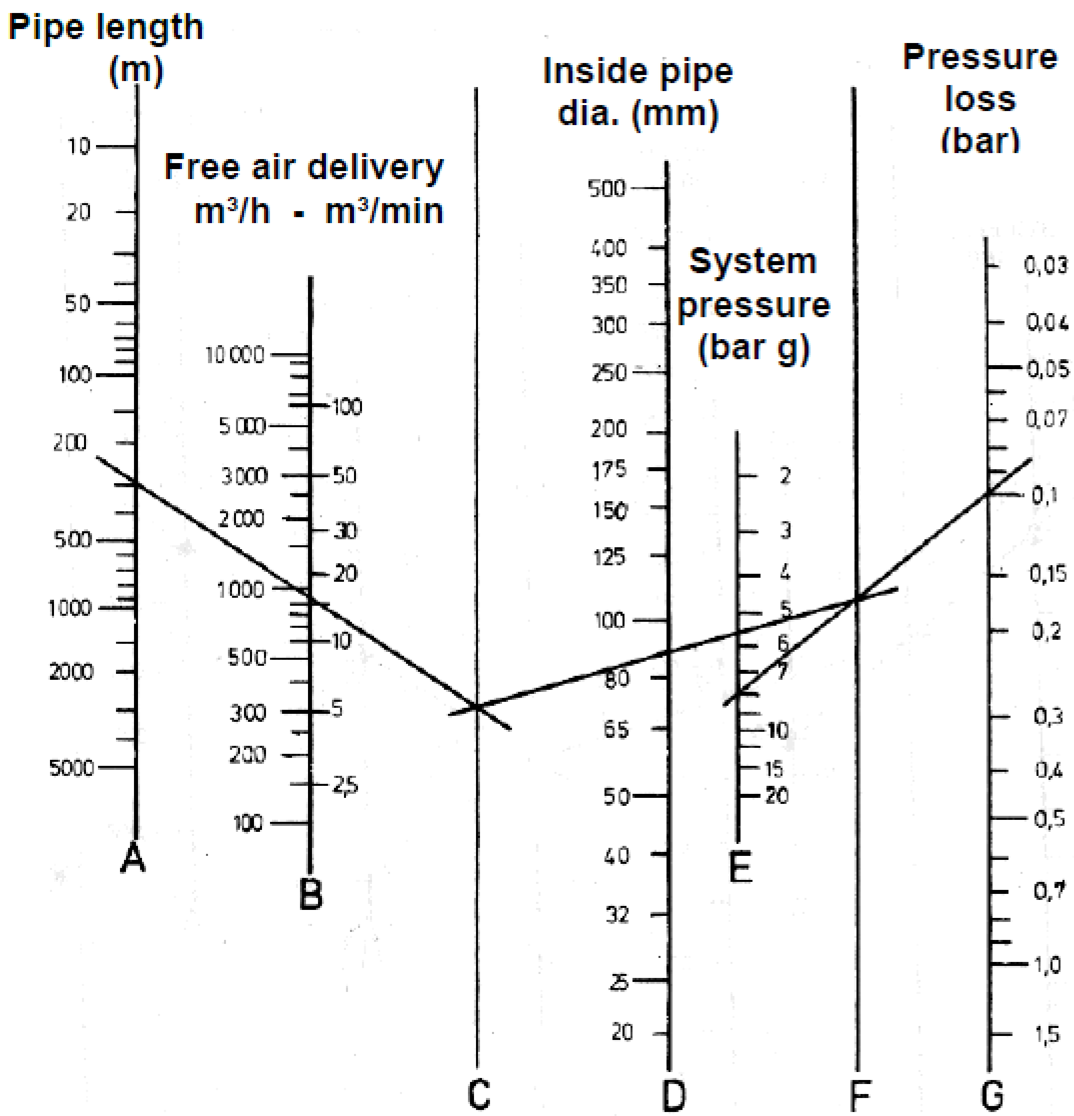



**Figure 2-2: Pipe sizing chart**

To use this chart, begin by marking the "A" axis at the point that corresponds to the total length of pipe and the "B" axis at the point that corresponds to the total air flow for that pipe. In the above example, 15 cubic meters of air pre minute must move through 300 meters of pipe. Draw a line through the points marked on "A" and "B" and extend the line through the "C" axis. Next, mark the "E" axis with the system gauge pressure and the "G" axis with the desired pressure loss. Draw a line between those two points and note where the line crosses the "F" axis. Finally, draw a line between the intersection points on the "C" and "F" axes. Where this line crosses the "D" axis, read the inside pipe diameter required to meet the operating conditions.

## 2.7 Compressed Air Storage Introduction

One of the most important and most overlooked aspects of a compressed air system is storage. The proper amount of compressed air stored in the system can allow production to continue, without interruption, during certain short peak demand events, when a compressor fails and another must start, and while additional compressors start in response to a sustained increase in demand.

The effect of storage on compressor efficiency is also important. Rotary screw compressors with Load/Unload run more efficiently with large storage volumes.

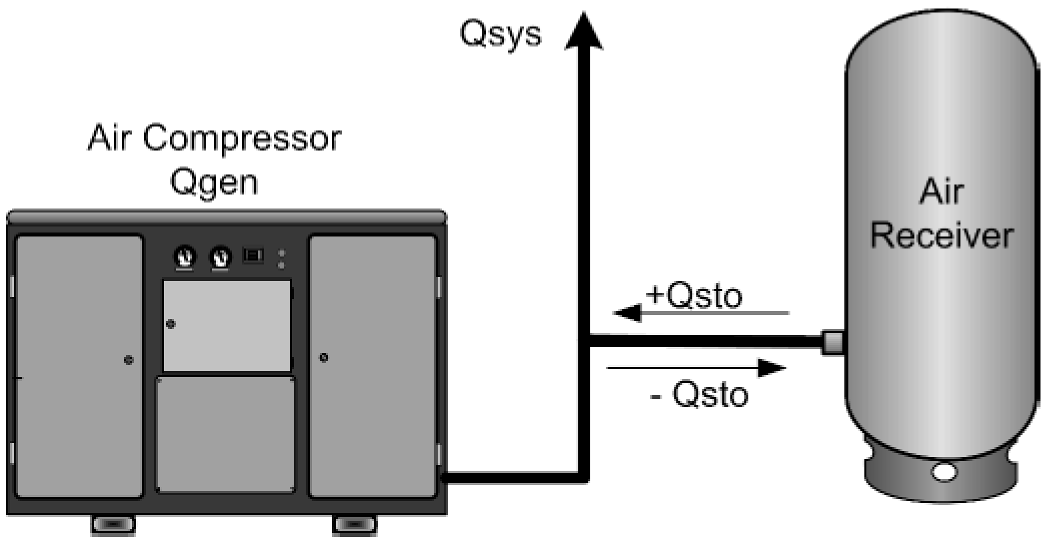



**Figure 2-3: System Flow Balance: System Airflow = Generation +/- Storage**

Useable compressed air available from storage is the product of pressure differential and available storage volume. The formula for determining the amount of useable air storage is:

**Equation 2-11: Usable air volume Va available from Storge**

$$V_a = V_s \times \left(\frac{P_{max} - P_{min}}{P_{amb}}\right)$$

Where:

$V_a$ = Useable compressed air in storage

$V_s$ = Total volume of storage system

$P_{max}$ = Maximum storage or receiver pressure (cut-out pressure)

$P_{min}$ = Minimum storage or receiver pressure required (cut-in pressure)

$P_{amb}$ = Absolute ambient air pressure

As the formula demonstrates, changes in actual volume of the storage system OR changes in pressure differential can create useable storage. It also demonstrates that if there is no pressure differential, there is no useable storage.

When properly applied, storage capacity allows implementation of a control strategy that will efficiently maintain balance between compressed air supply and demand.

## 2.8 Key Learning Points

1. Compressed air is a common method of transmitting energy to pneumatic tools and devices.

2. The work accomplished by compressed air is dependent on the weight of air delivered to the end use equipment.

3. The weight of air is dependent on the conditions of pressure, temperature, and relative humidity.

4. Pressure, volume, and temperature are interrelated, in this relationship air is treated as an ideal gas.

## 2.9 Key Energy Points

5. As compressed air energy is transmitted from one location to another, pressure loss is an irrecoverable loss of energy.

6. The amount of pressure loss is related to the velocity in the compressed air pipeline.

7. Compressed air energy can be stored.

8. The amount of useable compressed air energy in storage depends on the storage tank's volume and pressure differential between the storage pressure and minimum system pressure requirement

# 3. Supply Side: Compressors and Their Application

## 3.1 Types of Compressors

Compressors can be divided into two basic types, positive displacement and dynamic. Positive displacement compressors operate by trapping a volume of air in a confined space and then reducing the physical size of that space until the desired pressure is achieved. Dynamic compressors create pressure by imparting velocity to the air stream. Dynamic compressors do not trap a volume of air or gas. They use blades of various designs to accelerate the air to the point that pressure increases as the airflow is slowed in special chambers.

Positive displacement compressors can be subdivided into a number of classes. Two basic classes are reciprocating and rotary. A bicycle pump is an example of a reciprocating compressor. A piston travels in a cylinder with valves opening and closing to let air in, trap the air, compress the air, and allow the compressed air out.

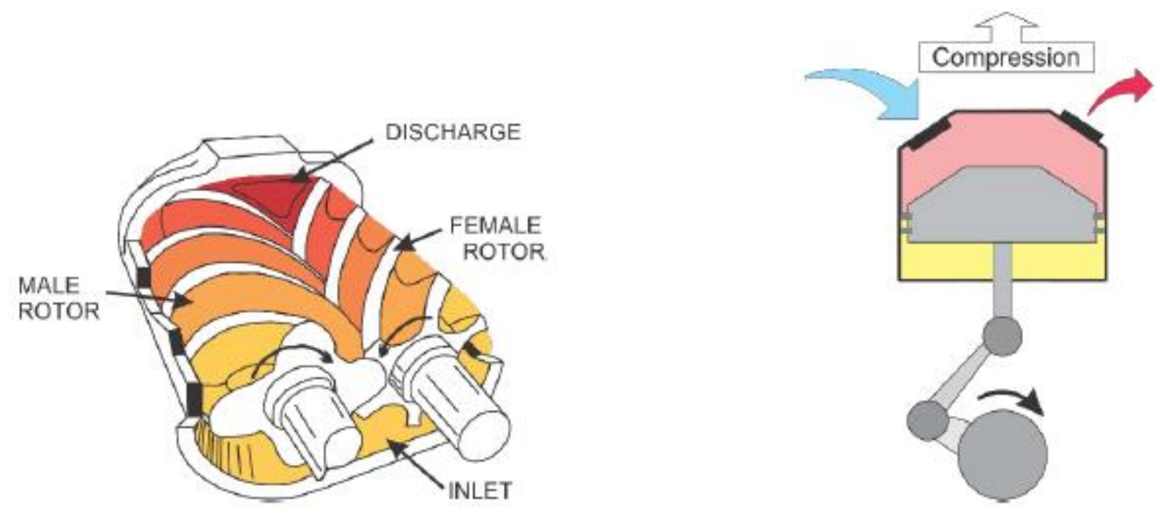



**Figure 3-1: Positive Displacement Rotary Screw & Reciprocating Compressors**

Rotary screw air compressors are the most commonly used positive displacement style of air compressor in industry today. Rotary screw compressors are positive displacement compressors just like the simple bicycle pump. During the compression process air is trapped in a flute of the female rotor. As the male lobe engages the female flute, the trapped volume is reduced and air pressure is increased.

Two common types of rotary compressors are rotary screw and rotary vane. There are many rotary style positive displacement compressors including scroll, liquid ring, hook & claw (roto-step), lobe style, and others.

## 3.2 Stages of Compression: Single-stage, Two-stage, Multi-Stage

The compressors shown in Figure 3-1 are examples of single stage compressors. Compression from intake to final discharge pressure takes place in a single stroke of the piston, or single revolution of the rotary screw compressor.

A two-stage compressor reaches the final discharge pressure by first compressing to an intermediate pressure with the first stage and then to the final pressure in the second stage. That is to say that there are two distinct steps to reach the final discharge pressure. The compression ratio of each stage is the square root of the overall compression ratio. Compressing to 7.5 bar gauge will result in a compression ratio per stage of 2.92. The compression ratio per stage of a three-stage compressor is the cube root of the overall compression ratio. Multiple-stage compressors have an efficiency advantage if the compressed air can be cooled between the compression stages. The power required to compress air is a function of the mass and the compression ratios.

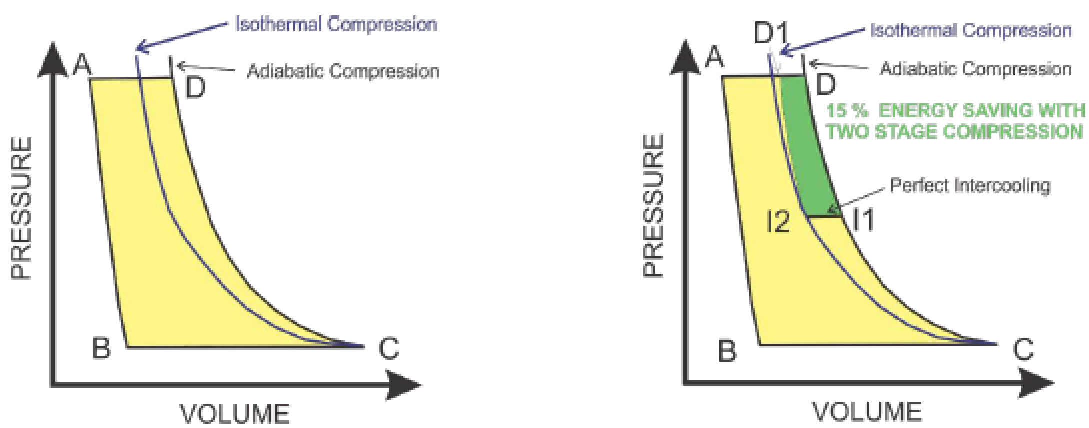



**Figure 3-2: Compression Cycle - Single Stage and Two Stage**

While the thermodynamics and theory of compression is interesting, the real world operation of air compressors requires a measure of compressor operating performance. The International Organization for Standardization (ISO) has developed a standard for Displacement compressors – Acceptance tests; ISO 1217:2009 specifies methods for acceptance tests regarding volume rate of flow and power requirements of displacement compressors.

Acceptance test codes were originally written to be a test method that could be used to confirm that a compressor built to a customer's specific requirements for flow, pressure and power actually met those requirements. That is fine for custom-built compressors, but most compressors sold to industry are not custom-built. Most compressors are of a standard design and built in batches or in continuous production quantities and are fully piped and wired as a complete, self-contained compressor. For these types of compressors, ISO developed ISO 1217

Annex C for fixed speed compressors and Annex E for variable speed compressors. These annexes are a "Simplified acceptance test for electrically driven packaged displacement air compressors".

Performance factors for fixed speed compressors include the following (and more):

- Rated Capacity at Full Load Operating Pressure.
- Total compressor package input power (kW) at rated capacity ($m^3$ / min) and full load operating pressure.
- "Specific Package Input Power at Rated Capacity and Full Load Operating Pressure.

Performance indicators are reported (for various capacity compressors) with-in tolerance specified in ISO 1217, Annex C as shown in Table 3-1 below.

**Table 3-1: Tolerance Specified in ISO 1217**

| Volume Flow Rate at specified conditions | | Volume Flow Rate | Specific Energy Consumption | No Load / Zero Flow Power |
|---|---|---|---|---|
| $m^3$ / min | $ft^3$ / min | % | % | +/- 10% |
| Below 0,5 | Below 15 | +/- 7 | +/- 8 | |
| 0,5 to 1,5 | 15 to 50 | +/- 6 | +/- 7 | |
| 1,5 to 15 | 50 to 500 | +/- 5 | +/- 6 | |
| Above 15 | Above 500 | +/- 4 | +/- 5 | |

### a. Specific Power

Compressed air **Specific Power** is a good measure of efficiency expressed as input power per unit of compressed air output; for example, kW / $Nm^3$/minute (kW / 100 scfm) are measures of power per unit of mass flow output. Specific power in terms of volumetric output could be expressed air kW / $m^3$/minute (kW / 100 acfm) for free air delivery (FAD).

Specific power can be applied to a single compressor's power and airflow or the total of all compressors supplying the system and total air flow to the system. Considering the energy input to air dryers along with purge air consumption of air dryers if present. Total supply side energy input and net air flow delivered to the system give specific power of the entire supply side.

### b. Specific Energy

Compressed air **Specific Energy** is a good measure of efficiency expressed as input energy per unit of compressed air output; for example, kWh / $Nm^3$ (kWh / 100 scf) are measures of power per unit of mass flow output. Specific power in terms of volumetric output could be expressed air kWh / $m^3$ (kWh / 100 acf) for free air delivery (FAD).

Specific energy can be applied to a single compressor's power and airflow or the total of all compressors supplying the system and total air flow to the system. Considering the energy input to air dryers along with purge air consumption of air dryers if present. Total supply side energy input and net air flow delivered to the system give specific power of the entire supply side.

### c. Specific Power for Various Compressor Types

While the basic compressor design is a significant factor in determining compressor efficiency, other design factors also affect the full load specific power of an air compressor. Factors include:

- Drive motor efficiency
- Mechanical drive losses; e.g. belt drive, gear drive
- Compressor speed and frictional loss
- Internal leakage (slippage) and resultant volumetric efficiency.
- Internal compressor valves, port size, airflow passages, and pipe size

As a result, with-in and compressor type there is a range of specific power depending on the individual manufacturer's compressor design and operating characteristics. When considering compressor performance, it is necessary to contact the compressor manufacturer for rated performance and factory test performance data.

**Table 3-2: Specific Power for Various Compressor Types**

| Specific Power for Various Compressor Types (typical range) | | | |
|---|---|---|---|
| Volumetric flow rate (free air delivery) | kW / $m^3$/ min | kW / 100 l/sec | kW / 100 cfm |
| Recip. Single Acting (sgl stage) | 7.8 - 8.5 | 47 - 51 | 22 - 24 |
| Recip. Single Acting (2 stage) | 6.4 - 8.1 | 38 - 49 | 18 - 23 |
| Recip. Double Acting (sgl stage) | 8.5 - 10.2 | 51 - 61 | 24 - 29 |
| Recip. Double Acting (2 stage) | 5.3 - 5.7 | 32 - 34 | 15 - 16 |
| Lubricated Screw (sgl stage) | 6.0 - 7.8 | 36 - 47 | 17 - 22 |
| Lubricated Screw (2 stage) | 5.7 - 6.7 | 34 - 40 | 16 - 19 |
| Lubricant Free Screw (2 stage) | 6.4 - 7.8 | 38 - 47 | 18 - 22 |
| Centrifugal (3 stage) | 5.7 - 7.1 | 34 - 42 | 16 - 20 |

Specific power measurements above as an indication of air compressor efficiency applies only to the full load operating condition. That is to say the compressor is delivering full rated capacity at the rated discharge pressure.

Many air compressors operate for some amount of time at less than their full load rated capacity. Air compressors are equipped with capacity control regulation. Most capacity controls respond to air pressure at the compressor discharge. When the air pressure reaches a predetermined set point the compressor stops producing air. System pressure will usually then fall until it reaches a lower pressure set point and the air compressor again begins compressing air. Other capacity control systems operate in a progressive fashion. As discharge pressure increases the compressor progressively reduces it output. If pressure. If discharge air pressure decreases the compressor will progressively increase its delivered airflow until it reaches full load capacity.

## 3.3 Reciprocating Compressors

Courtesy of Kaeser Compressors

**Figure 3-3: Two-stage, single-acting, duplex reciprocating compressor package**

Single-stage reciprocating compressors intake air and compress the air to the final discharge pressure in one stroke. The compression ratio of a single-stage compressor is the ratio of the final absolute discharge pressure to the absolute inlet air pressure. A compressor with a discharge of 7.5 bar gauge (8.5 bar absolute) will have an overall compression ratio of 8.5 to 1.

A single-stage compressor with an 8.5 to 1 compression ratio will require more power to compress the same volume than a two-stage compressor with 2.92 compression ratios per stage, for a total compression ratio of 8.52 (2.92 x 2.92).

Operating specific power:

- 7.8 – 8.5 kW/m3/min single stage single acting.
- 5.3 – 5.7 kW/$m^3$/min two stage double acting.

Reciprocating compressors can be further classified as single acting or double acting. A single acting compressor compresses the air or gas on one side of the piston. This is similar to the action of an automotive piston. A double acting compressor compresses on both sides of the piston. Compressing in both directions of travel is more efficient than compressing in just one direction. Therefore, a double acting compressor is usually more efficient than a single acting compressor.

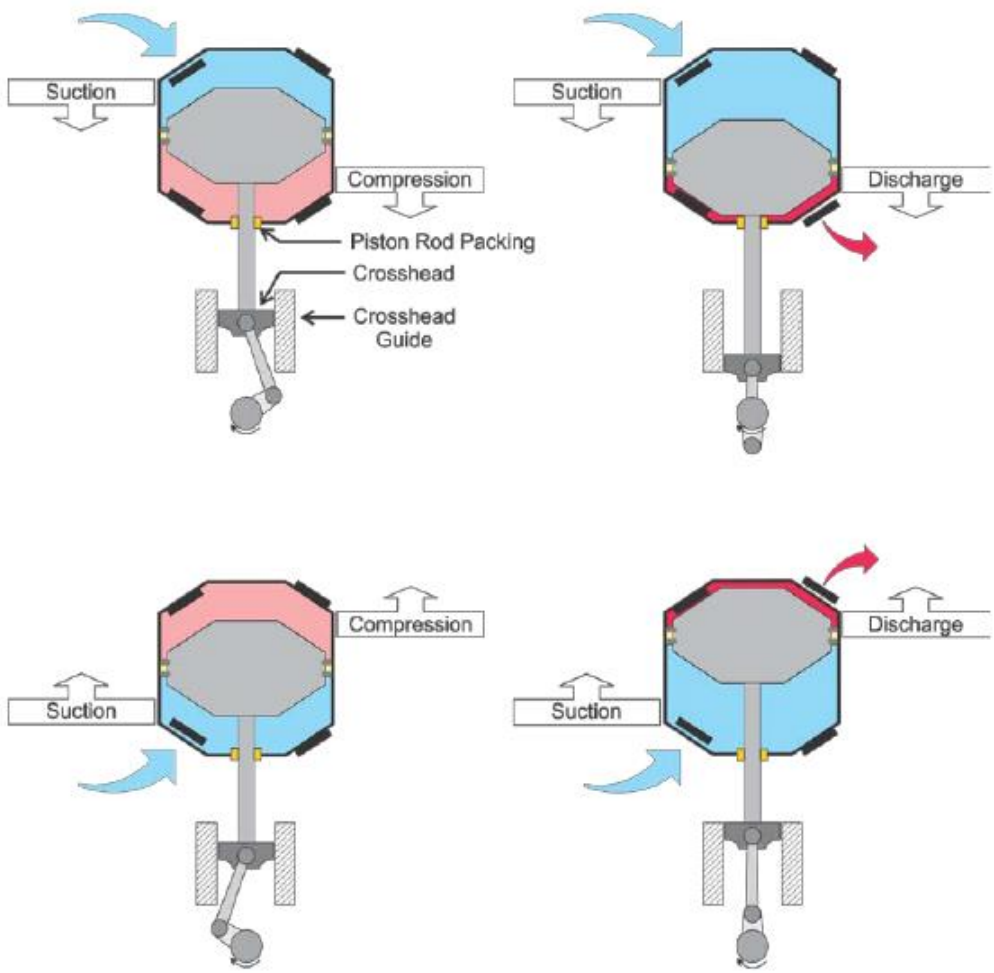



**Figure 3-4: Single Stage Double Acting Compressor**

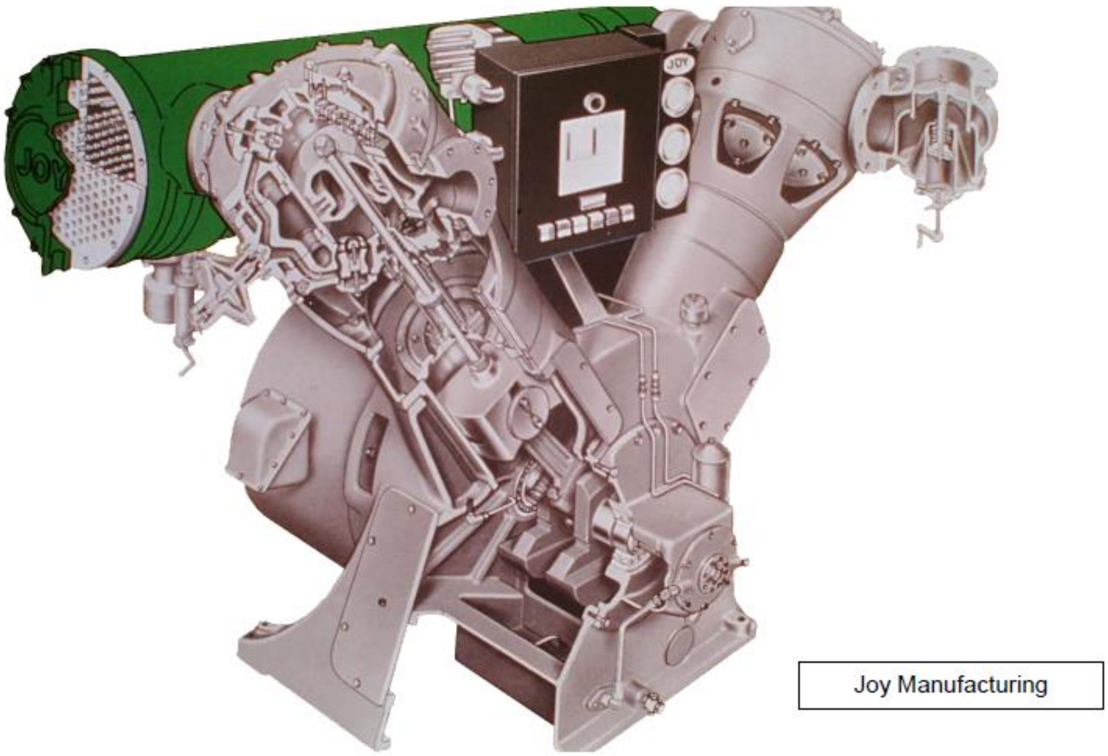



**Figure 3-5: Two-stage, double-acting reciprocating compressor (Joy Mfg.)**

All reciprocating compressors have design features that limit their overall efficiency. There is a clearance between the piston face and the top of the compression chamber. Air or gas that is compressed into this space will re-expand on the inlet stroke. This re-expanded air will occupy volume in the cylinder that would have otherwise been filled with ambient air through the inlet valve. Additionally, this compressed air has been heated and will provide some preheating to the inlet air, reducing the amount that can enter the cylinder. The valves and piston rings are wear items. A reciprocating compressor is most efficient when the valves and rings have been worn in and are properly sealing. From that time forward, the valves and rings gradually wear and lose their ability to maintain a tight seal. Air or gas can escape past these components and reduce the efficiency of the compressor.

## 3.4 Rotary Screw Compressors

**Figure 3-6: Lubricant-injected rotary screw compressor air end (Kaeser Compressors)**

Rotary screw compressors are the most common type of positive displacement compressor used in industry. These compressors can be subdivided into lubricant-injected and dry. The lubricant injected screw compressors are used primarily for general industrial air supplies. The dry type of rotary screw is commonly found in food, pharmaceutical, and electronics applications where contamination is a concern. With proper filtration and monitoring, lubricant injected compressors can also be used in some of these applications.

Rotary screw compressors have two rotors mounted inside a housing with an inlet port on one end and a discharge port on the other end. The male rotor has lobes that form a helix down part of the rotor. The female rotor has corresponding flutes in which the rotor lobes ride. As the pair of rotors passes the inlet port, air fills the flutes, or grooves, in the female rotor. The rotors continue to turn in the housing and pass a point where the air in the flute is cut off from the inlet port. Rotation continues, causing the volume of the trapped air to be reduced. Compression continues to take place until the pocket of trapped air reaches the discharge port. The compressed air is then pushed out of the discharge line and into a separator reservoir (if it is a lubricated screw) or directly to the compressor's aftercooler (if it is a dry screw).

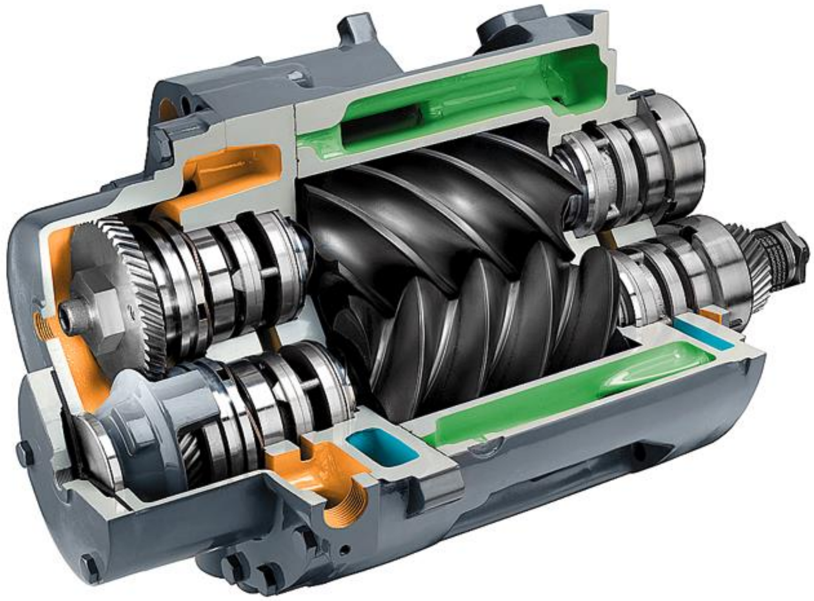


**Figure 3-7: Lubricant-free dry rotary screw compressor air end (Atlas Copco)**

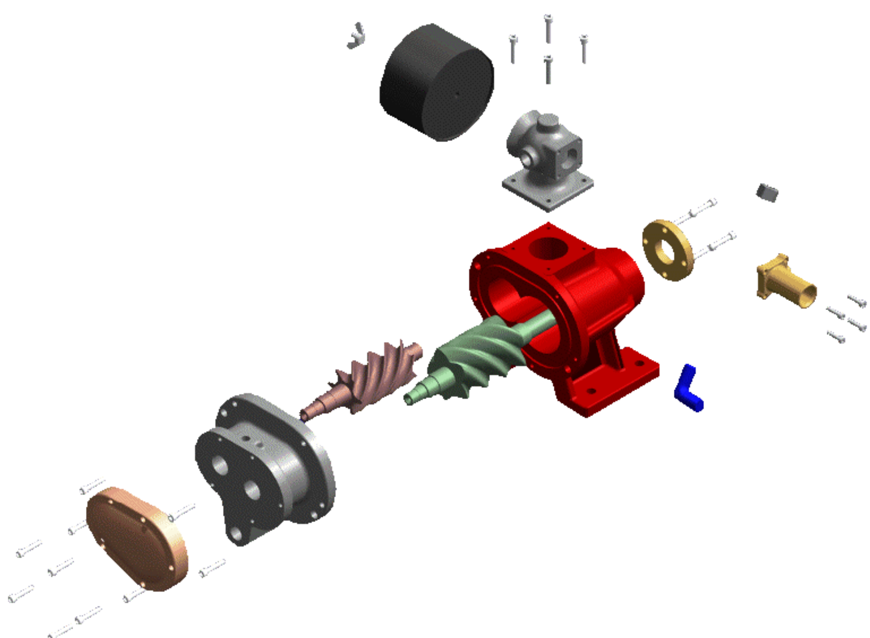


**Figure 3-8: Exploded view of typical single-stage, rotary screw air end (Kaeser Compressors)**

As discussed there are various geometric profiles used in rotary compressors. The helical rotary screw, asymmetrical profile with a female rotor with machined flutes will engage a male rotor with corresponding lobes that form the compression chamber along the length of the rotary screw compression element (also often referred to as an "air end").

Rotary screw compressors may be single-stage or two-stage. Remember, two-stage compressors compress air from a given inlet pressure to the final discharge pressure in two steps or stages.

The first stage takes a given volume from the absolute inlet pressure to an intermediate pressure. The second stage takes a volume at the intermediate pressure and compresses it to the final discharge pressure. While the overall compression ratio remains the same, regardless of the number of stages, the compression ratio per stage for a two-stage compressor is the square root of the overall compression ratio.

Since the power required to compress a gas is a function of the sum of the compression ratios per stage and the mass flow, single-stage compression, with a higher total compression ratio, would require more power than a two-stage compressor compressing the same volume of mass. That is the way things work...in theory.

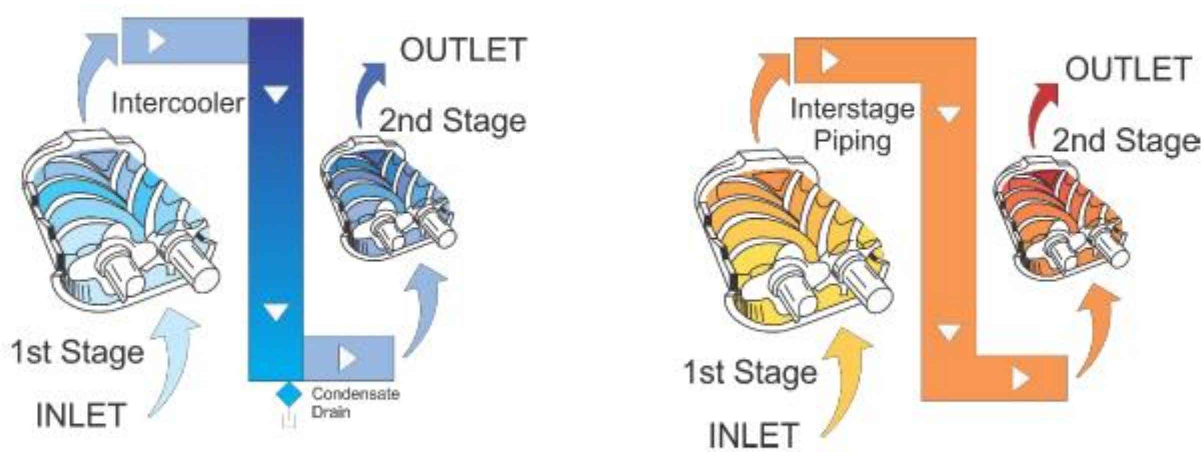



**Figure 3-9: Two-stage Lubricant Free & Lubricant Injected Compressors**

The ideal compression ratio per stage assumes several things. There must be perfect intercooling. The inlet temperature to each stage must be the same. Each stage must be sized and ported exactly. The compressor must run at a single design point or be able to vary the volume of both stages. In fluid-flooded, two-stage rotary screw compressors, intercooling is not possible. To cool the air stream between stages would result in water condensation that would immediately mix with the lubricating fluid as it entered the second stage. Some cooling fluid is usually injected between the stages, but it does not cool the interstage temperature to the ambient temperature. Sizing and porting of each stage is done for operation at full load.

Two stage rotary screw compressors generally operate between 5.6 and 7.8 kW/m3/min at full load capacity.

Operating specific power:

- 6.4 – 6.7 kW/$m^3$/min single stage lubricant injected.
- 5.6 – 6.0 kW/$m^3$/min two stage lubricant injected.
- 6.4 – 7.8 kW/$m^3$/min single stage lubricant free.

Real world performance will vary based on the manufacturer's design, compression ratio between stages changes, and operating pressure changes in the field. Furthermore, as the compressor controls reduce capacity there are inherent changes in the pressure ratio for each stage. This is presently and area of some controversy and discussion as to the actual part load capacity performance for two-stage rotary screw compressors with various types of capacity control.

### a. Lubricant Injected Rotary Screw Compressors

Lubricant injected rotary screw air compressors are the most common industrial air compressor in today's market. In addition to economical fist cost when purchased, installation and maintenance are much less costly as compared to the two-stage double acting reciprocating compressor. The disadvantage that lubricant injected rotary screw compressor is less efficient (6.0 to 7.8 kW / $m^3$/ min) than the reciprocating compressor (5.3 to 5.7 kW / $m^3$/ min) is overshadowed by the rotary screw's lower cost.

Operation of the rotary screw compressor creates a vacuum at the inlet of the rotary screw element. This suction created airflow through the intake filter, over the inlet control valve, and into the rotary screw element. Lubricant injection ports allow a mist of lubricant to enter the suction end of the rotary screw element and the air / lubricant mixture is compressed as the inter-meshing helical male and female screws rotate together. A mixture of high pressure air / lubricant exits the discharge end of the rotary screw element and is piped to the Lubricant Separator / Reservoir.

The piping and internal baffle create a rotational swirling flow of compressed air / lubricant mixture as it enters the lubricant reservoir. The large drops of oil impinge on the lubricant reservoir wall as a result of the swirling action and centrifugal force. The lubricant accumulates in the base of the reservoir.

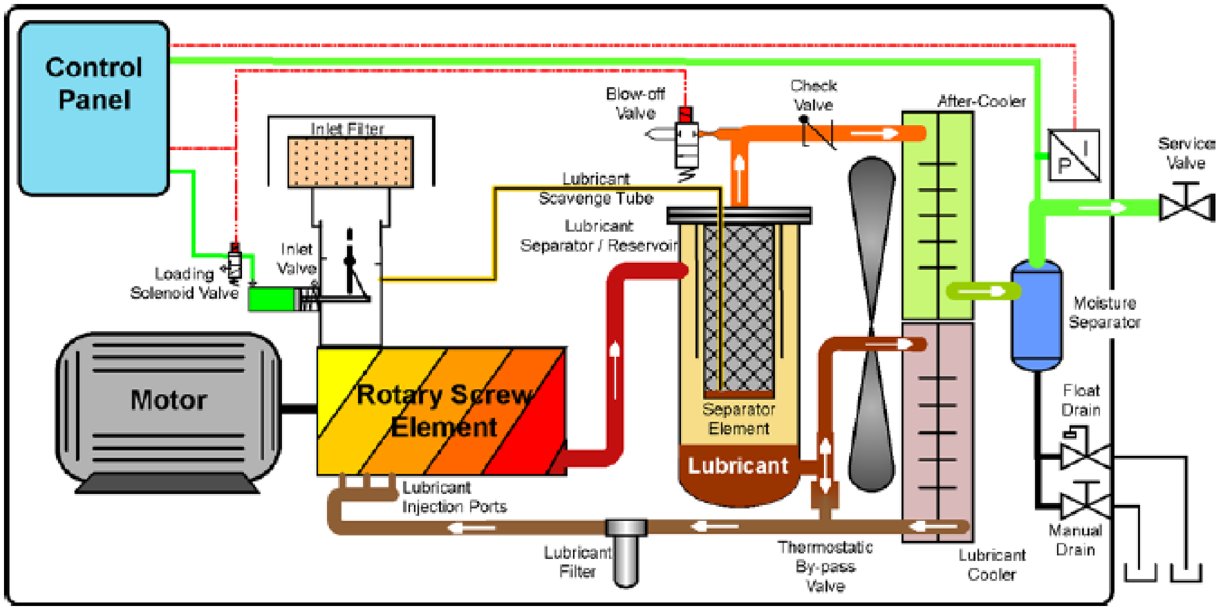



**Figure 3-10: Rotary Screw Compressor Internal Diagram**

Small lubricant droplets and aerosols are entrained in the compressed air as it passes through the separator filter element. The coalescing action of the separator filter element fiber media coalesce large drops of lubricant that fall by gravity and accumulated in the center of the separator filter element. A small diameter lubricant scavenge tube is connection to vacuum at the compressor intake and continually extracts the accumulated lubricant from the center of the separator filter element and returns the lubricant to the rotary screw element.

Compressed air containing a residual amount of lubricant (typically 2 – 5 ppm) is piped from the lubricant separator / reservoir outlet through a check valve to the compressed air after-cooler. The compressed air is cooled forming liquid condensate that is separated by the centrifugal moisture separator and discharge through the automatic condensate drain. The compressed air at line pressure enters the system from the compressor's service valve.

Lubricant in the reservoir is also at line pressure which pushes the lubricant cooler, filter and back into the lubricant injection ports at the suction end of the rotary screw element. Upstream of the lubricant cooler, a thermostatic by-pass valve will allow cold lubricant to bypass the lubricant cooler for rapid warm up of lubricant on start-up and to help maintain proper lubricant temperature and viscosity during periods of cold weather operation.

## 3.5 Rotary Screw Compressor Control Methods

Various air compressor designs have different full load efficiency shown in Table 4 - 3.2 expressed in terms of specific power (kW / $m^3$/minute). For most air compressor types full load capacity is the most efficient operating point. Variable speed drive compressors are an exception where-in the most efficient operation is often in the range of 80% to 85% capacity with slightly lower efficiency at full load capacity.

Few air compressors operate at full load capacity during every minute of operation. Or to put it another way, most air compressors operate for some amount of time at somewhat less delivered airflow as compared to their 100% full load rated capacity.

Rotary screw compressors have a number of different types of capacity control. Each capacity control method has unique characteristics and unique performance. When considering part load performance for a specific manufacturer and model of compressor a performance chart of power –vs– delivered airflow (Figure 3-11) can be created.

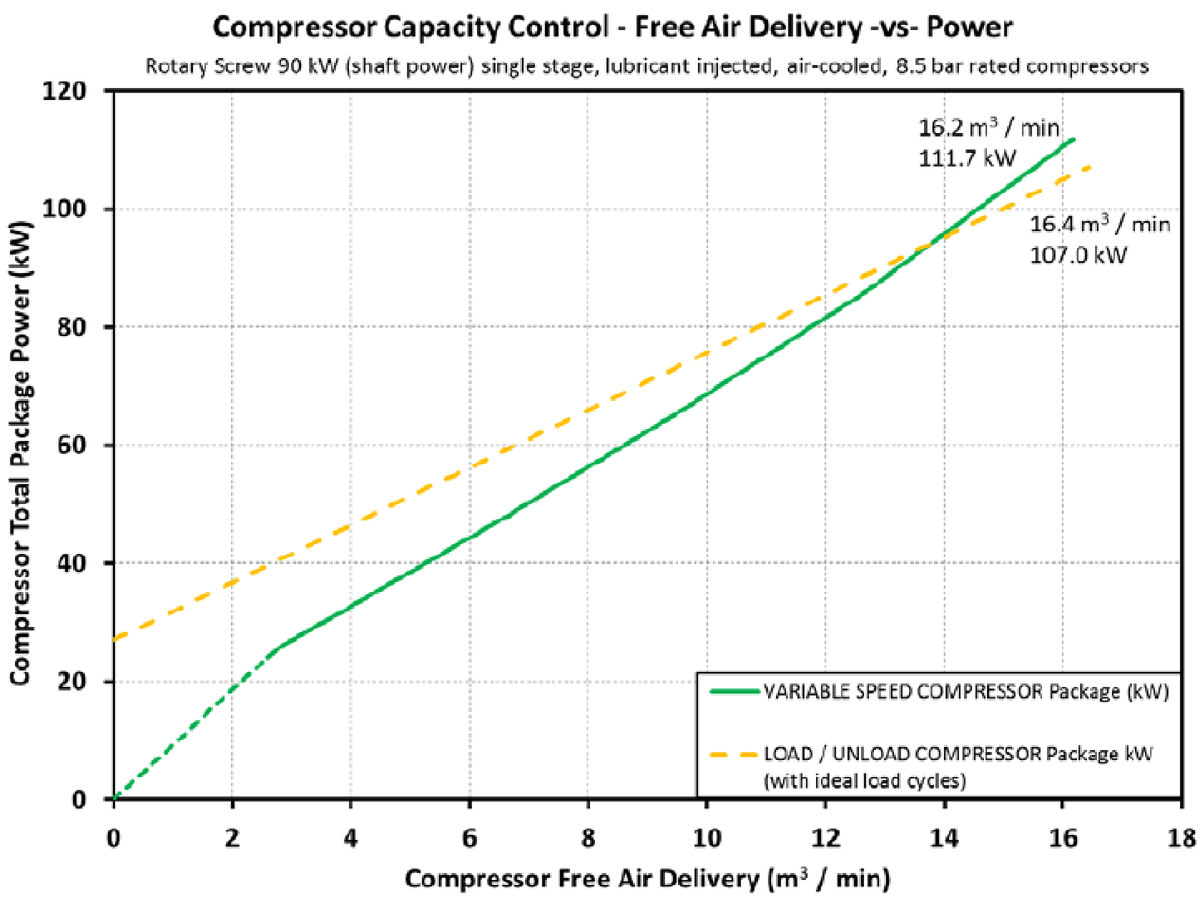



**Figure 3-11: Compressor Performance Curves Free Air Delivery -vs- Power**

Another method of describing part load performance is to chart specific power –vs– delivered airflow as in Figure 3-12 below.

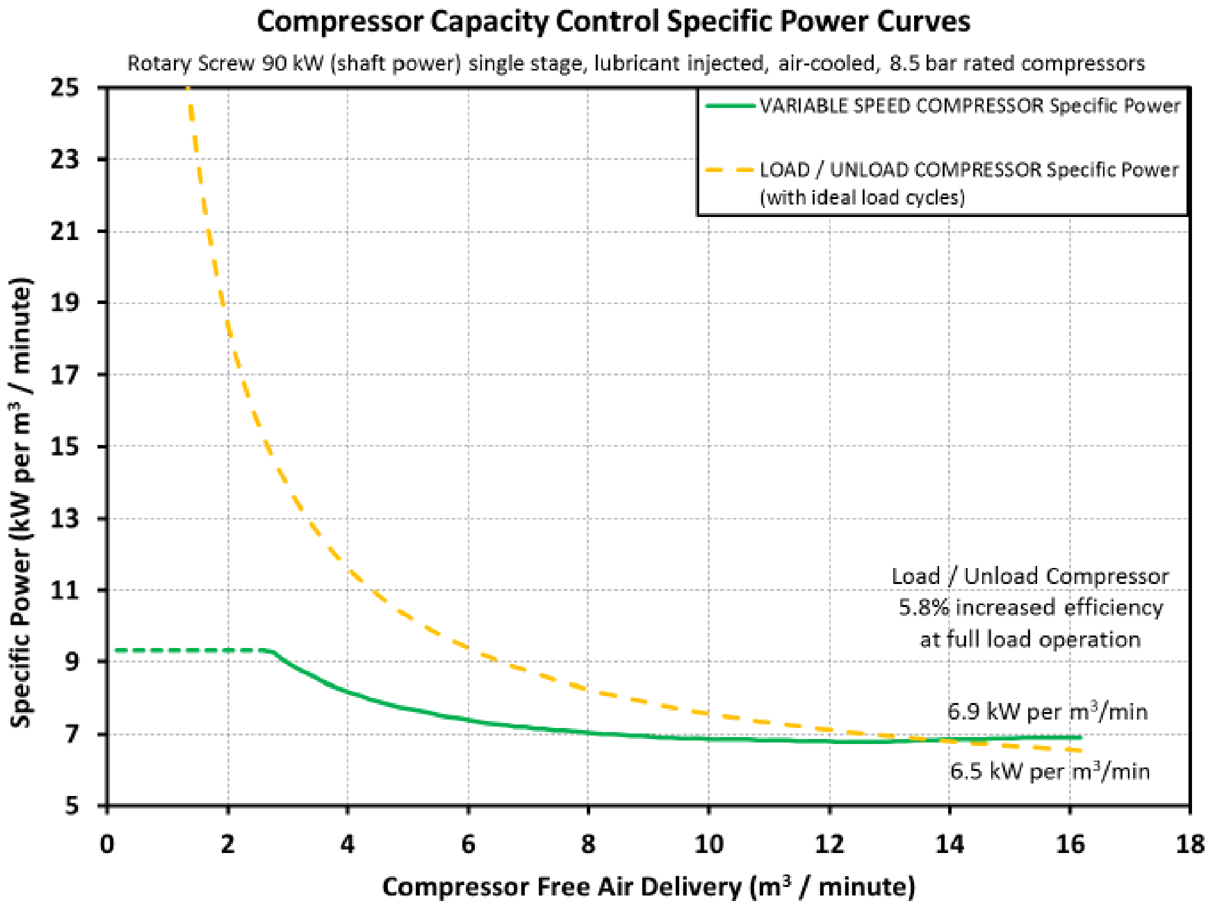



**Figure 3-12: Compressor Performance Specific Power Curves**

When comparing different types of capacity control, actual flow and power performance data for a specific model of air compressor, is the best comparison.

During this training various capacity control methods are discussed in generic terms without the actual performance data for specific air compressor manufacturers and models performance data. These generic comparisons relate percent of full delivered capacity and percent of full load power which are “typical” of various compressor control methods. Using a percentage comparison as in Figure 3-13 on the following page, does not reveal the 5.8% full load efficiency difference between the compressors shown in Figure 3-11 and Figure 3-12 above.

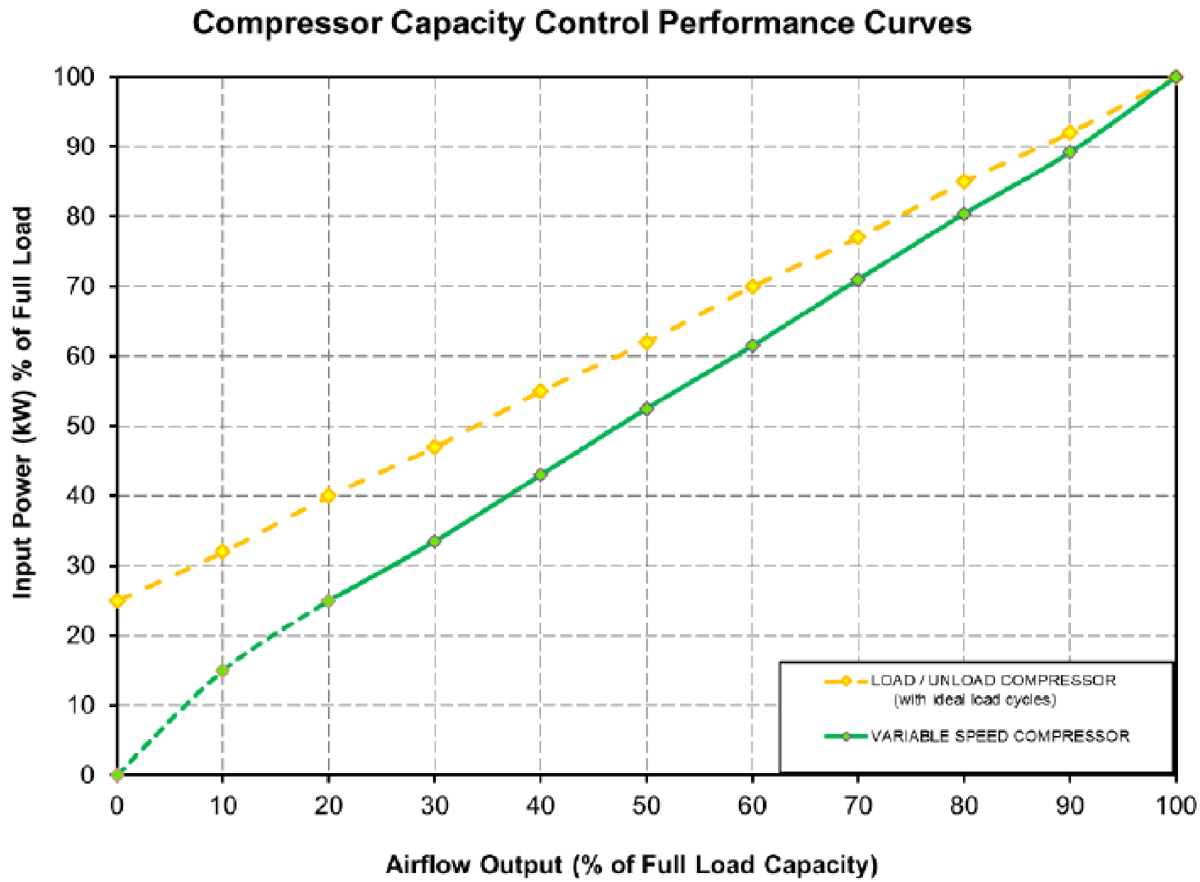



**Figure 3-13: Compressor Performance Curves % Capacity -vs- % Power**

## a. Start / Stop Control:

The simplest type of control uses a pressure switch to start and stop the compressor's drive motor. When the pressure falls to the low pressure or "cut-in" set point the pressure switch closes, the motor starts and the compressor operates. As the air pressure increases eventually reaching the high pressure or "cut-out" set point the pressure switch opens and the compressor's motor stops.

While start / stop control is appropriate for a small fractional kW compressor that might be used occasionally in the home or small shop, it is not appropriate for larger kW industrial air compressors. Of greatest concern is for high kW motors the inrush current associated with frequent starts will eventually overheat the motor windings causing damage or a complete motor failure.

### b. Load / Unload (On-line / Off-line or Load / No Load) Control:

Load / Unload control functions with a pressure switch using cut-in and cut-out set points to control the compressor. The difference is the unlike start / stop control, the electric drive motor runs continuously. When the cut-out pressure is reached, the pressure switch causes the compressor's inlet valve to close and no air is compressed. As the pressure decreases, the cut-in pressure is reached and the pressure switch signals the inlet valve to open and air is again compressed.

| Load / Unload Control |
|---|
| A capacity control method that allows an air compressor to run at full load or at zero (no load) compressed air flow delivery while the main motor driver continuously to run at constant speed. Load / Unload control is used with many different types of air compressors. |

Load / Unload control is one of the earliest control schemes for rotary screw compressors and was the simple adaptation of reciprocating-type controls to the rotary screw compressor. These pressure-based controls typically operate within a pressure range of about 0.7 to 1 bar. The compressor will run at full capacity until the measured system pressure reaches the upper set point of the pressure switch. When this set point is reached, a signal is sent to close the inlet valve and relieve some or all of the pressure in the lubricant separator reservoir. With the inlet valve closed, the compressor is said to be running unloaded (or "off-line" or in the No Load state).

The only air being compressed, at this point, is air that has leaked around the inlet valve plate. With most rotary screw compressor designs, this leakage is calibrated so that an amount of air is always being delivered to the air/lubricant reservoir to maintain enough positive pressure to ensure proper lubricant circulation. Other compressor designs have a pump used to circulate the lubricant and do not rely on air pressure in the air/lubricant reservoir to circulate the oil.

When the system pressure drops to the lower pressure set point, a signal is sent to reopen the inlet valve and the compressor is again running fully loaded (or on-line, or "in the Loaded or Full Load state).

Load / Unload control can adequately satisfy system demands, if applied to a system designed with the correct amount of compressed air storage capacity. Since the system's air demand continues even during times when the compressor is unloaded, storage volume is needed. Applying a compressor with this type of control to a system that does not have the correct amount of storage can result short duration unload time followed by a short time period with the compressor loaded. These rapid changes from load to unload and back is an operating condition referred to as "Short Cycling".

| Short Cycling (Load / Unload Control) |
|---|
| An operating condition where-in an air compressor with load / unload capacity control rapidly switches from a loaded state (delivering full rated air flow) to an unloaded state (delivering zero air flow) and back. During short cycling the compressor control will rapidly change state, usually several times per minute. |

When a compressor's control is short cycling can result in unacceptable fluctuations of system pressure, and lower than expected equipment life. In the case of lubricant injected type air compressor short cycling can also result in higher than expected energy consumption.

Lubricant-flooded rotary screw compressors must also have controls designed to limit the amount of lubricant carryover into the downstream air system. When a compressor with load/no load control is operating at full load, several things are happening. The system pressure is rising from the lower pressure set point to the upper set point (usually 0.7 bar higher than the lower set point). As the upper set point is reached, the inlet valve closes and the compressor stops producing air for the system. At this point, the lubricant separator element is saturated, having been subjected to the full flow of the compressed air and lubricant mix, and the lubricant is saturated with air bubbles. When the compressor control closes the inlet valve, it also opens a blowdown valve to relieve some of the pressure in the air/lubricant reservoir. The reservoir must retain some pressure, however, to provide scavenging capability as the separator element drains and to prevent foaming as the air in the lubricant expands. Without a pressure differential between the scavenge tube in the separator element and the return point on the compressor, lubricant could not be removed and this remaining lubricant would blow downstream when the compressor reloaded.

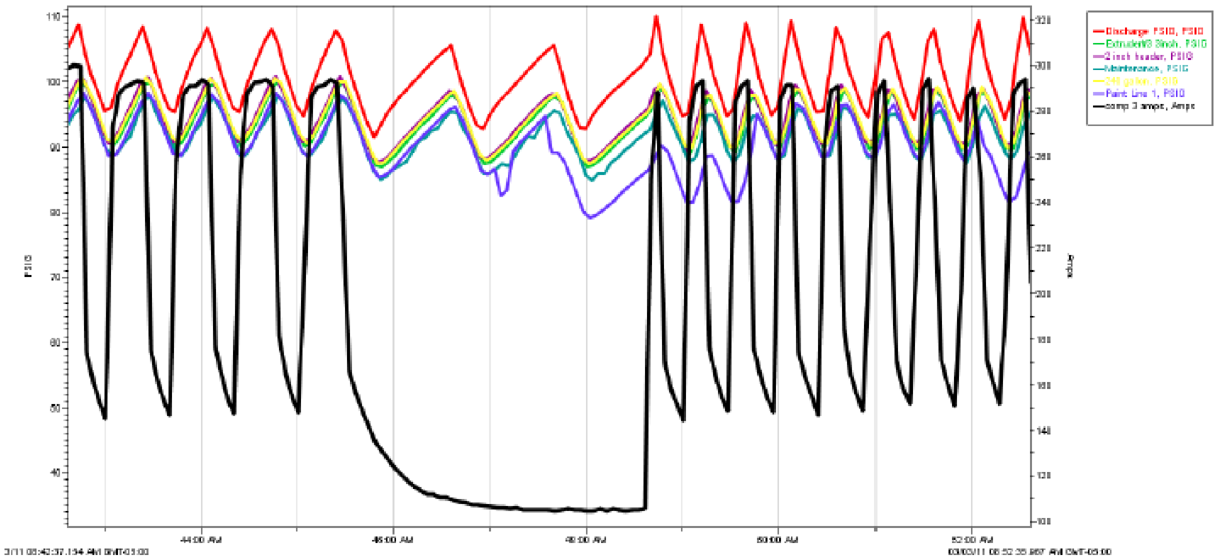
A small kilowatt compressor (less than 22 kW) may be able to relieve reservoir pressure in about 15 seconds. The reservoir does not maintain full pressure during this drain period, but uses a blowdown valve with properly sized orifice to maintain at least some pressure near the end of the cycle.

Since reservoir pressure is not instantaneously relieved, kilowatts do not immediately fall to the unloaded level when the inlet valve closes. Tests show that the kilowatt requirements typically fall to about 80% of the full load level when the inlet valve closes, and then fall steadily to the unload power requirement while the compressor is reducing the reservoir pressure. The actual average kilowatts required for a load/no load compressor operating at less than full capacity is a function of the following factors:

- Initial full load kW requirement
- Final full load kW requirement

- Initial unload kW requirement
- Reservoir blowdown time requirement
- Final unload kW requirement
- Load/unload cycle time



Courtesy of Draw Professional Services

**Figure 3-14: Load / Unload Cycles w/ Amperage**

To reach the optimum energy use with this type of control, the receiver (compressed air storage tank) must be large enough to make the blowdown time an insignificant portion of the total unloaded time. The normal load cycles in Figure 3-14 show amperage dropping to 140 amps. Only after another compressor was started to operate as the trim compressor did the Comp 3 unload cycle last long enough to reach the fully unloaded amperage of approximately 108 amps.

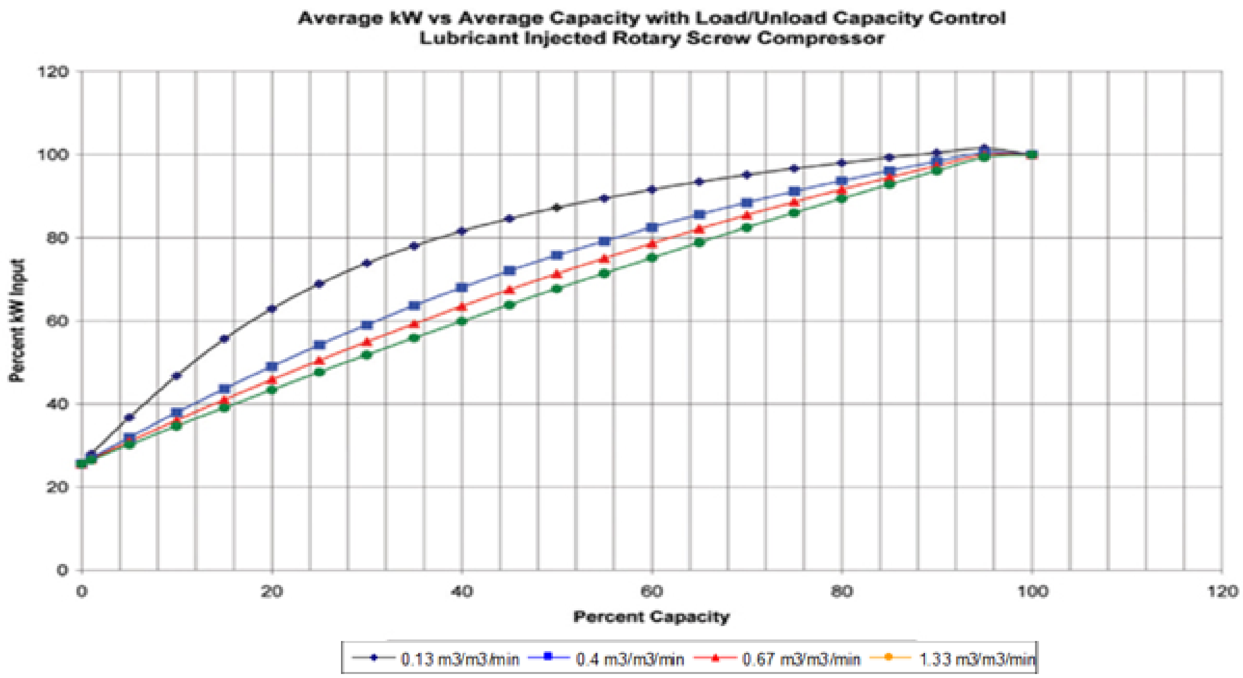



**Figure 3-15: Load / Unload power curve**

The impact of short cycling Load / Unload compressor control on the part load power curve is shown in Figure 3-15 above. As the compressed air system storage volume increases, the part load power performance is improved. The curve shown above is taken from Compressed Air Challenge® and is based on a specific circumstance with a compressor having 40 seconds of sump blow down time and a control pressure differential of 0.7 bar(g). Any change in these operating parameters will change the curves shown above.

In multiple machine applications, the number of machines is limited by the maximum acceptable system pressure variation. Each load/no load machine has to be set with upper and lower pressure points at least 0.14 bar different from the next closest compressor. Staggered in such a manner, compressors added to a system designed originally to operate between 6.9 and 7.5 bar would either have to waste kilowatts to compress to higher than required pressures, or operate at a lower than desired operating pressure.

### c. Modulation Control:

Load/No Load controls rely on a constant swing in discharge air pressure of about 0.7 bar. This constant fluctuation is undesirable in most applications because air device efficiency changes between 1% and 1.4% for each 0.07 bar of supply pressure change. In applications that have small storage capacities, load/no load controls produce rapid pressure fluctuations and excessive inlet valve wear. Modulation control addresses both of those issues by providing a constant system pressure with minimal valve movement at any given system demand. For modulating machines, increasing the size of the receiver does not reduce the power consumption by the same factors as with load/unload controls.

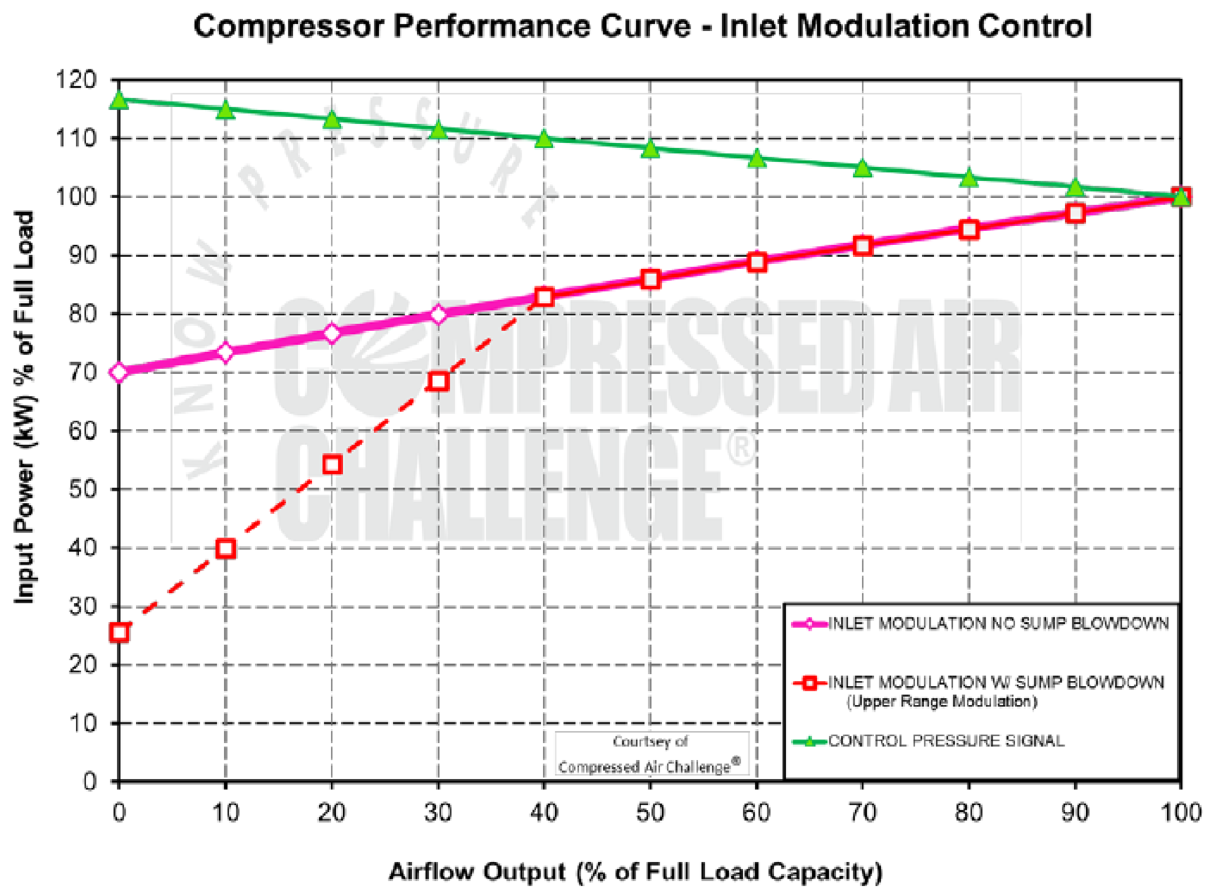



**Figure 3-16: Modulation power curve for rotary screw compressors**

Modulating controls usually use a 0.7 or 1 bar pressure band to determine the compressor response to the system demand. Setting this type of control for a full load operating pressure of 6.9 bar means that the inlet valve is completely open at all pressures below 6.9 bar. A rise in system air pressure above this setting indicates that the system is no longer using the full capacity of the compressor and the excess capacity is causing the pressure rise. As soon as the pressure rises above the full load setting, a signal is sent (pneumatically or electrically) to start closing the inlet valve to reduce the compressor capacity. Airends for modulating machines, like airends for load/no load machines, have a fixed displacement. The only way to modulate the capacity of a fixed displacement compressor is to reduce the absolute suction pressure between the inlet valve and the rotors by restricting the inlet flow. As the gas laws dictate, reducing the pressure by 10% reduces the mass in a fixed volume (the fixed displacement airend) by 10%. This process is seamless over the modulating range of the compressor. By the time the system pressure has risen to the upper limit of the control, the inlet valve is completely closed.

The power required to compress a gas is a function of the mass and compression ratio. Increasing system pressure triggers the inlet valve to reduce the inlet flow, resulting in reduced suction pressure. As the discharge pressure rises, suction pressure drops and the number of compression ratios increases. Because the number of compression ratios increases as the mass flow drops, part load operation of modulating compressors requires a significant percentage of their full load power requirement.

Modulating control mode is a very inefficient way to operate a partly loaded compressor. As can be seen in Figure 4 - 3.17 at 40% loading a compressor might consume over 80% of its full load capacity, more than twice its full load specific power. At 20% flow specific power increases to almost 4 times the full load rating. For this reason, most modulation compressors will switch to the more efficient load/unload mode below 40% of rated flow.

Modulating compressors running in multiple machine applications are subject to the same limits on numbers of machines as load/no load compressors. All modulating machines in a multiple machine application may be running at less than full load at the same time. This provides a very steady plant air pressure, but uses power very inefficiently.

### d. Rotor Length Control (Variable Displacement):

Rotor length control was developed to allow a compressor to match its output to the system demand without the penalty of increasing compression ratios. By controlling the effective length of the rotor compression area, inlet pressure can remain steady and compression ratios fairly constant over the upper 50% of the compressor's capacity. This method of reducing mass flow without increasing compression ratios provides a distinct power advantage when operating at part load. Several methods of controlling the effective length of rotors are currently in production. Although all types offer better efficiencies at some point in their part load operating range than modulating controls or load/no load controls, the design and manufacture of each type of rotor length control has both operating and efficiency differences.

The turn valve and the spiral valve are of essentially the same design. The companies that manufacture these designs use different control methods, but the mechanics of controlling the effective length of the compression area are the same with both valves. Both designs incorporate a number of ports in the low-pressure inlet end of the rotor housing, near where the two rotor bores meet. These ports are as deep as the housing is thick. Below these ports is a cylindrical shaped valve with a spiral cut groove that either seals the port or opens to a cavity that connects to the inlet air passageway. Opening the port even a small amount prevents compression from beginning until the rotor tips pass the partition in the rotor bore casting that separates the ports. This effectively reduces the trapped volume of air to be compressed and reduces the power required.

Closing the ports creates a clearance pocket in the rotor bore. As the tip of the rotor passes over this pocket, some of the air being compressed slips around the tip of the rotor and passes from an area of higher pressure to an area of lower pressure. This hurts the efficiency at load levels above 50%, when compression is taking place in the part of the rotor bore that contains these pockets. Typically, this efficiency loss is about four percent, according to one of the co-inventers and verified by test results. A compressor with a turn or spiral valve will either use more power to make the same air as the identical compressor without pockets or it will produce less air at the same power.

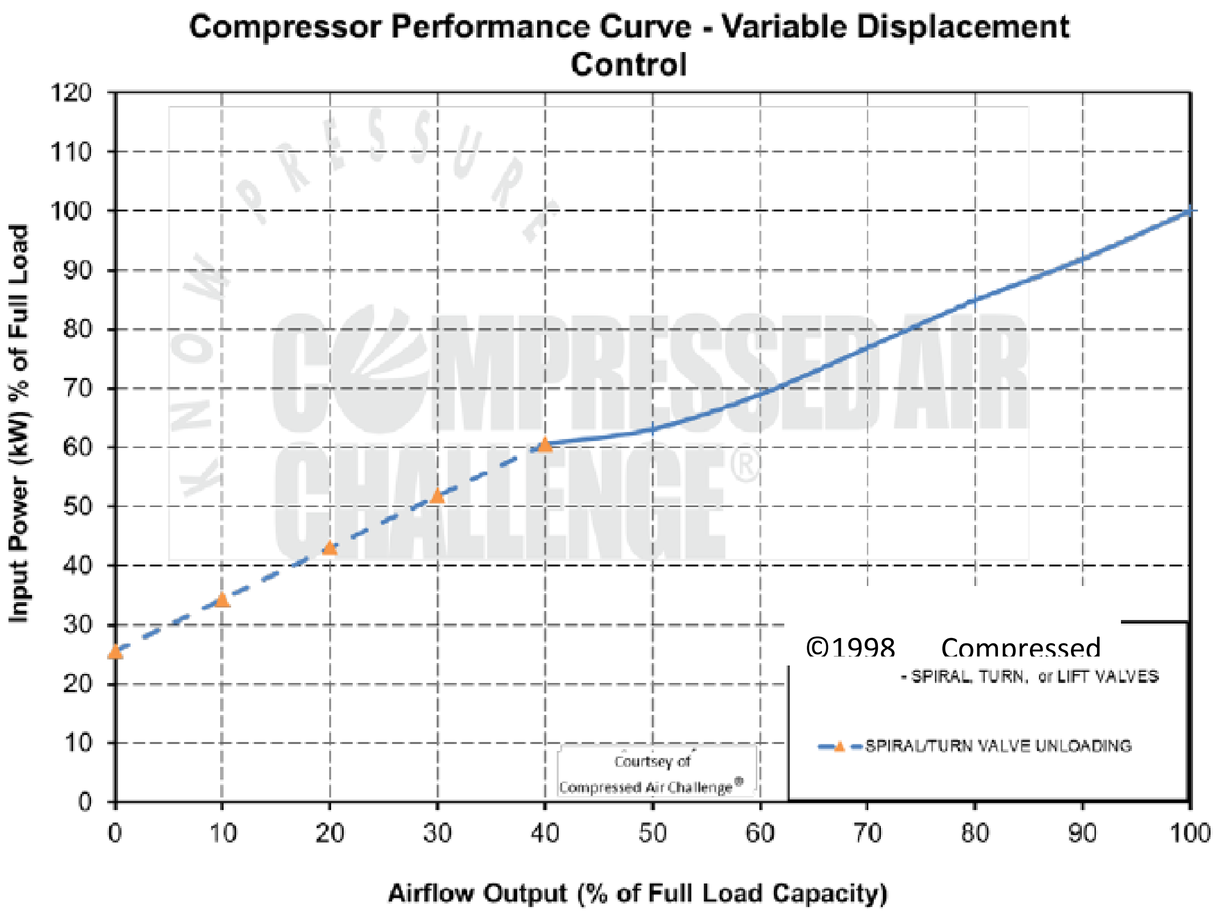



**Figure 3-17: Variable displacement power curve for rotary screw compressors**

## e. Variable Speed Control:

Variable speed drives are available for many rotary screw compressors. This variable speed is usually achieved by using a variable frequency drive or a switched reluctance drive. Variable speed machines are able to match the air supply with the air demand by changing the speed of the airend. With little variation in discharge pressure and no variation in inlet pressure, this type of control is very efficient, when properly designed and packaged.

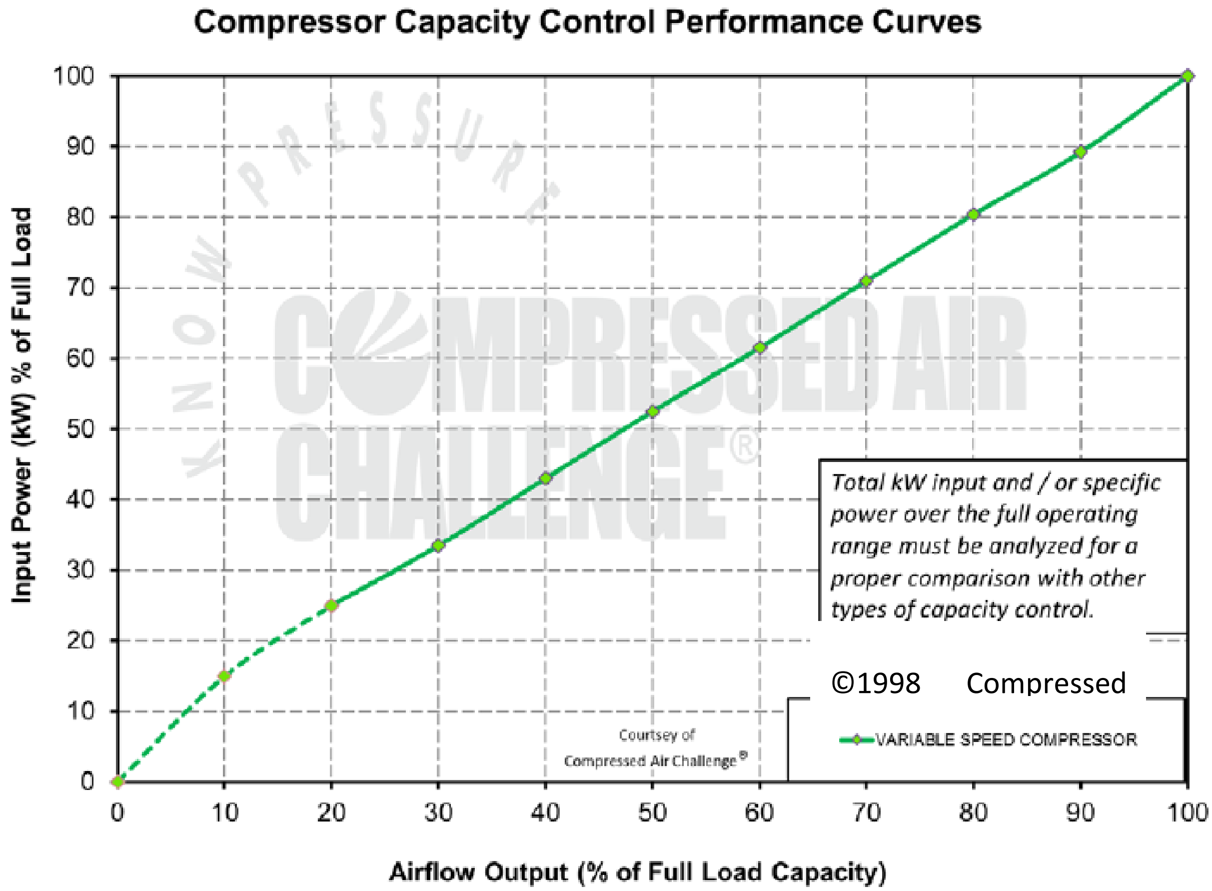



**Figure 3-18: Variable speed drive power curve for rotary screw compressors**

Variable speed drives work by converting AC electrical current to DC current, and then supplying a pulsed DC current that simulates AC current at different frequencies. There are some inherent losses with this process, so care must be taken when specifying a variable speed drive compressor. This type of control works best in single compressor applications with a wide range of loads and little time at full load. In multiple compressor applications, the variable speed control should be limited to the trim compressor only. It must be sized such that its effective control range exceeds the full capacity of the base load machines in the system. This type of control has the best part load efficiency of any other control type. At continuous full load operation, however, the electrical losses through the drive make it a less attractive method of control.

Applying a variable frequency controller to a standard rotary screw compressor may not produce the expected results in energy savings. Rotary screw compressors operate efficiently through a particular range of speeds. Turning the airend too fast OR too slow will hurt the efficiency of the compressor. If an existing compressor is operating near the edge of the efficiency curve, slowing the compressor may actually require more power and not less.

### f. Two-Stage Rotary Screw Capacity Control

When air delivery of a two-stage compressor is less than full capacity, the ideal compression ratio is not achieved resulting in decreased efficiency.

There are two primary electro-pneumatic control schemes for two-stage, flooded screws on the market today. First is simple inlet modulation. An inlet valve on the first stage responds to increasing system pressure by restricting the inlet flow. Second is variable displacement (spiral valve). This method also controls only the first stage flow. Neither method addresses second stage capacity.

Variable speed drive capacity control is also used. Another control method combines electro-pneumatic variable displacement and variable speed drive control.

### g. Electro-Pneumatic Control

At full load, both types of compressors are operating with inlet valves wide open and have near ambient pressure entering the rotor housing. As system pressure rises, the modulating control will start to close the inlet valve, resulting in decreased pressure at the rotor face. If the system requires 75% of the compressor's capacity, the inlet valve will restrict flow to the point that the absolute pressure at the rotor face is 75% of ambient. At sea level, that would mean that the pressure at the inlet to the rotors would be 0.75 bar instead of 1 bar. The first stage would still compress this air 2.92 times and would discharge the air at 2.19 bar. Assuming that the system pressure had risen slightly in order to signal the machine to modulate, the second stage would now be compressing from 2.19 bar to 8.65 bar. This would result in a second stage compression ratio of 3.95. The total compression ratio is the sum of the two stages, 2.92 + 3.95, or 6.87. This increase in total compression ratio offsets the decrease in mass flow and makes part-load performance follow a standard modulating power curve.

There is presently some discussion and controversy with regard to the impact of compression ratio change on total compressor package performance.

Using a variable displacement airend on the first stage improves part-load performance only slightly. The gas laws dictate that the fixed displacement second stage will draw the interstage pressure down to the exact same level as in the modulating example, when running at less than full load. The same number of molecules occupying the same volume will result in the same pressure. This means that the second stage compression ratio will be the same as the modulating machine's second stage. The first stage compression ratio will drop because the interstage pressure has dropped. The inlet valve remains open to the ambient pressure and it only has to compress up to 2.19 bar. In practice, exactly matching the first stage's built-in pressure ratio to the interstage pressure at various load levels does not happen. If the airend was designed to optimize full load performance, it is probably doing some over-compression at part load.

Manufacturers in the United States who participate in CAGI (the Compressed Air and Gas Institute) are presently discussing this very issue. It seems there may be agreement to adopt a test method to measure and report performance of two stage electro-pneumatically controlled compressors using a measurement of specific power. If adopted, the data will likely be presented in a format similar to the method used by CAGI Members to report VSD compressor performance. Until then the discussion of variable displacement part load capacity control performance will likely continue.

### h. Application of VSD compressors and "Control Gap"

When applying a variable speed drive (VSD) compressor running in the same system with other fixed speed compressors some care should be exercised in the system design. Variable speed compressors respond to a pressure signal and speed up or down to maintain a relatively tight pressure band ± 0.10 to 0.15 bar.

As an example, consider a variable speed unit with a maximum speed output of 10.0 $m^3$/minute and a minimum speed output of 2.5 $m^3$/minute. This would make the control range of this unit 7.5 $m^3$/minute. This VSD unit will maintain a ± 0.10 to 0.15 bar pressure band by running up and down in speed within this 7.5 $m^3$/minute control range.

When the system demand falls below 2.5 m3/minute, the unit would stop and start to maintain the system pressure. In this system example there is a low flow control gap between 0 and 2.5 $m^3$/minute, which will cause the unit to stop and start. When the compressor is stopping and starting, the pressure swing could be as wide as 0.3 to 0.6 bar depending on system storage.

On the other hand, if the demand exceeds 10.0 $m^3$/minute the variable speed unit will be at full speed and the pressure will fall as the demand exceeds the compressor's full load capacity. The rate of the pressure drop can be compensated for or eliminated by additional storage in the way of larger air receivers depending on how long the demand exceeds the unit's capacity. If the demand exceeds 10 m3/minute for a long time, additional compressor capacity must be brought on-line.

If the demand goes to 10.1 $m^3$/minute and the next unit is a fixed speed 10.0 $m^3$/minute unit, the fixed speed unit will bring on line 10.0 $m^3$/minute and its corresponding power for an additional 0.1 $m^3$/minute requirement. The 10.0 $m^3$/minute unit will cause the VSD unit to slow down. Since the VSD unit operating at minimum speed still produces

2.5 $m^3$/minute for a 0.1 $m^3$/minute demand, the system pressure will rise, unloading or throttling the 10.0 $m^3$/minute unit. This is a full flow control gap.

To avoid this situation, the control range of the VSD unit should exceed the full flow of the next base load unit.

Put in the context of the next base load fixed speed compressor's capacity, it's full load capacity must be less than the control range of the VSD compressor. In this example the 10 m3/minute VSD compressor has a control range of 7.5 m3/minute. That is the difference between maximum flow of 10 m3/minute and minimum flow of 2.5 m3/minute (10 – 2.5 = 7.5).

Since the VSD compressor has a control range of 7.5 m3/minute, instead of using a compressor with the same (10 m3/minute) full load capacity as the VSD, a 7.0 m3/minute fixed speed unit would be a good choice to run with this 10.0 m3/minute variable speed compressor.

When the demand exceeds the capacity of the VSD, the fixed speed compressor would start and run fully loaded. If the demand were 10.1 m3/minute, the fixed speed would contribute 7.0 m3/minute and the VSD would slow to contribute 3.1 m3/minute. Since the demand exceeds the full capacity of the fixed speed unit and is greater than the supply of the VSD at minimum speed, the VSD is able to maintain a stable pressure and the compressors do not load and unload. If the demand exceeded 17 m3/minute, another 7 m3/minute fixed speed compressor could be used. This would provide a seamless transition from 2.5 m3/minute all the way up to 24 m3/minute.

As all examples are theoretical, it is highly recommended that a flow profile be done to determine actual requirements based on real time and real site conditions. This will allow the user a more exact understanding of this system and possibly uncover many energy saving opportunities while actually improving overall system performance.

## 3.6 Rotary Vane Compressors

Rotary vane compressors are less common than reciprocating and screw compressors, but popularity of these types are increasing. Rotary vane compressors consist of a cylindrical rotor offset in a casing. The vanes are inserted radially into slots machined into the cylindrical rotor.

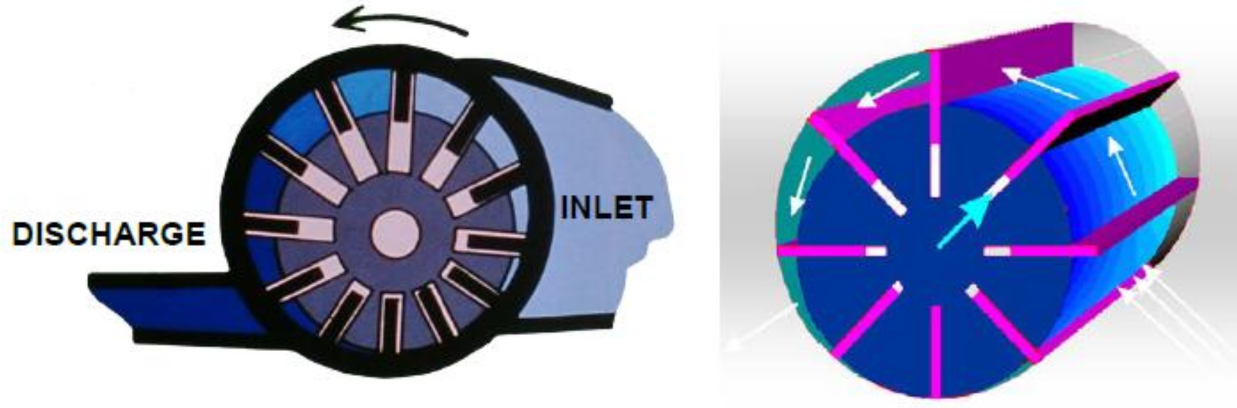



**Figure 3-19: Rotary vane cylindrical rotor offset**

As the rotor turns, the vanes are pressed against the housing by centrifugal force or by springs. The rotor and the housing do not have the same centerline. The inlet port is located in an area where the volume of the compression chamber is increasing. Once the rotor passes the inlet port, a volume of air is trapped. Continued rotation of the rotor squeezes the trapped air into a smaller volume until the desired compression has been reached. At that point, the rotor passes a discharge port in the housing and the air is pushed out.

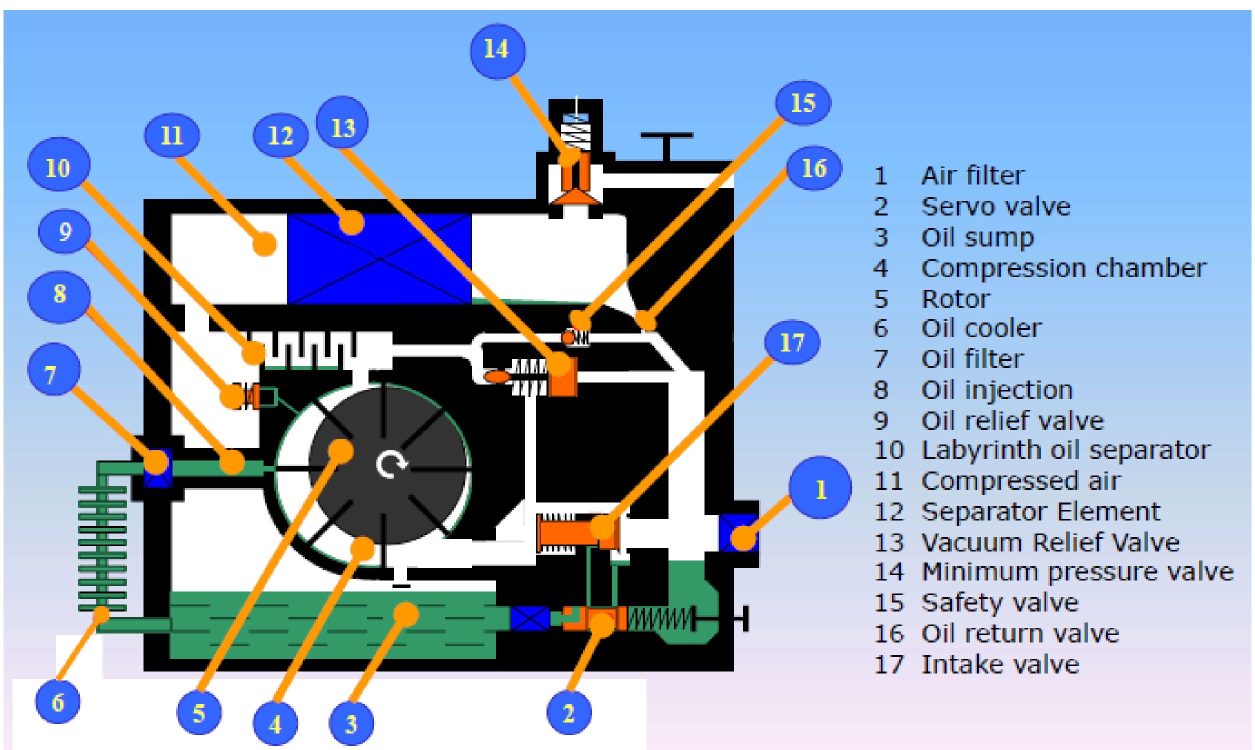



**Figure 3-20: Rotary vane compressor internal operation**

Traditionally, frequent overhaul requirements due to vane wear made this type of compressor best suited for light duty with limited operating hours. Recent technological advances and the availability of superior quality oils have, however, made rotary vane compressors more competitive in comparison with rotary screw compressors.

Rotary vane compressors are very efficient in smaller sizes of up to 15kW due to rotary screw compressors losing volumetric efficiency at slow speeds where the male and female rotors interlock, referred to as the "dead-band" area. Rotary vanes are also more adept to VSD's due to screw compressors having some volumetric or "blow-back" losses at low rotational speeds, which is not the case with the rotary vanes. Rotary vanes compare favorably against rotary screw compressors up to 90 kW, whereafter their efficiency diminishes. Rotary vanes have relatively lower maintenance costs due to slower rotational speeds, with bearing lasting up to 100,000 operating hours, and major services required at 10,000 operating hours.

The auxiliary components of the vane and screw machines are similar. They both use an oil tank, separator, oil cooler, thermal bypass, minimum pressure valve, high temperature switch, and some means of controlling capacity. The maintenance of these items i.e. oil, filter and separator changes, would be similar. However, if repair is needed on the compressor alone then the screw compressor is more difficult and costly to own. In most cases air end repairs are handled on a factory exchange basis. It is important for the vanes to be length adjusted. All the vanes of one set must be replaced at the same time.

## 3.7 Dynamic Compressors

Dynamic compressors are divided into two groups, centrifugal (or turbo) and axial. Both types compress air by accelerating the inlet air stream and then converting the velocity to pressure. Centrifugal compressors are the most common dynamic compressor for general industrial applications.

## 3.8 Centrifugal Compressors

Centrifugal compressors have one or more sets of impellers that accelerate the inlet air stream. The impellers generate approximately 50% of the pressure. The remaining pressure is generated when the diffuser and volute slow the air stream velocity. This is the same principle used by turbochargers on engines. Engines use exhaust gases to turn the impeller and centrifugal compressors generally use an electric motor and gear sets as a power source for the impeller. Impellers in these compressors turn at very high speeds, up to 50,000 revolutions per minute. Using velocity to build pressure requires several stages of compression to reach normal plant operating pressures. Two and three stage centrifugal compressors are common. No lubricant is injected during compression, so intercooling between stages is possible.

Centrifugal compressors are available in capacities that range from about 15 $m^3$/min to 3000 $m^3$/min or more. From about 45 to 50 $m^3$/min and larger, these types of compressors have a slight efficiency advantage over a very large rotary screw compressor. Operating costs for centrifugal compressors range from about 5.7 to 7.1 kW/$m^3$/min. This advantage, however, is only valid if the compressor is running as a base load compressor. Dynamic compressors must have an airflow close to their design point in order to function properly. They do not perform efficiently in applications with fluctuating demands.

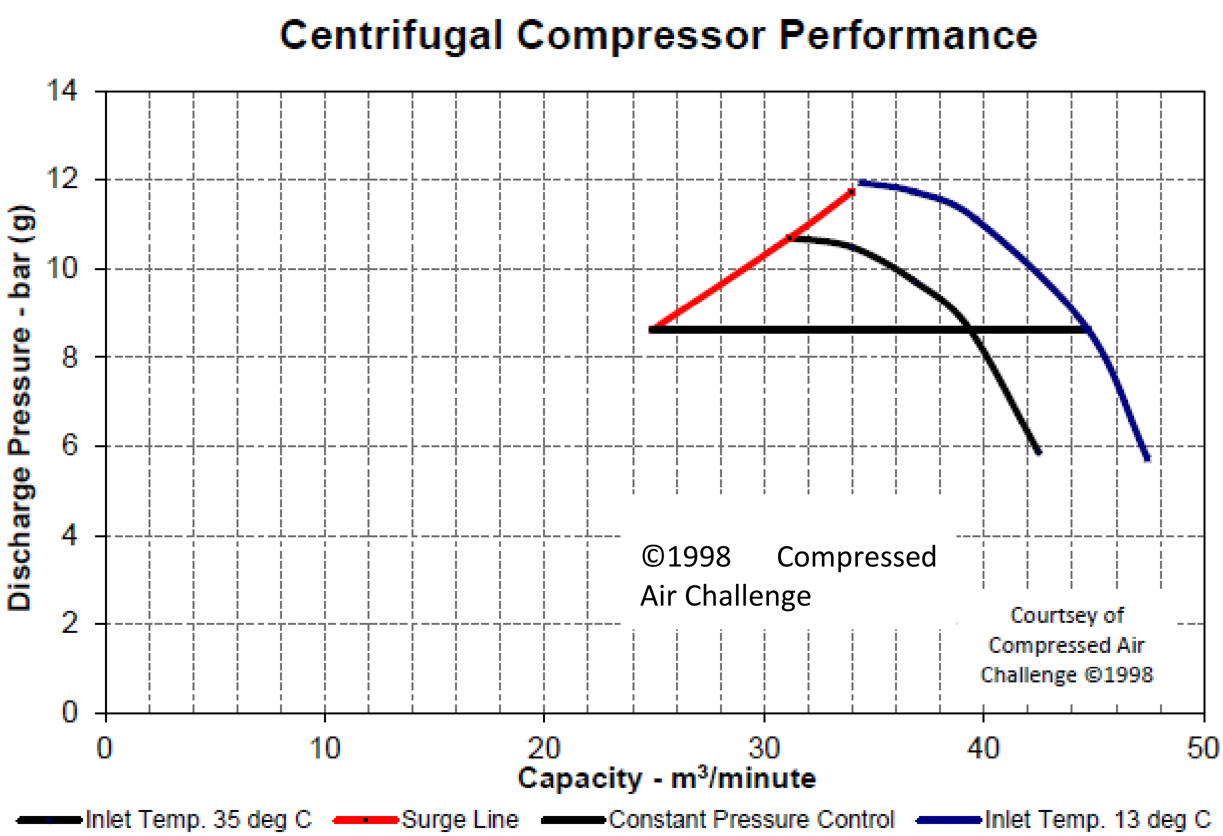



**Figure 3-21: Sample centrifugal compressor performance curve**

## 3.9 Axial Compressors

Axial compressors operate on a principle similar to a jet engine. The compressor consists of a rotor with multiple rows of rotating blades inside a housing with multiple rows of stationary blades. The air is compressed axially, down the rotor plane. Axial compressors are generally used in very high volume applications such as petro-chemical plants. Capacities of axial compressors can be as high as 30,000 $m^3$/min.

## 3.10 Centrifugal Compressor Performance

Centrifugal compressors are the most common dynamic compressor used in industrial compressed air systems. They are provided as factory packaged units that are relatively easy to install. In larger size centrifugal compressors both first cost and compression efficiency become more attractive, particularly in the larger capacity range; typically, from 400 kW & 50 $m^3$/minute (500 HP & 2,000 cfm) and larger. Capacities of to 400 to 560 $m^3$/minute (15,000 cfm to 20,000 cfm) and even larger are available. However, as the compressor output get larger, machines are less likely to be factory packaged.

Centrifugal compressors can generate air with low specific power operating very efficiently while with-in their turndown capacity range. Below the turndown range many centrifugal compressors blow-off excess capacity to atmosphere, a very inefficient way to control the compressor.

In recent years the available capacity range of centrifugal compressors has gotten smaller. Centrifugal compressors can now be found as small as 75 kW and 100 HP (450 cfm). The capacity range of large rotary screw compressors and small centrifugal machines is beginning to overlap as rotary screw compressors are getting larger, and centrifugal air compressors are getting smaller.

More combined systems with a mix of positive displacement, and centrifugal or dynamic machines are being installed. Mixing positive displacement and dynamic compressors in a system presents challenges in maintaining efficient and reliable control. Fundamentally due to the nature of dynamic compression, control of centrifugal compressors achieves constant pressure. While positive displacement compressors use an upper and lower pressure limit to load and unload the compressor's capacity. Undesirable interaction of the positive displacement pressure control band with constant pressure controls of a centrifugal compressor can decrease overall system efficiency.

## 3.11 Centrifugal Compressor Drivers

A wide range of drivers are used for centrifugal compressors. The most common driver is an electric motor. Smaller packages typically use 230 or 460 Volt three phase power. Higher horsepower motors are very commonly medium voltage units operating on 2,300 or 4,160 Volt power. Beginning at 400 kW (500 HP) optional synchronous motors with 1.0 and / or leading power factor are available which may be helpful in maintaining or improving the overall plant power factor.

Other drivers used include engine drive, both natural gas and diesel. Very large compressors could use gas turbine drive. Steam turbine drives are used more often. Alternative drives may be considered for many reasons.

Steam turbine drives can in certain situations be very attractive if excess low cost steam is available. It is possible that summer time periods result in excess steam used for space heating during the winter months. If seasonal rates for electricity are high during summer peak periods it may be economically justified to operate steam driven compressor. A decision to employ this operating strategy would require thorough economic study as compressor capacity with an alternative drive would have to be available for the winter months.

Alternative drives might be attractive if electrical curtailment might interrupt production. In some areas “interruptible power” can be purchased. Interruptible power is electrical energy used subject to an agreement that the utility can notify the plant and within a specified time period (10 or 20 minutes) the energy use will be eliminated. One way this could be done is by shutting down an electrically driven compressor by switching over to an air compressor which is not electrically driven.

## 3.12 Multi-Stage Centrifugal Compressors

Centrifugal compressors are dynamic compressors. That is to say the air entering the eye (center) of the impeller is accelerated through the impeller and exits the tip at high velocity and at only slightly higher pressure than when it entered the impeller. The high velocity air then enters the diffuser section and the volute of the stage where the air velocity is slowed, and velocity energy is converted to pressure energy.

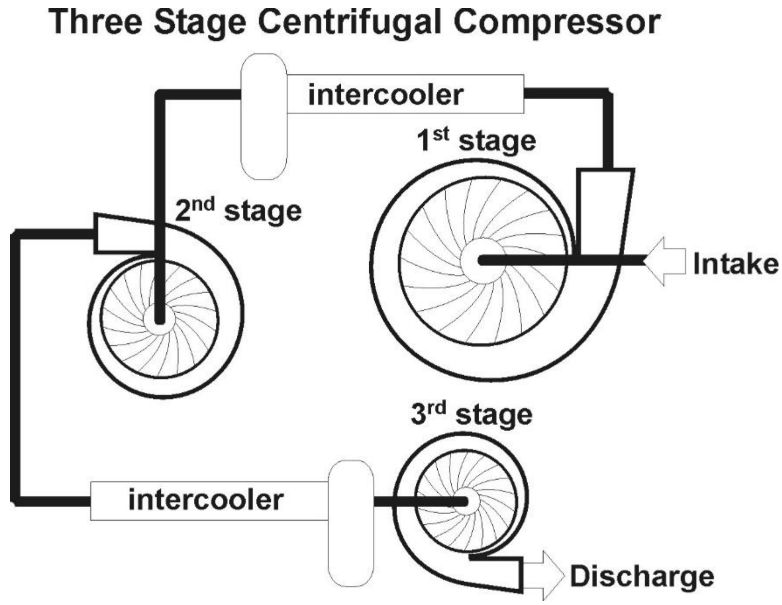



**Figure 3-22: Three Stage Centrifugal Compressor w/ intercoolers**

After exiting the stage, the air passes through inter-stage piping, the inter-cooler, inter-stage separator, and into the following stage impeller where the process is repeated. For plant air pressure range 7 to 9 bar (100 to 130 psig) centrifugal compressors typically have three stages of compression. For three stages of compression the compression ratio of each stage is the cubed root of the compressor's total pressure ratio. Some compressor manufacturers offer a less expensive and less efficient two stage designs where each stage compression ratio is the square root of the overall pressure ratio.

Just as positive displacement reciprocating compressors have a given bore and stroke which determine the displacement and performance characteristics, centrifugal compressors have a given geometry which determine performance characteristics. Aerodynamic design and dynamic compression result in a relationship of head (pressure) and flow shown in the diagram below.

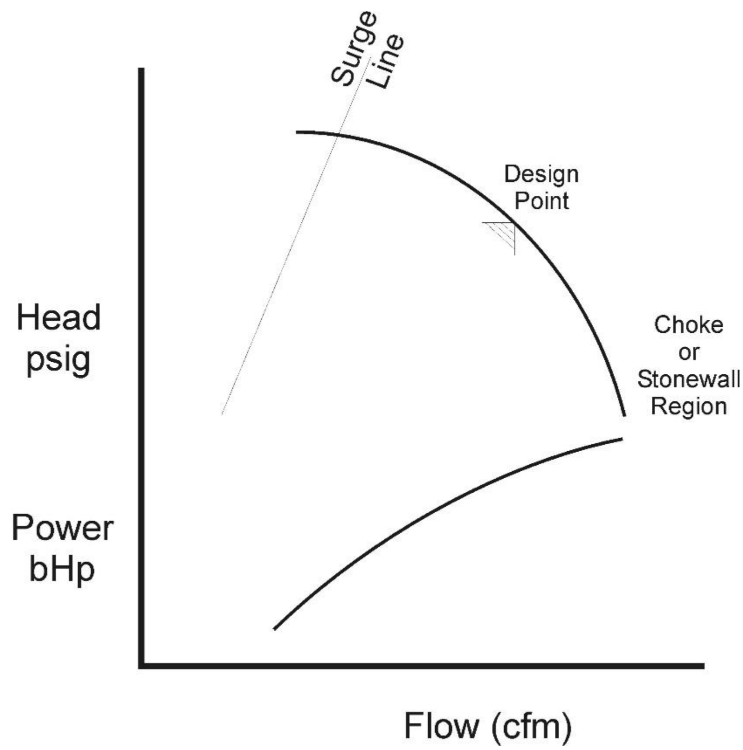



**Figure 3-23: Centrifugal compressor performance curve w/ power**

The particular manufacturer's aerodynamic design will produce the native head / flow performance curve of an air compressor. Some designs result in steeper curves, some in flatter curves. The amount of pressure rise allowable before reaching the surge point will vary based on aerodynamic design. If the compressor is operated without any inlet control, it will operate at points along the curve. As the system air demand decreases pressure will rise, flow and power will decrease.

The relationship of power and pressure for dynamic compressors is opposite that of positive displacement machines. For positive displacement machines, power increases by 6% per bar (1% for every 2 psig) increase in discharge pressure. However, for centrifugal compressors, power will decrease as pressure increases. Air delivery (flow) also decreases as pressure increases.

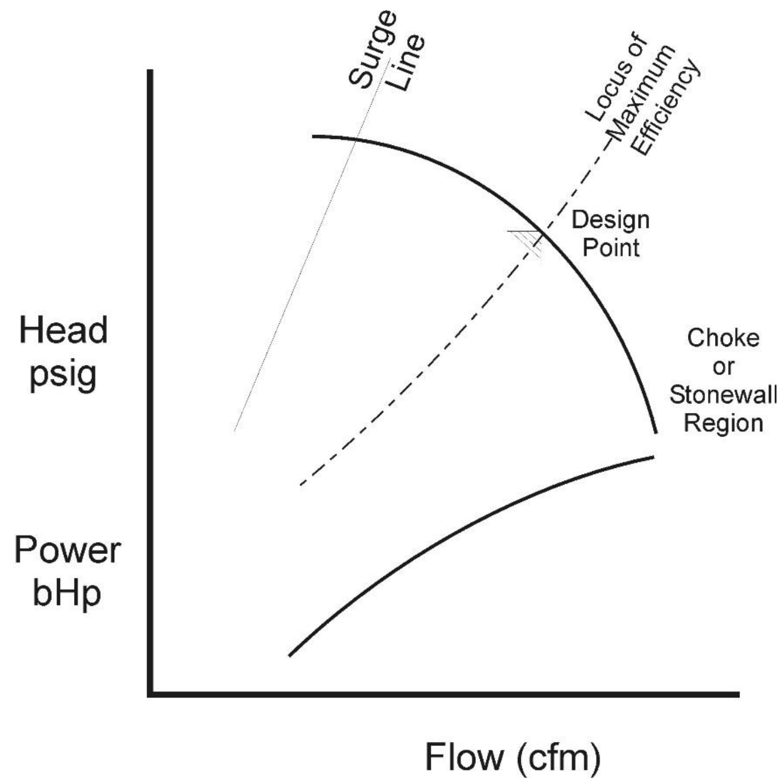



**Figure 3-24: Centrifugal compressor performance curve w/ locus of max efficiency**

There is another characteristic performance line (shown above) related to centrifugal compressor performance. That is the "Locus of Maximum Efficiency". If the centrifugal compressor designer made an efficient aerodynamic selection, the compressor's design point is located near the point where the locus of maximum efficiency intersects the performance curve.

It is interesting to note that as head pressure decreases delivered air flow from the compressor increases. Power also increases. As you move away from the locus of maximum efficiency toward the choke region, the specific power (kW/ $m^3$/min) increases due to declining efficiency.

Moving along the performance curve increasing head will decrease delivered airflow and power. It is necessary to prevent the discharge pressure from rising to a point where surge occurs. Surge takes place as the compressor's discharge flow decreases and the aerodynamic performance reaches a point where the compressor discharge pressure is lower than the system pressure. The point where airflow reversal occurs (air flows from the discharge back toward the inlet) is called surge. Surge causes vibration and mechanical stress on the compressor. If a compressor is operated with frequent surge events, significant mechanical damage to the compressor is likely to occur.

As a safety protection to prevent surge from occurring most centrifugal compressor manufacturers have a blow-off valve at the compressor discharge. As the compressor controls sense conditions when surge may occur, the blow-off valve begins to open and it will discharge excess airflow to atmosphere. By maintaining the compressor's delivered airflow at or above minimum safe flow, surge is prevented from occurring. It is very inefficient to operate centrifugal compressors in blow-off as energy used to compress air being blown to atmosphere is wasted.

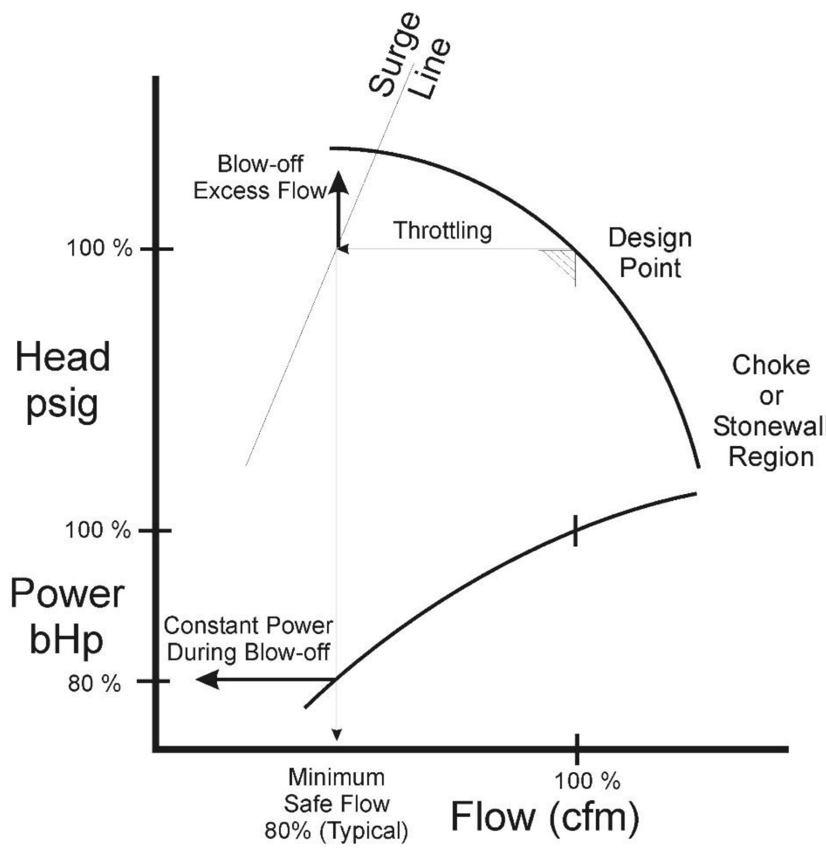



**Figure 3-25: Centrifugal compressor performance curve w/ design point, throttling line and blow-off**

It is usually desirable to control the inlet flow to the compressor reducing it air delivery at a pressure below the point at which surge will occur. This range of control is sometimes referred to as "Throttling" or "Turndown". The inlet of the compressor is fitted with a modulating control valve (throttling valve) or inlet guide vanes (IGV's). As pressure increases, the inlet is restricted creating a partial vacuum and reducing the compressor's air delivery and consumed power.

This control method allows a wider range of turndown and lower minimum safe flow than allowing system pressure to rise along the native aerodynamic performance curve until reaching the surge point. Inlet guide vanes allow a greater degree of turndown (25% or 30% of full capacity) and improved part load power reduction as compared to inlet throttling. Because of the initial cost of IGV's they are usually available as an additional cost option on compressors 375kW and larger.

Even with turndown control there is still a need to blow-off excess air and prevent surge from occurring if the system air demand is less than the maximum turndown range.

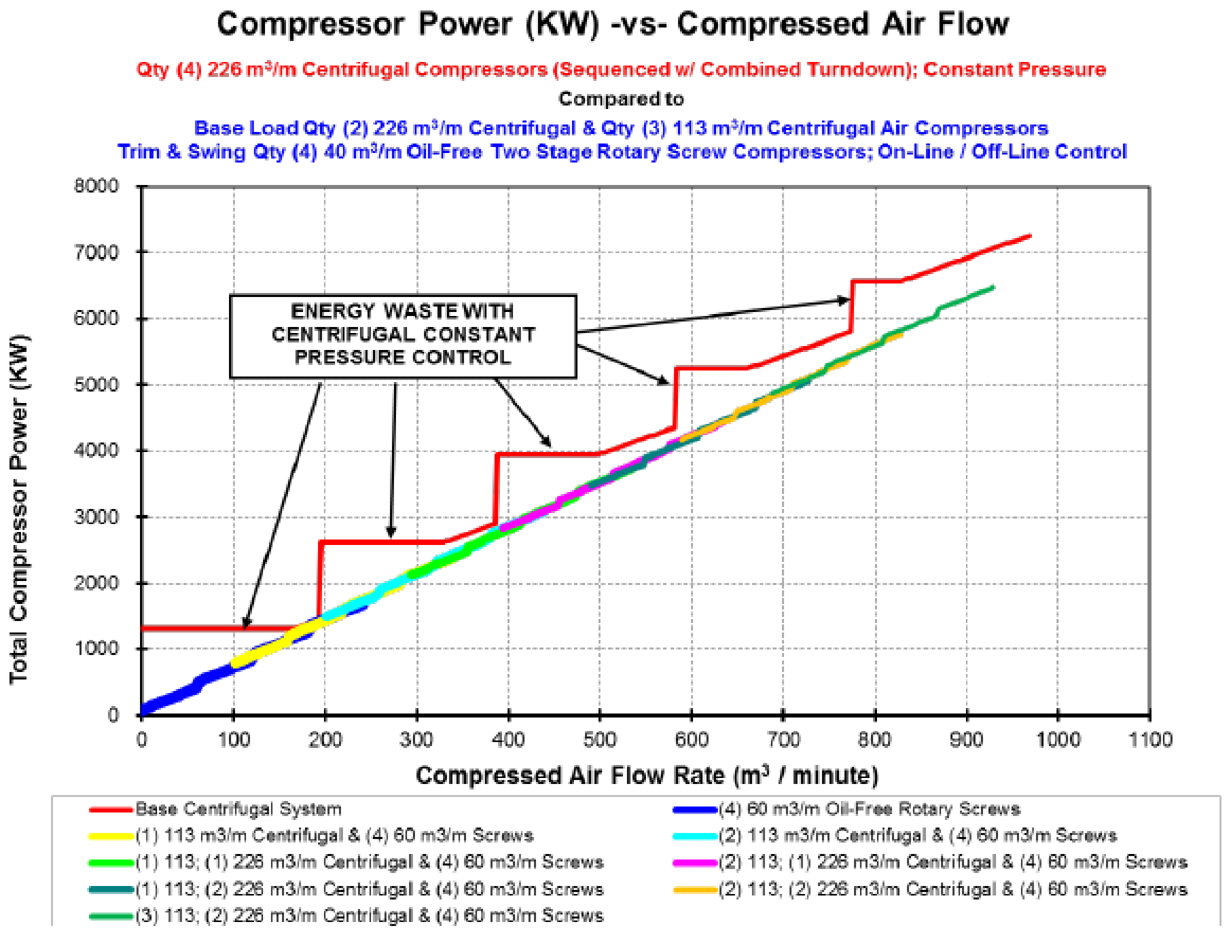



**Figure 3-26: Multiple centrifugal compressor performance -vs- multiple positive displacement rotary screw performance**

When multiple centrifugal compressors are in a single system the amount of turndown can be improved by throttling multiple compressors at the same time. The chart above shows the capacity performance from 0 to 962 $m^3$/min with a total of 4 centrifugal compressors rated at 226 $m^3$/min each. As more compressors are added the control range operating in blow-off is reduced. This strategy requires multiple smaller centrifugal compressors. One of the benefits of centrifugal compressors is their economy of scale for large machines.

The best application for a centrifugal compressor is base load capacity always operating within its throttling range. Any time a centrifugal compressor is blowing off a portion of its capacity, efficiency suffers. Another method to minimize blow-off is available with sophisticated microprocessor control which monitors operation of the compressor and operates the machine closer to the surge point. Many factors affect the actual surge point of a compressor. The surge point of an air compressor changes with; inlet pressure, inlet temperature, cooling water temperature, condition of the machine, coolers, etc. By monitoring many factors affecting the operation of the machine and surge point, the control can use a wider throttling range without causing the compressor to surge. Even with this improved control a centrifugal compressor has to use blow-off through a significant portion of its capacity range.

The chart above also illustrates the power –vs- flow relationship for a mixed system of load / unload positive displacement compressors operating as trim machines with various centrifugal compressors operating as base load.

With proper compressor selection and control the system can operate without any compressed air wasted to blow-off. Connecting positive displacement compressors with dynamic compressors must provide proper control. Centrifugal compressors are designed with constant pressure control, while positive displacement compressors operate over a control pressure band of 0.4 to 0.7 bar 1 bar or more. Consider a system with two compressors a centrifugal, and a rotary screw compressor.

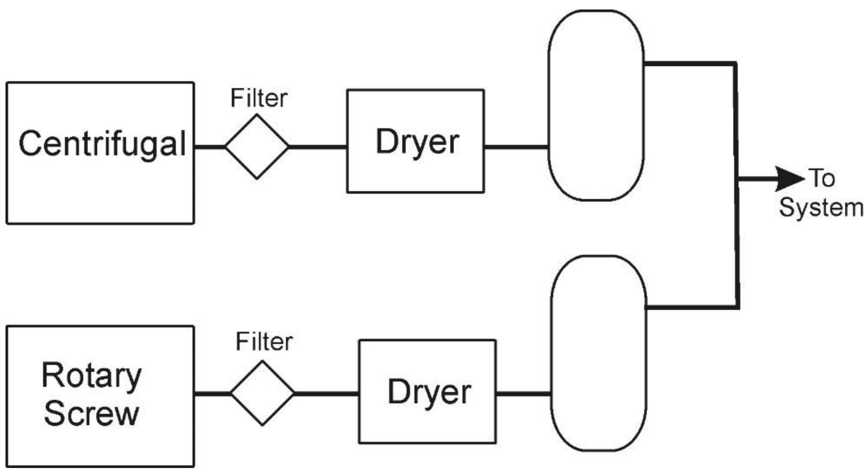



**Figure 3-27: Block Diagram combined system centrifugal and rotary screw compressors**

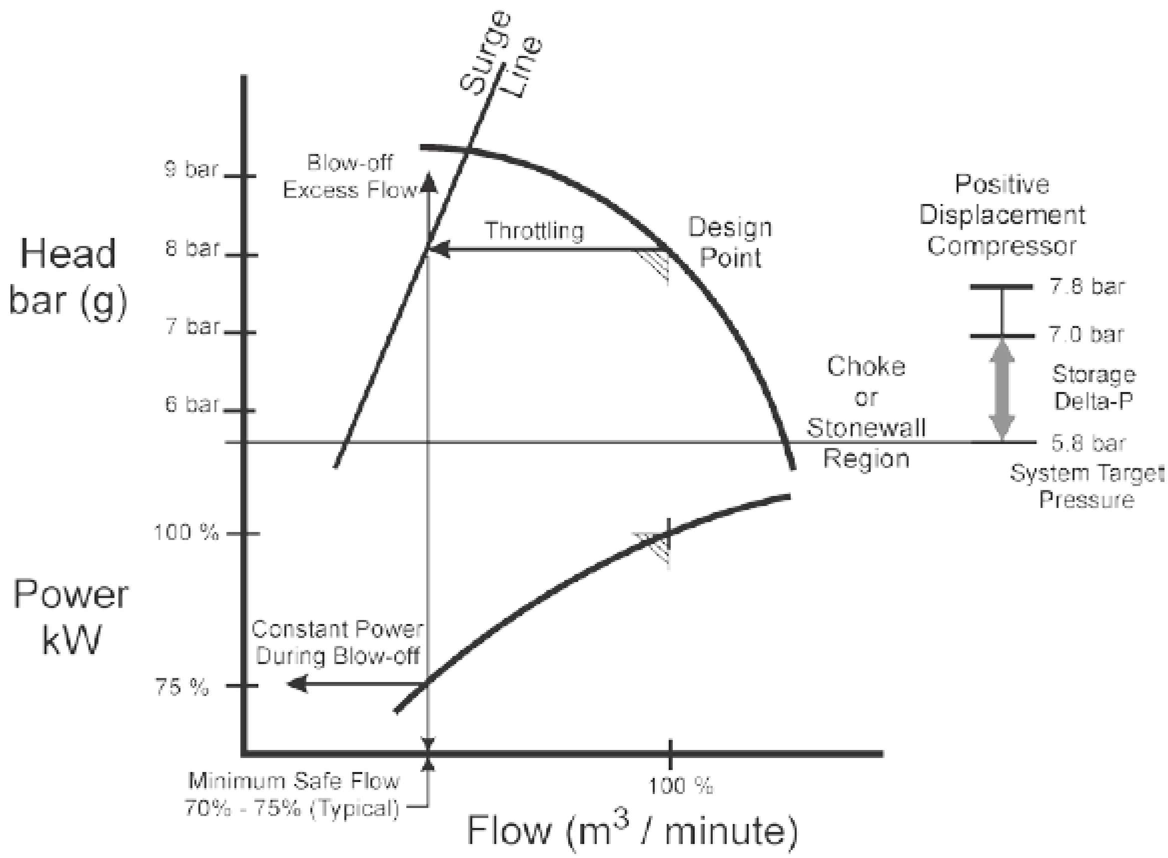



**Figure 3-28: Performance profile combined system centrifugal and rotary screw compressors**

The system design above provides the ability to base load the centrifugal compressor, and trim with the load / unload rotary screw compressor. The throttling pressure of the centrifugal compressor is 8.0 bar with the control pressure band of the rotary screw from 7.0 bar to 7.8 bar. If the lowest optimum pressure for the system is 5.8 bar target pressure, there is a significant component of artificial demand.

Lowering the throttling pressure of the centrifugal compressor can cause extremely inefficient operation. Suppose the throttle setting is reduced to 7.2 bar. As the rotary screw compressor loads up, the pressure will rise. When the pressure reaches the throttling pressure, 7.2 bar the centrifugal will begin to reduce its output. The system pressure cannot rise to 7.8 bar, the unload point of the rotary screw compressor. In fact, if the system air demand continues to decline, the centrifugal compressor will eventually begin to blow-off. At the point the output of the rotary screw compressor is effectively feeding the blow-off of the centrifugal compressor.

Artificial demand could be reduced by lowering the pressure of the rotary screw compressor. There are two issues associated with this. As system pressure falls the centrifugal compressor will operate lower on its curve, away from the locus of maximum efficiency, toward the choke region. Secondly as system pressure is reduced closer to the target pressure there is less storage pressure differential available. If storage is not sufficient to support event demands, system pressure may drop below the target pressure.

One alternative is to install a flow control at the intermediate point between the supply and demand sides of the system. This will allow the 5.8 bar target pressure to be maintained.

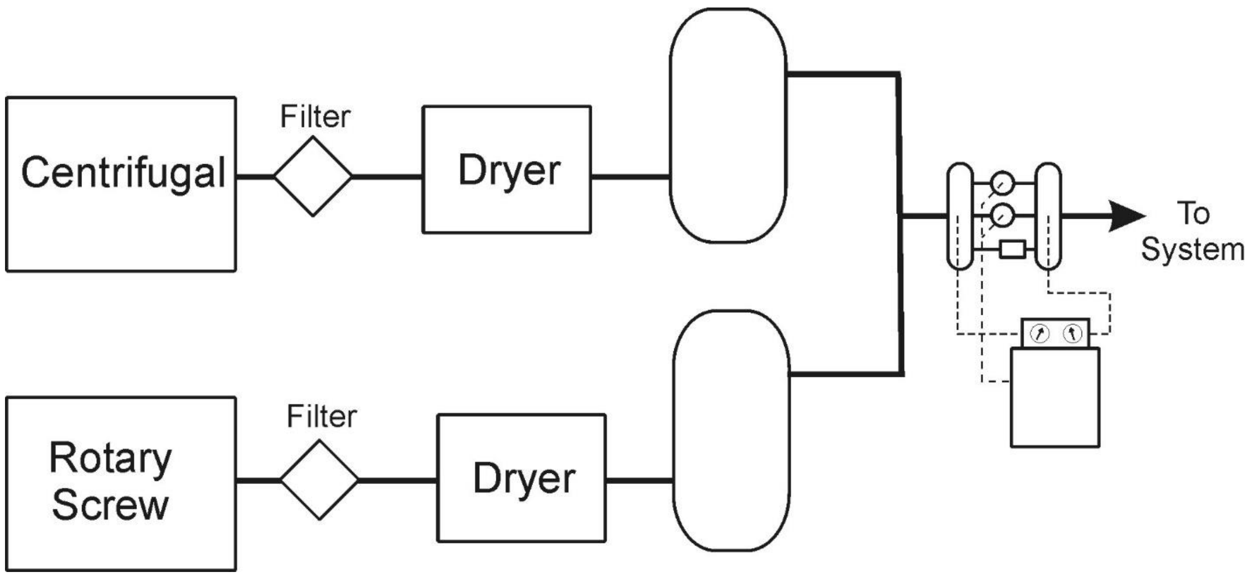



**Figure 3-29: Block Diagram combined system centrifugal and rotary screw compressors w/ pressure flow control**

In the configuration above the system target pressure can be set to 5.8 bar eliminating artificial demand. Higher pressure upstream provides useable air storage. However, the centrifugal compressor's discharge is still subject to the swing of the rotary screw's control pressure band. Also the potential exists to misalign the control pressure and throttling pressure settings where the rotary screw compressor will feed the centrifugal compressor's blow-off.

To prevent the interaction of discharge pressure between the centrifugal and rotary screw compressor, it is necessary to separate the discharge piping connections. By placing the discharge of the centrifugal compressor downstream of the flow control, the centrifugal compressor's discharge will be isolated from the control pressure band of the rotary screw compressor.

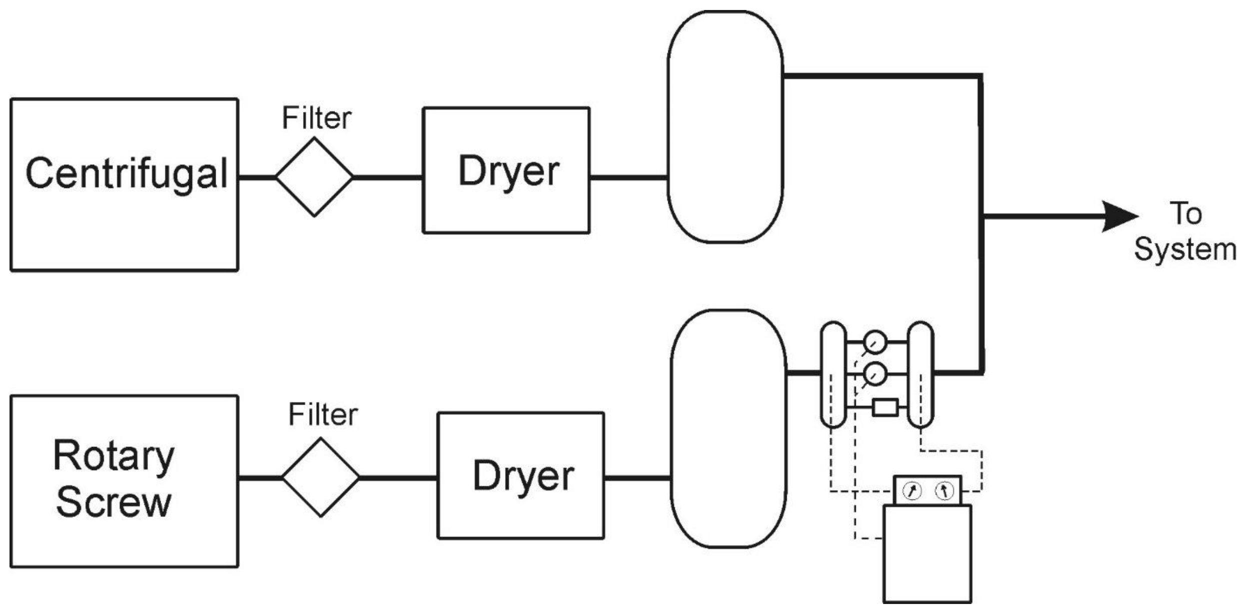



**Figure 3-30: Block Diagram combined system centrifugal and rotary screw compressors w/ pressure flow control for rotary screw**

The approach shown in the block diagram above provides isolation of the centrifugal compressor discharge from the rotary screw control pressure band. However, with 5.8 bar target pressure for the system, the centrifugal compressor will be operating well into the choke region. There are two alternatives to deal with this issue. If the compressor is due for major rebuild, it may be possible to redesign the compressor's aerodynamics to a lower design pressure. Check with the manufacturer to determine if a re-aero is possible. Limitations of the compressor frame, and inter-stage piping / coolers, may prevent lower design pressure.

The second alternative is to apply back pressure control at the discharge of the centrifugal compressor. This allows a set point to be maintained which allows the centrifugal compressor to operate at its design point while discharging air downstream of the intermediate flow control. Isolation of the centrifugal compressor's discharge from the control pressure band of the rotary screw is achieved without the centrifugal operating in the choke region.

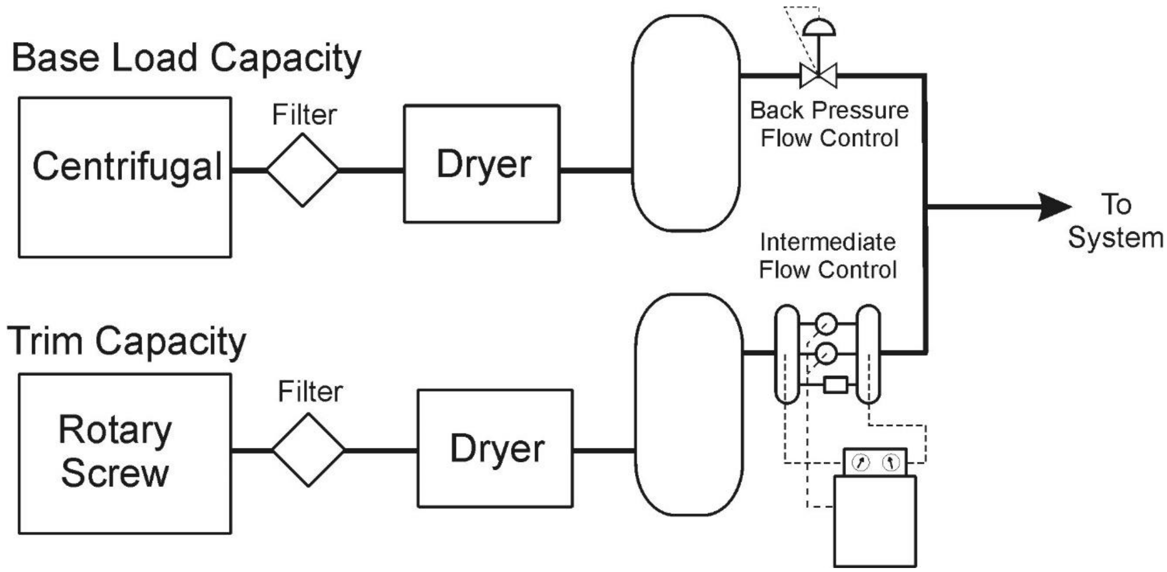



**Figure 3-31: Block Diagram combined system centrifugal and rotary screw compressors w/ pressure flow control for rotary screw and back pressure valve for centrifugal**

When configuring a mixed compressor system as shown above, there are several performance factors to consider. Trim storage must support the permissive start-up time of compressors. In the case of centrifugal compressors start-up may require a minute or longer. Should storage support the unanticipated shutdown of a compressor and start-up of replacement capacity?

The back pressure control valve must have a flow coefficient Cv selection consistent with the centrifugal compressor capacity and the imposed Delta-P while operating in an effective portion of its control range. The speed of response of the back pressure valve must be controlled appropriately to the upstream receiver volume. The back pressure valve can potentially cause rapid upstream pressure changes which may affect the centrifugal compressor's controls.

For systems that have multiple compressors in multiple locations, it can be virtually impossible to adjust compressor pressure set points in a manner which prevents blow-off from occurring. Pressure gradients across the system between compressor locations are constantly changing as the system air flow changes. To account for the normal pressure variations that occur might require operators to adjust pressure control set points several times each day. Flow control valves properly applied at the various compressors provides a practical way control multiple compressors across various locations in the system.

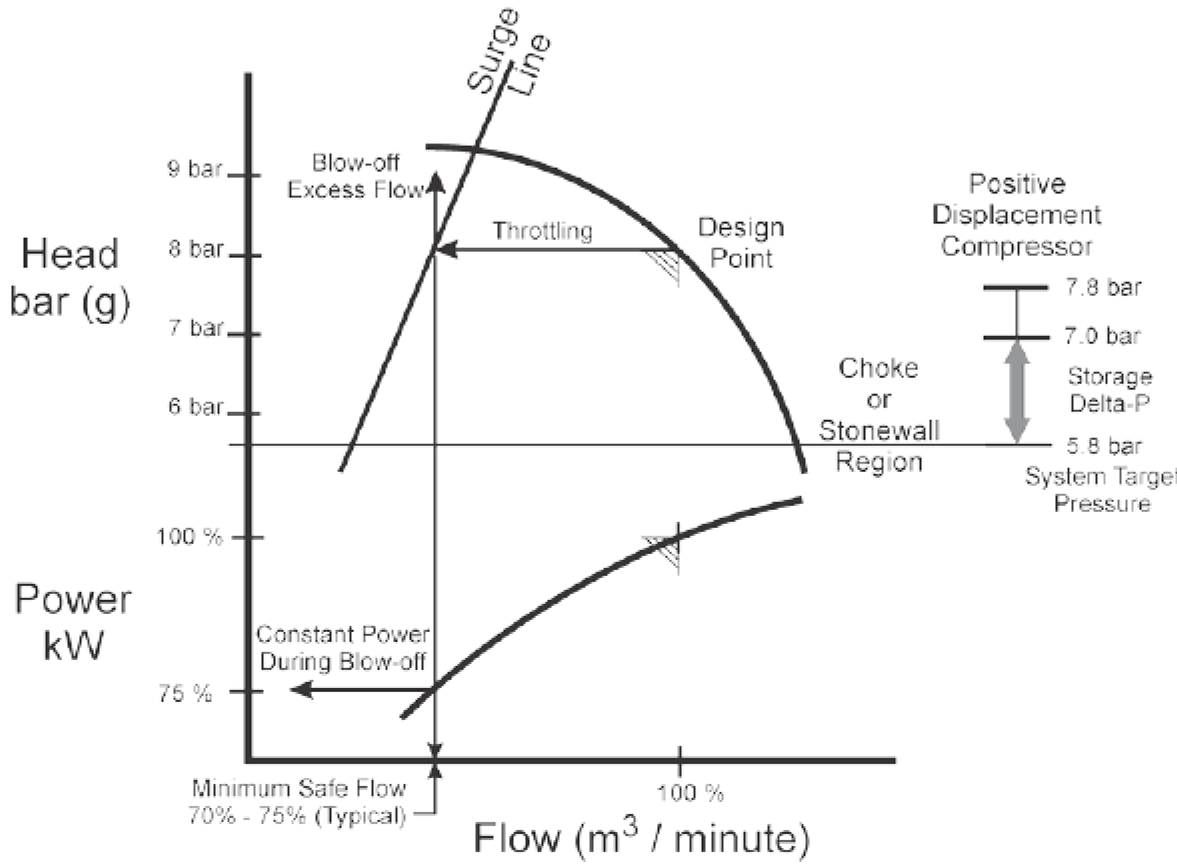



**Figure 3-32: Performance profile combined system centrifugal and rotary screw compressors w/ pressure / flow control eliminating artificial demand.**

Intermediate flow control provides controlled storage differential, and eliminates artificial demand by controlling the system target pressure. Back pressure flow control keeps centrifugal compressors operating on their design point. Back pressure control also prevents the centrifugal compressor from going to blow-off. If pressure begins to rise, the back pressure control opens allowing the centrifugal compressor's air delivery to enter the system. As system pressure then begins to rise the increase is sensed at the intermediate flow control. The intermediate flow control then closes reducing flow entering from the trim capacity section of supply. Trim compressors designed to operate efficiently at part load conditions respond by reducing their air delivery and consuming less power.

Mixed systems operating a combination of centrifugal and positive displacement compressors frequently have significant waste to blow-off and artificial demand that can be eliminated with properly designed control.

## 3.13 Centrifugal Compressor Control Methods

The axial compressor is better suited for constant flow applications, whereas the centrifugal design is more applicable for constant pressure applications. This is because the characteristic curve of the axial design is steep, and that of the centrifugal design is flat. The characteristic curve of a compressor plots its discharge pressure as a function of flow, and the load curve relates the system pressure to the system flow. The operating points are the intersections of these curves. The normal operating region falls between the low and the high demand load curves in Figure 3-33. Axial compressors are more efficient; centrifugal ones are better suited for dirty or corrosive services.

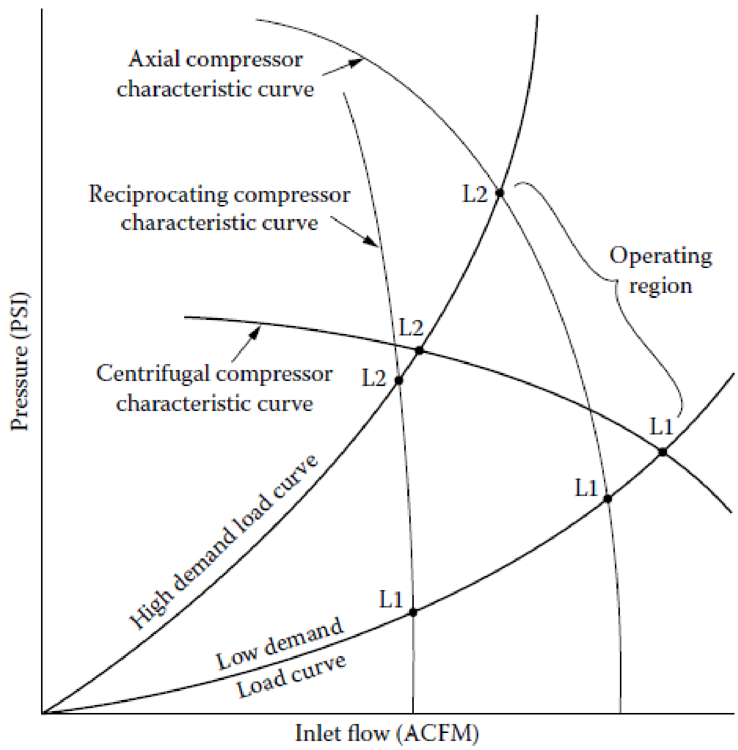



**Figure 3-33: Characteristic curves comparison of different compressor types**

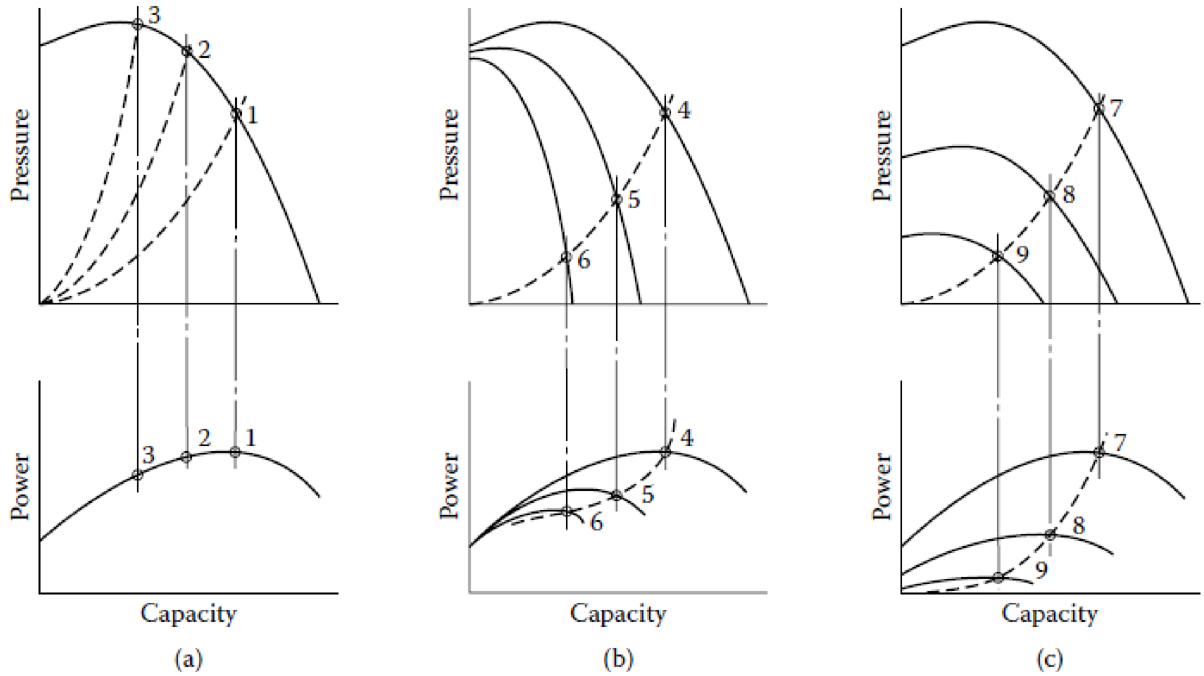



**Figure 3-34: The efficiencies of discharge throttling (left), suction throttling (centre), and variable speed control (right).**

Compressor loading can be reduced by throttling a discharge or a suction valve, by modulating a pre-rotation vane, or by reducing the speed. As is shown in Figure 3-34, discharge throttling is the least energy efficient and speed modulation is the most energy efficient method of turndown. Suction throttling is a little more efficient and gives a little better turndown than discharge throttling, but it is still a means of wasting that transporting energy that should not have been introduced in the first place.

Guide vane positioning, which provides pre-rotation or counter-rotation to the gas, is not as efficient as speed modulation, but it does provide the greatest turndown. Speed control is the most efficient, as small speed reductions result in large power savings because of the cubic relationship between speed and power.

If the discharge pressure is constant, flow tends to vary linearly with speed. If the discharge head is allowed to vary, it will change with the square of flow and, therefore, with the square of speed as well. This square relationship between speed and pressure tends to limit the speed range of compressors to the upper 30% of their range.

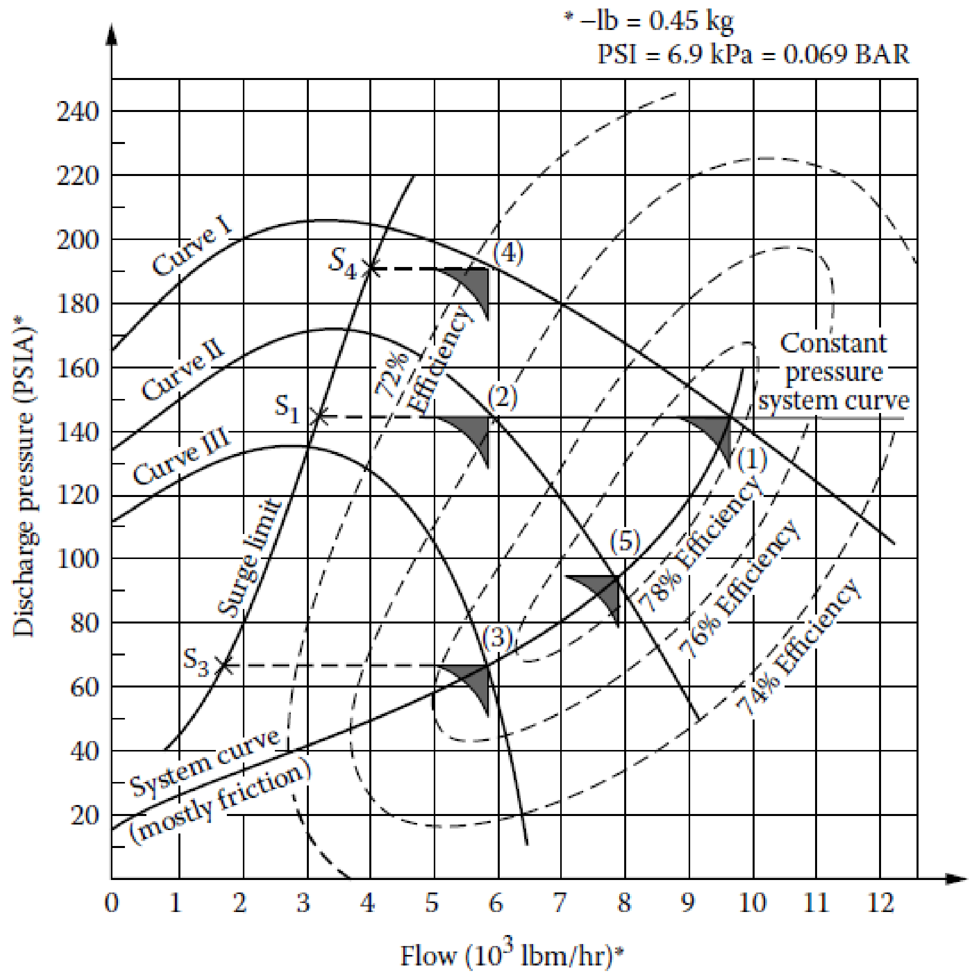



**Figure 3-35: Characteristic centrifugal curve with operating points at intersection with system curve (Imperial units)**

## a. Suction Throttling

One can control the capacity of a centrifugal compressor by throttling a control valve in the suction line, thereby altering the inlet pressure. From Figure 3-35 it can be seen that the discharge pressure will be altered for a given flow and a new compressor curve will be generated. Consider first that the compressor is operating at its normal inlet pressure (following curve I) and is intersecting the "constant pressure system" curve at point (1) with a design flow of 9600 lbm/hr (4320 kg/hr) at a discharge pressure of 144 psia (1 MPa) and 78% efficiency. If it is desired to change the flow to 5900 lbm/hr (2655 kg/hr) while maintaining the same discharge pressure, it would be necessary to shift the compressor from curve I to curve II, at 74% efficiency.

### b. Discharge Throttling

A control valve on the discharge of the centrifugal compressor may also be used to control its capacity. In Figure 3-35, if the flow is to be reduced from 9600 lbm/hr (4320 kg/hr) at point (1) to 5900 lbm/hr (2655 kg/hr), the compressor must follow curve I and therefore operate at point (4), at 190 psia (1.3 MPa) discharge pressure and 72% efficiency. Surge is more likely to occur in a mostly friction system when discharge throttling is used than when suction throttling is used.

### c. Inlet Guide Vanes

This method of control uses a set of adjustable guide vanes on the inlet to one or more of the compressor stages. By pre-rotation or counter-rotation of the gas stream relative to the impeller rotation, the stage is unloaded or loaded, thus lowering or raising the discharge head. The effect is similar to suction throttling as illustrated in Figure 3-35 (curves II and III), but less power is wasted because pressure is not throttled directly. Also, the control is two-directional, since it may be used to raise as well as to lower the band. It is more complex and expensive than throttling valves but may save 9 to 15% on power and is well suited for use on constant-speed machines in applications involving wide flow variations.

The guide vane effect on flow is more pronounced in constant discharge pressure systems. This can be seen in curve II, where the intersection with the “constant pressure system” at point (2) represents a flow change from the normal design point (1) of 9600 – 5900 =3700 lbm/hr (1665 kg/hr). The intersection with the “mostly friction system” at point (5) represents a flow change of only 9600 – 7800 = 1800 lbm/hr (810 kg/hr).

## d. Variable Speed

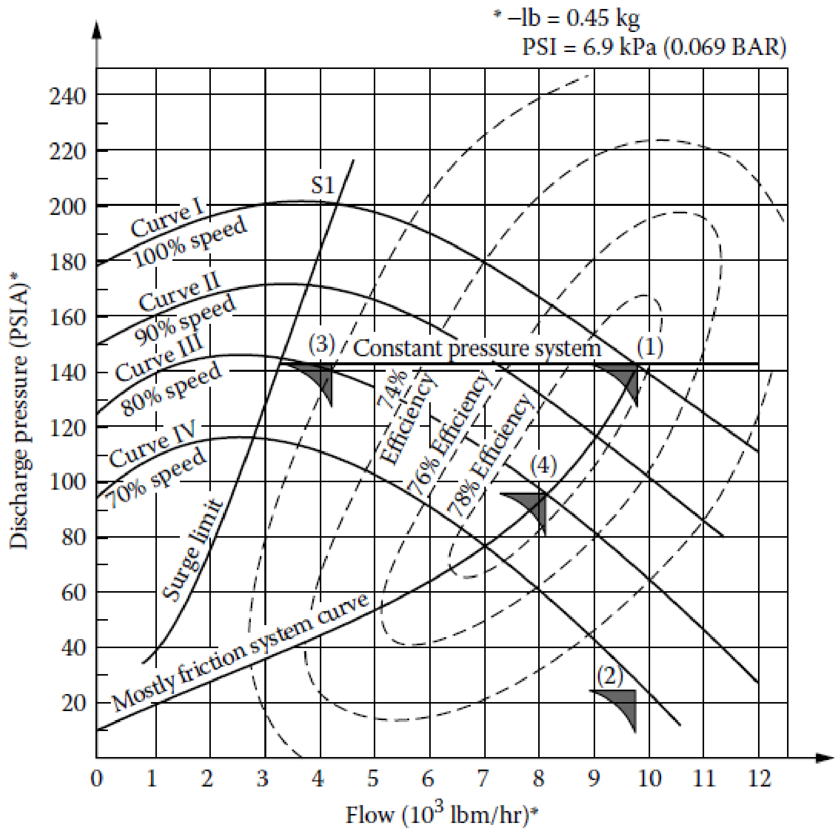



**Figure 3-36: Control of centrifugal compressor capacity by speed variation (Imperial units)**

The obvious advantage of speed control from a process viewpoint is that both suction and discharge pressures can be specified independently of the flow. The normal flow is shown at point (1) for 9700 lbm/hr (4365 kg/hr) at 142 psia (0.98 MPa). If the same flow is desired at a discharge pressure of 25 psia (173 kPa), the speed is reduced to 70% of design, shown at point (2). In order to achieve the same result through suction throttling with a pressure ratio of 10:1, the pressure drop across the valve would have to be (142−25)/10 = 11.7 psi (81 kPa), with the attendant waste of power, as a result of throttling. This is in contrast with a power saving accomplished with speed control, because power input is reduced as the square of the speed.

One disadvantage of speed control is apparent in constant pressure systems, in which the change in capacity may be overly sensitive to relatively small speed changes. This is shown at point (3), where a 20% speed change gives a flow change of (9600 − 4300)/9600 = 55%. The effect is less pronounced in a "mostly friction system," in which the flow change that results from a 20% speed change at point (4) is (9600 − 8100)/9600 = 16%.

## 3.14 Operation and Maintenance

Operation and maintenance of centrifugal compressors with particular attention given to routine checks is very important. Centrifugal compressors are capable of long periods of operation with little major repair and overhaul. However, a minor problem with a component requiring routine operational checks such as an intercooler drain can result in major repair expense.

Routine daily checks should follow the manufacturer's recommendations which may include the following and more:

- Air intake filter pressure drop
- Air inlet temperature and pressure
- Inter-stage temperatures and pressures
- Cooling water inlet, and outlet pressures and temperatures at each cooler on the compressor.
- Lubricant pressure and temperature
- Check / verify proper operation of all condensate drain traps.
- Check vibration levels.

When daily checks are made they should be recorded in a log. The log must be reviewed to assess trends of temperature, pressure, and/or vibration changes that may forewarn of a more serious condition. In many applications these checks are made once per shift of operation (3 times / day for 24 hour running).

On a somewhat less frequent basis, perhaps monthly operational checks should be made. Particular attention should be given to critical surge control, and capacity control functions. Check air control piping, lubricant piping, and water piping for leaks or breaks. Clean and/or replace condensate traps as necessary. Check lubricant level, check lubricant sump venting, and service as required. Check cooling water system components including pressure, and temperature gauges, clean and/or replace inline strainers.

Other checks for loose bolts, fittings, wear and tear are recommended at particular intervals. Follow the manufacturer's recommendations. Work with the compressor manufacturer, and your lubricant supplier to adopt a regular lubricant testing program. Follow instructions for lubricant replacement. Centrifugal compressors critically rely on proper condition of the lubricant. Poor lubricant performance can, in a very short period of time, lead to major component failures and expensive (10's of thousands of dollars) repair.

Routine maintenance is an important aspect of operating any machinery. Compressor designs other than centrifugal compressors can be somewhat more forgiving to less optimal operational checks and routine maintenance. If the plant has a history of marginal effectiveness carrying out routine checks / maintenance that will prevent small problems from becoming large problems, it may be advisable to consider other compressor designs. Run to failure maintenance of centrifugal compressors can be very expensive.

## 3.15 Key Learning Points

1. There are two broad categories of industrial air compressors, positive displacement and dynamic.

2. Reciprocating compressors are positive displacement compressors.

3. Rotary screw compressors are also positive displacement compressors.

4. Rotary screw compressors are the most common type of industrial air compressor.

5. There are many different types of part load capacity control for rotary screw compressors.

6. Centrifugal air compressors are the most common type of dynamic compressor used by industry.

7. Aerodynamic design determines the head -vs- flow performance curve for centrifugal air compressors.

8. Performing poor routine maintenance for centrifugal air compressors can lead to expensive failures of major air compressor components.

## 3.16 Key Energy Points

9. Different types of part load capacity control have different part load power characteristics.

10. Operating centrifugal compressors with blow-off control can be extremely inefficient.

11. Operating in the stonewall (or choke) region of a centrifugal compressor's performance range is inefficient.

12. In systems with rotary screw compressors it is most efficient to have all compressors operate at full load with only one air compressor at part load for trim capacity.

13. When operating multiple centrifugal air compressors in a system it is more efficient to have multiple compressors operate at part load within their throttle throttling range as opposed to operating in blow-off.

14. When operating a system using a combination of positive displacement and centrifugal compressors requires special attention to control strategy and the system's pressure profile.

# 4. Air Treatment

When air is compressed, everything in the air is compressed as well. Every compressor, regardless of type, acts to concentrate the pollutants in the ambient air. These pollutants include solid particles, hydrocarbon (oil, gasoline and diesel fumes) vapors, chemical vapors and water vapor. Using a lubricant-free, or dry compressor will not guarantee clean compressed air. If the pollutants in the ambient air are not removed from the compressed air, they will be concentrated in the air distribution system and in the equipment using the compressed air.

## 4.1 Air Quality Standards

There are various guidelines and standards that help define air quality requirements. Two of the most common are ANSI/ISA-7.0.01-1996 Quality Standard for Instrument Air, and International Standard ISO 8573-1 Compressed air-Part:1 Contaminants and purity classes. The ISO 8573-1 international standard is very helpful in choosing the right system for the production of compressed air and the treatment of compressed air. The standard replaces vague quality terms such as “water-free“, “oil-free“ or “dust-free“ with simple numerical values and sorts them into defined classes of quality. The ISO 8573-1 standard specifies purity classes of compressed air in respect to particles, water, and oil regardless of the source of the compressed air. The standard does not recommend appropriate class levels for various compressed air demands, only the acceptable amount of various contaminates in a class level.

Compressed air purity classes for particulate contaminations specify a range of particle size and a corresponding allowable level of contamination, given in terms of the number of particles per cubic meter.

**Table 4-1: Particulate Contamination Classes**

| Class | Particle Size Range (µm) | (µm) | Particle Count [1] Max Number of Particles |
|---|---|---|---|
| 0 | As specified per application (more stringent than class 1) | | |
| 1 | 0.5 | 1.0 | 1 |
| 2 | 1.0 | 5.0 | 10 |
| 3 | 1.0 | 5.0 | 500 |
| 4 | 1.0 | 5.0 | 1,000 |
| 5 | 1.0 | 5.0 | 20,000 |
| 6 | -- | 5.0 | $\leq$ 5 mg/m$^3$ |
| 7 | -- | 40.0 | $\leq$ 10 mg/m$^3$ |

Humidity and liquid water classes are also defined. Humidity values are given as pressure dewpoint and liquid water as concentration of liquid water CW (g/m$^3$). For reference CW for saturated air at 30$^o$C is roughly 30 grams per cubic meter).

[1] The full specification includes particle counts for smaller particle sizes. Only the largest particle size ranges are included here.

**Table 4-2: Humidity and Liquid Water Classes**

| Class | Humidity<br>Pressure Dew Point | | Liquid Water Concentration<br>$LSL^2 \leq C_W \leq USL^3$<br>(g/m³) | (g/m³) |
|---|---|---|---|---|
| 0 | As specified per application (more stringent than class 1) | | | |
| 1 | ≤ -70°C | ≤ -94°F | - - | - - |
| 2 | ≤ -40°C | ≤ -40°F | - - | - - |
| 3 | ≤ -20°C | ≤ -4°F | - - | - - |
| 4 | ≤ +3°C | ≤ +37°F | - - | - - |
| 5 | ≤ +7°C | ≤ +45°F | - - | - - |
| 6 | ≤ +10°C | ≤ +50°F | - - | - - |
| 7 | - - | | 0 | 0.5 |
| 8 | - - | | 0.5 | 5.0 |
| 9 | - - | | 5.0 | 10 |

Oil contamination classifications are specified as total concentration of oil including aerosol, liquid, and vapor (mg/m³).

[2] LSL – Lower Spec. Limit

[3] USL – Upper Spec. Limit

**Table 4-3: Oil Classes (Total Concentration)**

| Class | Total Oil Concentration (Aerosol, Liquid, and Vapor) (mg/m$^3$) |
|---|---|
| 0 | As specified per application (more stringent than class 1) |
| 1 | ≤ 0.01 |
| 2 | ≤ 0.1 |
| 3 | ≤ 1.0 |
| 4 | ≤ 5.0 |

**Table 4-4: Typical Air Quality Class Recommendations**

| Typical Air Quality Classes[4] | | | | | | | |
|---|---|---|---|---|---|---|---|
| Application | Dirt | Water | Oil | Application | Dirt | Water | Oil |
| Air agitation | 3 | 5 | 3 | Industrial hand tools | 4 | 5-4 | 5-4 |
| Air bearing | 2 | 2 | 3 | Handling, food, beverages | 2 | 3 | 1 |
| Air gauging | 2 | 3 | 3 | Machine tools | 4 | 3 | 5 |
| Air motors, heavy | 4 | 4-1 | 5 | Mining | 4 | 5 | 5 |
| Air turbines | 2 | 2 | 3 | Packaging & textile machines | 4 | 3 | 3 |
| Brick & glass machines | 4 | 4 | 5 | Plant air, general | 4 | 4 | 5 |
| Cleaning machine parts | 4 | 4 | 4 | Precision pressure regulators | 3 | 2 | 3 |
| Construction | 4 | 5 | 5 | Process control instruments | 2 | 2 | 3 |
| Conveying powder products | 2 | 3 | 2 | Rock drills | 4 | 5-4 | 5 |
| Fluidics, power circuits | 4 | 4 | 4 | Sand blasting | - | 5-2 | 5 |
| Fluidics, sensors | 2 | 2-1 | 2 | Spray painting | 3 | 3-2 | 3 |
| | | | | Welding machines | 4 | 4 | 5 |

[4] Source: Compressed Air Challenge® Training Course; Advanced Management of Compressed Air Systems. These recommendations are typical only and are offered with no explicit or implied warranty or liability. For certain applications, more than one class may be considered. Ambient conditions will influence the selection, especially dew point. Point of use equipment manufacturers should be consulted to determine their specific needs.

## 4.2 Drying Compressed Air

The first task of an air treatment system is to remove the water from the compressed air. With inlet conditions of 20°C, 70% relative humidity (saturation) and 1 bar absolute, a $5m^3$/min compressor will compress enough water vapor to produce 30 liters of water per day. Compressing the water vapor changes the dew point. Dew points within the compressed air system are referred to as pressure dew points. As a rule, higher pressures result in higher pressure dew points. In this example, raising the pressure from 7 bar to 12 bar will increase the pressure dew point from 51°C to 62°C. At the point in the compressed air system that the temperature cools to the pressure dew point, the compressed water vapor will condense into liquid water.

If the 30 liters / day of water vapor entering the compressor intake, about 20 liters of water will condense in the after-cooler (based on 7 bar working pressure and 30°C discharge temperature). Since the air leaving the after-cooler will be at 100% RH, a portion of the remaining 10 liters will condense as the air cools in the piping system. As the air continues to cool within the distribution piping, water will continue to condense.

A refrigerated dryer with a 3 ºC pressure dewpoint rating and a 25 ºC inlet temperature, will condense about 74% of the remaining 10 liters. As long as the pressure in the distribution system stays relatively constant, the temperature of the compressed air must drop below 3ºC before further condensation occurs. The remaining water would stay in a vapor form until released to atmosphere, at which time the cooling effect of the drop in pressure may cause the vapor to condense.

Typical condensation points include; the lubricant separator tank of a rotary screw compressor, the after-cooler, storage receiver, refrigerated dryer and piping. Condensation in two of these areas is very undesirable. Water condensing in the lubricant separator tank will result in water being sent to the compressor bearings instead of lubricant. Liquid water in the compressed air distribution lines will damage equipment, cause the air-lines to rust, and increase maintenance costs.

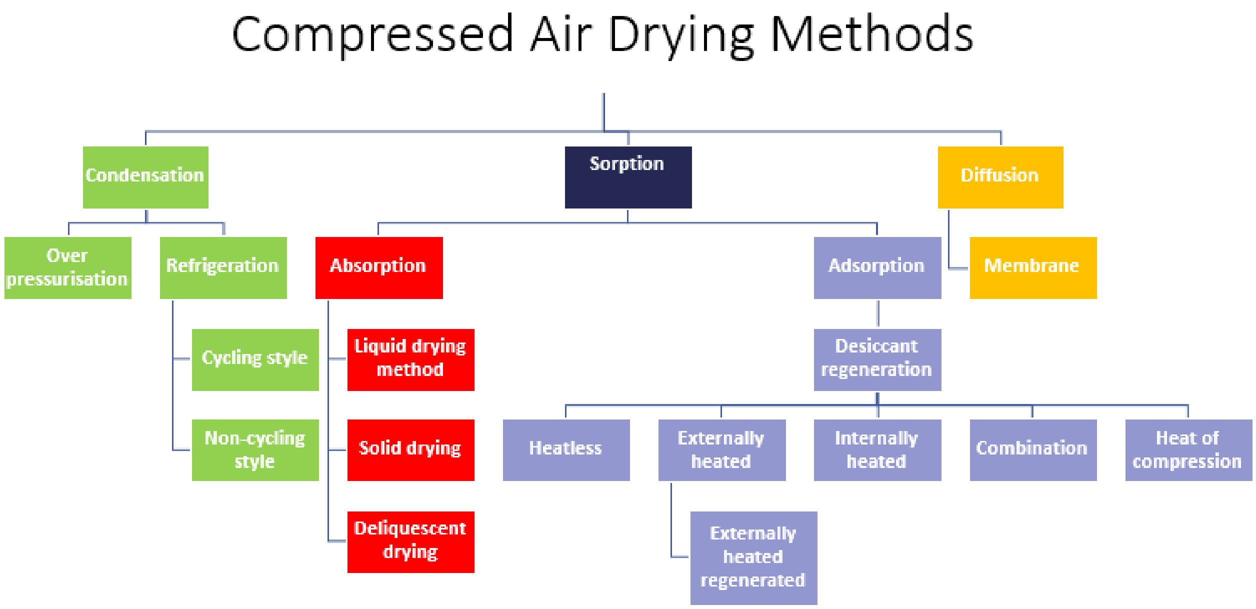



**Figure 4-1: Drying Methods**

- Sorption: Removal of humidity by the action of either Adsorption, or Absorption
  - Adsorption: Physical process – Humidity is bonded to the drying medium by molecular force.
  - Absorption: Chemical process – Humidity is extracted by a chemical reaction with the drying medium.
- Condensation: Water vapor is reduced by cooling air below the dew point temperature and removal of the liquid water that is formed;
- Over-pressurization with subsequent expansion; as air expands its volume increases and relative humidity decreases.
- Mechanical Cooling with the refrigerant circulation of a refrigerated dryer, cooling form condensate which is removed.
- Diffusion: Humidity permeates through a membrane due to the differences in partial pressure of water vapor and the other gases in air.

### a. Drying by Over-pressurization

Over-pressurization is the simplest method of drying compressed air. The air is usually compressed to 20 times the desired working pressure. A 15 bar application would typically see a pressure of 300 bar during the drying process. Over-pressurization concentrates the water vapor and radically changes the pressure dew point. The over-pressurized air is fed through an after-cooler with a moisture separator and trap. The air is then allowed to expand to the actual desired working pressure. Very low pressure dew points can be achieved in this manner. However, the energy required to dry air using this method is very high and the equipment is very expensive. Other methods are much more energy efficient, cost less, and produce the same result.

### b. Drying by Cooling

Drying by cooling is the method used be several different devices. After-coolers both air-cooled and water-cooled condense water vapor to liquid water by cooling the compressed air.

The working principle of a refrigerated dryer can be divided into four stages:

- Warm compressed air enters the dryer and is initially cooled in a heat exchanger by the cold compressed air leaving the refrigerated portion of the dryer.
- Further cooling occurs in another heat exchanger with refrigerant circulation. Typically, the air is cooled to about 3°C to 5°C.
- The condensate is separated from the compressed air by a separating system and then drained out of the dryer.
- The dried air is reheated as it passes through the heat exchanger containing the warm incoming air stream. Once through the process, the relative humidity of the compressed air has been lowered to about 10-25%.

Refrigerated dryers are usually rated for inlet air that is 100% saturated, about 40°C, and at a pressure of about 7 bar. Special dryers are available for higher inlet temperatures and higher pressures.

### c. Drying by Diffusion

A membrane dryer consists of a bundle of thin, hollow manmade fibers that were specifically developed for this application. These fibers are treated mechanically and chemically so that water vapor contained in the compressed air permeates through very fine pores in the surface of the fibers. Moist compressed air flows into one end of the fiber bundle, the water vapor permeates through the pores to the outside, and dry air leaves the other end of the fiber bundle. During this process, a small volume of air also escapes and acts as a purge to push the water vapor out of the housing. The water vapor molecules are drawn through the pores because the partial pressure of water vapor inside the tubes is greater than the partial pressure outside of the tubes.

These dryers work best as point of use dryers. They require no outside power source and can easily be installed in the compressed air line. They are excellent for use in hazardous areas. They are not well suited for large scale drying due to the amount of purge air loss (typically 20% to 40% of rated flow) and their susceptibility to contamination.

### d. Drying by Absorption

Absorption is a chemical process by which water vapor in the compressed air is absorbed by a hygroscopic drying medium (desiccant) through a chemical reaction. In most industrial applications, a solid drying agent is used. The drying agent is usually in a pellet form and held in a vertical tank. Air flows into the bottom of the tank and then up through the desiccant bed. The water vapor is partially absorbed by the desiccant. Condensate collects at the bottom of the tank to be drained away. Because the desiccant is being liquefied, new desiccant must be added periodically. This type of dryer can only suppress the pressure dewpoint to about 15°C. Dryers of this type should only be considered in applications where further cooling of the air in the distribution system is impossible. If the inlet air temperature is less than 30°C, the drying medium moistens, packs down and causes excessive pressure drop.

### e. Drying by Adsorption (Heatless)

In a typical dryer of this type, the compressed air flowing from the compressor and the air receiver must be free of liquid and solid contaminations (of particle size down to 0.01 mm). The compressed air then flows, via the changeover valve, into the flow distributor where it is evenly distributed over the complete cross section of the desiccant chamber. In the mass loading zone, the main proportion of moisture in the air is removed by the desiccant through the adsorption process. The second third of the chamber extracts the remaining moisture from the air, allowing the required dew point to be reached. The last third of the chamber serves as a safety reserve. The compressed air leaves the desiccant zone via the outlet diffuser and is then purified of dust particles (particle size down to 1 mm) by a particulate filter.

The desiccant is regenerated with a portion of air dried in the first chamber. This purge is guided in counter-flow through the second chamber. Because of the expansion of the air, the capability of absorbing moisture is increased and the dry air is quite able to regenerate the desiccant. The volume of purge required is dependent on physical laws and can be optimized by the adjustable purge jet (if one is fitted to the dryer). The saturated purge air exhausts from the dryer via a silenced purge exhaust.

This type of dryer does not require much energy for operation. The dryer does require a significant amount (up to 15% to 20%) of dry, compressed air, and the cost of that air should be considered when choosing a heatless, regenerative dryer.

### f. Drying by Adsorption (Internally Heated)

The compressed air flowing from the compressor package (after-cooler, centrifugal separator, and the air receiver) flows initially through a pre-filter that removes solid and liquid dirt particles and oil aerosols prior to going into the adsorption dryer. The adsorption dryer extracts water vapors from the air. This is achieved by routing the air via an inlet valve into the first drying chamber, which is filled with desiccant, where the moisture is removed from the air by adsorption. The air leaves the dryer via a changeover valve and is fed to an after-filter downstream to remove any desiccant dust.

During the adsorption process in the first chamber, the second chamber is regenerated. Heating the desiccant bed with an internal electric heater and purging out the water vapor accomplish this. A small proportion (2-3%) of previously dried and expanded air transports the moisture via the outlet valve to the atmosphere. Around 5% of the compressed air is required for pressure cooling of the desiccant.

When the first chamber is almost saturated with moisture, automatic valves reverse the process so that first chamber is regenerated and the air is dried in the second chamber. This changeover occurs either according to a determined time interval or according to the effective water loading, depending on the type of controller.

### g. Drying by Adsorption (Externally Heated)

Externally heated desiccant dryers dry though the same process as internally heated dryers. The primary difference is the heater configuration used to increase temperature during regeneration of the desiccant bed.

Blower Purge Dryers regenerate the desiccant bed using a low pressure blower to draw in ambient air that is cleaned by an inlet filter. A heater heats up this purge air to a temperature of 120-160°C before it is fed in counter-flow for regeneration of the desiccant bed in the second chamber. When the regeneration process is complete, that is, when the moisture is extracted from the desiccant, a temperature sensor switches off the heater. The blower air, which is now cool, extracts the heat from the desiccant. In the last phase of this cooling procedure compressed air is routed via a purge jet to prevent atmospheric air from saturating the desiccant with moisture again. Finally, the purge air leaves the dryer via a pipe work system to the open air.

The volume of purge air required averages approximately 2% of the dryer capacity. It is important to understand the actual purge flow rate during the cooling phase is greater than the 2% average purge. This cooling procedure allows a constant, low pressure dew point to be achieved under all operational conditions.

There are other variations of desiccant dryers that are designed to further reduce the amount of purge air required or achieve lower dew points by pre-drying the air with a refrigerated air dryer.

### h. Drying by Adsorption (Heat of Compression)

Heat of compression dryers use the heat from lubricant free compressors to provide the heat to regenerate the desiccant. Typically, a hot stream of air from the first stage of the compressor is directed through the regenerating side of the dryer to drive off the moisture. These dryers do not hold a constant low dewpoint like the other desiccant dryers but provide a dewpoint suppression below inlet air temperature. Even so, excellent dewpoints can be achieved for energy levels similar to refrigerated dryers. Heat of compression dryers can only be used with lubricant free compressors.

## 4.3 Selection of the dew point

The term "pressure dew point" (PDP) is used to describe the degree of dryness of compressed air. This is the saturation temperature, or the minimum temperature that all of the water vapor will remain as a gas. Thus, a PDP of +5°C means that the compressed air is saturated at only +5°C.

If for example; an after-cooler has an outlet temperature of +30°C, then 1 m3 of compressed air still contains around 30 g of water. If a refrigerated dryer with a PDP of +5°C is connected the water content of 1 $m^3$ of air is 7 grams. Then 23 g (30 – 7) of water is separated by the dryer's cooling action for each $m^3$.

Or, looked at differently, if the saturated water content of the compressed air at +20°C is 100%, then at a pressure dew point of +8°C, the water content is reduced by 52%, at +2°C by 70% and at –20°C by 94%. The following points must be observed when determining the optimal pressure dew point:

- Outlet temperature of the compressed air at the after-cooler or air receiver
- Temperature of the ambient air, according to seasons
- Temperature of the walls along which the piping will be laid (below windows too)
- Whether or not the piping will be laid in the open air
- The lowest possible temperature when the equipment is switched off and the piping system cools down (depending on inside and outside temperatures)

If these points are observed the lowest temperature occurring in the compressed air piping will be determined. If only a small section of the piping is exposed to the open air, a relatively high flow velocity may maintain enough heat to prevent condensation. The pressure dew point is determined either by using a pressure dew point that is below the lowest temperature in the piping system or by the requirements of the respective area of application of the compressed air. If a PDP of +4°C is required, a dryer for +2°C should be selected but only needs to operate at +4°C. A 1°C lower dew point means approximately 4 % more power consumption.

## 4.4 After-cooling

The after-cooler may be a permanent component of the compressor. However, because the outlet temperature of the after-cooler is basically the same as the inlet temperature of the dryer, the function of the after-cooler must be taken into account when selecting the dryer. A compressor operates with either air or water-cooling. If air-cooling is used, the compressed air temperature can be 10° to 20°C above the actual cooling air temperature. That means an outlet temperature of 30° to 45°C in summer and 10° to 20°C in winter. On water-cooled after-coolers, a temperature difference of around 10°C can be expected. The achievable outlet temperature in this case depends on the temperature of the water, which can vary considerably. For example, cooling tower water is approximately 30°C, tap water is 10° to 15°C.

For every 11°C increase in compressed air temperature the amount of water vapour doubles. The associated economic appraisal must be calculated in connection with the selection of the compressor.

## 4.5 Selection of dryer type

Normally four types of dryer are available:

- Refrigeration drying with a pressure dew point down to +2°C (1°-5°C)
- Adsorption dryers with pressure dew point from –20°C down to –80°C; desiccant dryers can be either heat regenerated or heatless regenerated
- Absorption dryers; the pressure dew point depends on the inlet air temperature and partially on the ambient temperature. For example PDP = -11°C for an inlet temperature and ambient temperature of 40 °C. This type of dryer is only used in special cases.
- Sorption drying with pressure dew point around – 30 °C. This method is only suitable for oil free air.

The various types of air drying require different energy intensity. Generally, lower compressed air pressure dew point performance requires increased energy per unit of compressed air flow. The following relative energy performance is from Compressed Air Challenge®:

- 1.6 kW / 100 l/s Refrigeration drying
- 0.4 - 0.6 kW / 100 l/s Adsorption[5] dryers with pressure dew point of –40°C
- 0.4 kW Absorption dryers; the pressure dew point varies.
- 1.6 kW / 100 l/s Sorption drying with pressure dew point of – 30 °C.

If the piping is laid indoors, a refrigeration dryer is generally sufficient. These dryers function mostly in two stages. In the first stage, the inlet air is cooled by the cold air leaving the dryer. This removes approximately 60% of the water. The rest is removed by refrigeration. About 90% of all dryers are refrigeration dryers.

Almost without exception, desiccant adsorption dryers are used where pressure dew points below 0°C are required. Water vapor is bonded to the desiccant by molecular force within these dryers. The desiccant can then be regenerated with or without heat.

Heatless regenerated dryers may be cheaper to purchase, but very expensive to operate. Their inlet temperature can be as high as +60°C. In contrast, heated dryers are very expensive to procure but operation is very much cheaper. Their inlet temperature should not exceed +40°C.

[5] The power requirement, includes pressure drop through the dryer and associated filtration, but excluding the cost of replacement desiccant. Absorption style dryers do not have any direct electrical energy input.

Absorption is, in comparison, a chemical process during which the water binds with the desiccant.

## 4.6 Dryer location

Generally, the location of a dryer in the compressed air system cannot be easily determined without understanding the dynamics of the system. Under conditions of high fluctuations in air demand, a location upstream of the air receiver may be advantageous as the dryer only needs to be matched to compressor performance. The outlet temperature of the after-cooler is the inlet temperature of the dryer. If the air receiver is located such that it cools the compressed air downstream of the after-cooler, further consideration should be given to locating the dryer downstream of the air receiver. As water has already been separated in the air receiver, a smaller dryer could be chosen. However, the dryer must have a capacity large enough to accommodate the maximum possible volume flow. There is a risk in sizing the dryer to the actual air demand because experience shows that, with time, more and more consumers are connected to the air main.

## 4.7 Key Learning Points – Air Treatment Technology

1. Air quality standards should be consulted to establish and measure contamination classes.

2. Application of air treatment equipment should achieve the necessary cleanliness class for the applications being supplied. Overtreatment of compressed air should be avoided because of the additional expense and inefficiencies.

3. Various filter types remove specific contaminants. Select and apply filters based on the type of contaminate to be removed and the degree of cleanliness required.

## 4.8 Key Energy Points – Air Treatment Technology

4. Consider the cost of pressure drop when applying and maintaining compressed air filters.

5. Various types of air dryers achieve different pressure dew point performance. Lower dew point performance increases initial equipment cost and operating cost.

6. Dry compressed air to an appropriate humidity class. Over drying the compressed air is very expensive and should be avoided.

# 5. Demand Side: Eliminate Compressed Air Waste

## 5.1 Minimize Compressed Air Demand – Eliminate Waste

As described above, reducing the pneumatic workload by 1 kW will reduce the power required to compress that air by 6.67 kW. This makes the demand side the obvious place to start optimizing the system. Changes made here to reduce compressed air demand will have the largest impact on reducing energy consumption.

Only a portion of the compressed air supplied to a system is used by production operations. Much of the compressed air delivered to the system; often 50% or more, is wasted. Compressed air waste is a result of decisions made with respect to design, operation, and maintenance of the compressed air system.

Compressed air and energy waste is a result of:

- inappropriate use of compressed air,
- operating the system at excessively high pressure,
- inadequate and / or uncontrolled compressed air energy storage
- leakage, including open drip legs and condensate drains,
- irrecoverable pressure loss in restrictive piping, filters, and other components
- inefficient compressor control strategy.

## 5.2 Leakages

Everyone knows compressed air systems leak. What is not necessarily known is how much leakage there is, or how much it costs. Reasonably well-maintained systems can lose 20% of the air generation to leakage. Poorly maintained systems might lose 40% or 50%, or even more! A 3mm hole with a well-rounded orifice will flow about 0.5$m^3$/minute at 6 bar. In one year, a leak of that size will leak over 240,000 $m^3$ of air***. The percentage lost to leakage should be less than 10% in a well-maintained system***

**Table 5-1: Flow of compressed air through an orifice & power loss**

| | Air Consumption at 6 bar (g) (m3/min) | | Power Loss (kW) | |
|---|---|---|---|---|
| Hole Diameter | sharp orifice 0.61 coefficient | rounded orifice 0.97 coefficient | Shaft Power 6.2 kW / $m^3$/min. | Package Power 7.1 kW / $m^3$/min. |
| 1mm | 0,040 | 0,064 | 0,25 to 0,40 | 0,28 to 0,45 |
| 2mm | 0,16 | 0,25 | 0,62 to 1,5 | 1,1 to 1,8 |
| 3mm | 0,35 | 0,56 | 2,2 to 3,1 | 2,5 to 4,0 |
| 4mm | 0,63 | 1,00 | 3,9 to 6,2 | 4,5 to 7,1 |
| 6mm | 1,42 | 2,26 | 8,8 to 14,0 | 10,0 to 16,0 |

## a. Tests to Estimate Overall Leakage Rate

The tests described here to estimate total plant compressed air leakage must be performed during a time when plant production is idle. The measurements and calculations made will quantify the compressed airflow rate to the plant during the test. That airflow rate is not necessarily 100% compressed air leakage. During non-production time periods there are typically three types of compressed air consumption:

1. Leakage
2. Idle equipment
3. Residual compressed air use

Leakage is compressed air lost from the system through openings to atmosphere that can be repaired there-by preventing the escape of air from the compressed air system.

Idle equipment consumption is compressed air demand that is associated with normal production operations. However, during non-production time periods the compressed air use can (and should be) turned off.

Residual compressed air use is an end use of compressed air supplying production equipment or a process application that must continue to be supplied even during time periods when normal production is not operating.

After completing the tests and calculations describe here it is up to the compressed air system expert and plant personnel to estimate the amount of compressed airflow that is attributed to each of the three types of compressed air consumption.

## b. Leakage Test with Load / No-Load Air Compressor Control

In systems with load / unload air compressor operating as trim capacity, the approximate amount of leakage can be determined measuring the load cycle of the trim compressor. The load cycle measurement should be made during a time when normal production is idle. The formula used for this method is:

**Equation 5-1: Leakage Flow Rate Calculated for Load / No-Load trim Compressor**

$$Q = \frac{V_C \times T_{FL}}{T_{FL} + T_{NL}}$$

Where:

| | | | | | |
|---|---|---|---|---|---|
| Q | = | Flow Rate ($m^3$ / minute) | $T_{FL}$ | = | Time at full load (minutes) |
| $V_C$ | = | *Compressor volume flow rate ($m^3$ / minute) | $T_{NL}$ | = | Time at no load (minutes) |

*NOTE: using compressor mass flow rate ($Nm^3$ / minute) would improve accuracy. Since this is an estimate only volume flow rate is used.

## c. Sample Calculation – Leakage Test L / NL Compressor

For example, when measuring several load cycles of the trim compressor operating during a time when normal production is idle, the following is observed.

| | | | | |
|---|---|---|---|---|
| $T_{FL}$ = 1 m 10 s | 1 m 19 s | 1 m 15 s | 1 m 22 s | Total 5.1 m |
| $T_{NL}$ = 0 m 35 s | 0 m 32 s | 0 m 38 s | 0 m 35 s | Total 2.3 m |

The trim air compressor is rated for 8.2 $m^3$ / minute and is in good working condition and delivering its rated airflow. Using Equation 5-1:

$$V_L = \frac{8.2\ m^3/\min \times 5.1m}{5.1m + 2.3m} = 5.6m^3/\min$$

## c. Compressed Air System Pressure Bleed-Down Test

The pressure bleed-down test can be applied to any system regardless of the type(s) of air compressor(s) used. It is necessary to estimate the total volume of the compressed air system including air receivers, and plant air piping.

In compressed air systems without load / no-load air compressors a system bleed-down test can be performed. Again this test should be conducted when normal production operations are shutdown. Operate the necessary compressor(s) to maintain the system at its normal operating pressure ($P_I$). Then when possible, unload or shutdown all of the compressors in the system. Measure the time required for the pressure to decay to some lower pressure ($P_F$). Then to determine the non-production plant airflow rate. The formula for this method is:

In small compressed air systems, shutting off all air use and measuring the time required for the pressure to drop can determine the leak rate. The formula for this method is:

**Equation 5-2: Flow Rate Calculation for Bleed down Test**

$$Q = \frac{V_R \times (P_I - P_F)}{T \; x \; P_A}$$

Where:

| | | | | | |
|---|---|---|---|---|---|
| Q | = | Flow rate | $P_F$ | = | Final pressure |
| $V_R$ | = | *Receiver & piping volume | T | = | Measuring period (time) |
| $P_I$ | = | Initial pressure | $P_A$ | = | Atmospheric pressure |

* NOTE: System volume is an estimate that can be calculated based on receiver volumes and estimated pipe volume of the system's largest diameter and longest pipelines.

## d. Sample Calculation – System Bleed Down Test

For example, a compressed air system with 25 $m^3$/ minute of total air compressor capacity has a total volume (including air receivers and piping) of 7.63 $m^3$.

A bleed down test is performed which resulted in the pressure decay curve shown in Figure 5-1 below.

The normal system pressure ($P_I$) was 6.33 bar at the time when all compressors were unloaded (or stopped). The system pressure was allowed to decay to 3.17 bar ($P_F$) (50% of the normal working pressure) and the time measured was 3 minutes 20 seconds (T = 3.33 minutes).

Substituting the test results into Equation 5-2 the compressed air non-production system flow rate is calculated to be 7.23 $m^3$ / minute.

$$Q = \frac{7.63 \text{ m}^3 \times (6.33 \text{ bar} - 3.17 \text{ bar})}{3.33 \text{ minutes} \times 1 \text{ bar}} = 7.23 \text{ m}^3/\text{min}$$

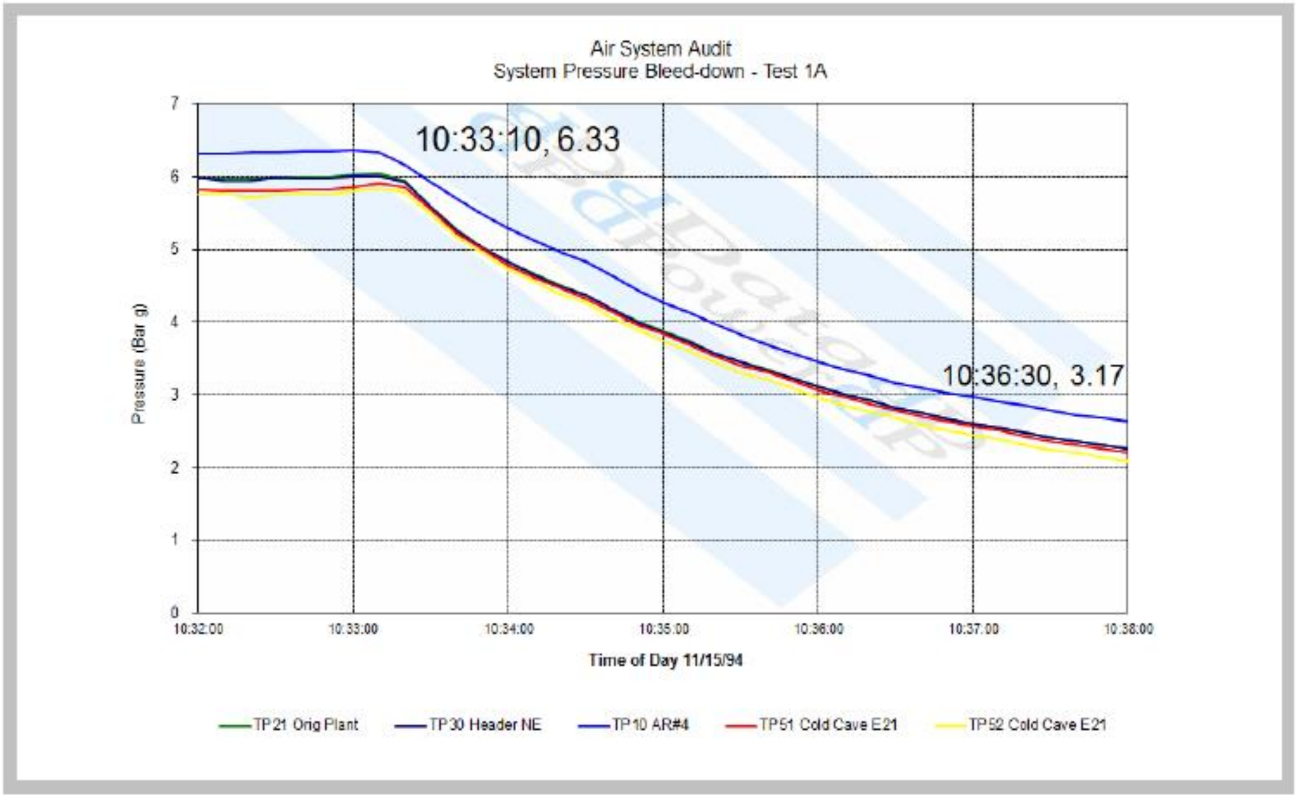



**Figure 5-1: System Leakage Pressure Bleed-down Test**

Using Equation 5-2, the calculated airflow rate is actually the rate at the bleed-down test's average pressure of 4.75 bar. However, the normal plant pressure is 6.33 bar. The non-production compressed air flow rate would be greater at the higher pressure. Equation 5-2 must be modified to calculate the non-production airflow rate at the normal system pressure.

To correct the airflow rate Equation 5-3 includes a factor of 1.25 in the calculation. This factor corrects the calculated air flow rate from the average pressure during the bleed-down test to the normal system working pressure. The factor of **1.25 is only valid** if the initial pressure ($P_I$) is the plant's normal working pressure **AND** if the final pressure ($P_F$) for the bleed-down test is ½ of the initial pressure ($P_I$).

**Equation 5-3: Bleed down Test w/ Air Flow Rate adjusted to normal pressure**

$$Q = \frac{V_R \times (P_I - P_F)}{T \ x \ P_A} \times 1.25$$

Where:

| | | | | | |
|---|---|---|---|---|---|
| Q | = | Flow rate | $P_F$ | = | Final pressure |
| $V_R$ | = | Receiver & piping volume | T | = | Measuring period (time) |
| $P_I$ | = | Initial pressure | $P_A$ | = | Atmospheric pressure |

Multiplying the result from Equation 5-2 Flow Rate Calculation for Bleed down Test (7.23 m$^3$/ minute) by 1.25 indicates that the plant's total non-production compressed airflow rate is 9.0 m$^3$ / minute, which is the flow rate that would exist when the plant is at its normal working pressure of 6.33 bar.

## d. Estimating the Leakage Portion of Non-Production Airflow Rate

It is important to remember that the non-production airflow rate calculated here is not necessarily all leakage. As discussed earlier there are 3 components of the non-production air demand; leakage, idle equipment, and residual air demand.

## e. Sample Calculation – Estimate Leakage Component of Non-production Airflow

For example, two air demands that are not leakage are below:

***The plant has a large paint area with residual air demand of 1 m$^3$ / minute required 24 / 7 to power air motors driving paint mixers.***

***There is also a robotics assembly cell that has an airflow rate of 0.7 m$^3$ / minute when the robots are idle. Note: there is a shutdown procedure that allows the robots to be parked in which case the compressed air can be shutdown. The plant has decided to leave the robots idle due to the time required for shutdown and start-up procedures of the robots.***

Therefore, accounting for 1.7 m$^3$ / minute airflow rate of residual, and idle equipment demand, the adjusted leakage rate is estimated to be 5.53 m$^3$ / minute which is 22% of the system capacity.

## f. Other Methods to Estimate Overall Compressed Air Leakage

In very large compressed air systems that operate 24 hours per day, and supplying multiple production areas, it may be almost impossible to have a time when all production is shutdown. One alternative may be; if continuous airflow measurement is available, leakage trends may be tracked during holidays or other times of minimum production.

In this situation, another leakage estimating method may be to perform a partial ultrasonic leakage survey, covering perhaps 10% of each production area. Estimating the leakage for each are is a simple ratio of the area surveyed and the total production area. This method may also give insight as to where to best begin leak detection and repair.

The following table shows an estimate of compressed air leaks for the operating conditions shown. The data is based on a sharp edged orifice with a flow coefficient of 0.61. Leakage and cost could be increased by as much as 60% for a well-rounded hole (flow coefficient = 0.97).

**Table 5-2: Discharge of Air through an Orifice**

| Gauge pressure before orifice, bar | **Diameter of Orifice, mm** (note: calculated flow rate assumes orifice coefficient of 0.61) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 15 | 20 |
| 4 | 0.03 | 0.11 | 0.25 | 0.45 | 0.70 | 1.01 | 1.38 | 1.80 | 2.28 | 2.82 | 6.34 | 11.28 |
| 4.5 | 0.03 | 0.12 | 0.28 | 0.50 | 0.78 | 1.12 | 1.52 | 1.98 | 2.51 | 3.10 | 6.98 | 12.40 |
| 5 | 0.03 | 0.14 | 0.30 | 0.54 | 0.85 | 1.22 | 1.66 | 2.16 | 2.74 | 3.38 | 7.61 | 13.53 |
| 5.5 | 0.04 | 0.15 | 0.33 | 0.59 | 0.92 | 1.32 | 1.79 | 2.34 | 2.97 | 3.66 | 8.24 | 14.65 |
| 6 | 0.04 | 0.16 | 0.35 | 0.63 | 0.99 | 1.42 | 1.93 | 2.52 | 3.19 | 3.94 | 8.87 | 15.78 |
| 6.5 | 0.04 | 0.17 | 0.38 | 0.68 | 1.06 | 1.52 | 2.07 | 2.70 | 3.42 | 4.23 | 9.51 | 16.90 |
| 7 | 0.05 | 0.18 | 0.41 | 0.72 | 1.13 | 1.62 | 2.21 | 2.88 | 3.65 | 4.51 | 10.14 | 18.03 |
| 7.5 | 0.05 | 0.19 | 0.43 | 0.77 | 1.20 | 1.72 | 2.35 | 3.06 | 3.88 | 4.79 | 10.77 | 19.15 |
| 8 | 0.05 | 0.20 | 0.46 | 0.81 | 1.27 | 1.82 | 2.48 | 3.24 | 4.11 | 5.07 | 11.40 | 20.27 |
| 8.5 | 0.05 | 0.21 | 0.48 | 0.86 | 1.34 | 1.93 | 2.62 | 3.42 | 4.33 | 5.35 | 12.04 | 21.40 |
| 9 | 0.06 | 0.23 | 0.51 | 0.90 | 1.41 | 2.03 | 2.76 | 3.60 | 4.56 | 5.63 | 12.67 | 22.52 |
| 9.5 | 0.06 | 0.24 | 0.53 | 0.95 | 1.48 | 2.13 | 2.90 | 3.78 | 4.79 | 5.91 | 13.30 | 23.65 |
| 10 | 0.06 | 0.25 | 0.56 | 0.99 | 1.55 | 2.23 | 3.03 | 3.96 | 5.02 | 6.19 | 13.94 | 24.77 |

There is a component of artificial demand in leakage. The amount of compressed air lost to leakage is reduced at lower system pressure. Controlling system pressure can result in significant reduction in leakage cost without even fixing one leak. Repairing leaks is obviously important, however, repair priorities must be based on cost / pay back considerations.

### g. System Problems Caused by Leaks

Compressed air leaks can also contribute to problems with system operations, including:

- Fluctuating system pressure, which can cause air tools and other air-operated equipment to function less efficiently, possibly affecting production
- Leaks can lead to adding unnecessary excess compressor capacity, resulting in higher than necessary costs
- Increased running time lead to decreased service life, increased maintenance of supply equipment (including the compressor package) due to unnecessary cycling and increased run time, and increased unscheduled downtime.
- Although leaks can occur in any part of the system, the most common problem areas are: couplings, hoses, tubes, fittings, pipe joints, quick disconnects, FRLs (filter, regulator, and lubricator), condensate traps, valves, flanges, packings, thread sealants, and point of use devices. Leakage rates are a function of the supply pressure in an uncontrolled system and increase with higher system pressures. Leakage rates are also proportional to the square of the orifice diameter. (See table below.)

### h. Leak Detection

The best way to detect leaks is to use an ultrasonic acoustic detector, which can recognize high frequency hissing sounds associated with air leaks. These portable units are very easy to use, and consist of directional microphones, amplifiers, and audio filters, and usually have either visual indicators or earphones to detect leaks. Costs and sensitivities vary, so test before you buy. A simpler method is to apply soapy water with a paintbrush to suspect areas. Although reliable, this method can be time consuming and messy.

While leakage can come from any part of the system, the most common problem areas are:

- Couplings, hoses, tubes, and fittings,
- Pressure regulators,
- Open condensate traps and shut-off valves, and
- Pipe joints, disconnects, and thread sealants.

## i. How to Fix Leaks

Leaks occur most often at joints and connections. Stopping leaks can be as simple as tightening a connection or as complex as replacing faulty equipment such as couplings, fittings, pipe sections, hoses, joints, drains, and traps. In many cases leaks are caused by bad or improperly applied thread sealant. Select high quality fittings, disconnects, hose, tubing, and install them properly with appropriate thread sealant.

Non-operating equipment can be an additional source of leaks. Equipment no longer in use should be isolated with a valve in the distribution system.

Another way to reduce leaks is to lower the demand air pressure of the system. The lower the pressure differential across an orifice or leak, the lower the rate of flow, so reduced system pressure will result in reduced leakage rates. Stabilizing the system header pressure at its lowest practical range will minimize the leakage rate for the system. ***Once leaks have been repaired, the compressor control system should be re-evaluated to realize the total savings potential.***

## j. A Leak Prevention Program

A good leak prevention program will include the following components: identification (including tagging), tracking, repair, verification, and employee involvement. All facilities with compressed air systems should establish an aggressive leak program. A cross-cutting team involving decision-making representatives from production should be formed.

A leak prevention program should be part of an overall program aimed at improving the performance of compressed air systems. Once the leaks are found and repaired, the system should be re-evaluated.

## k. Leak Quantification Example

A chemical plant undertook a leak prevention program following a compressed air audit at their facility. Leaks, approximately equivalent to different orifice sizes, were found as follows: 100 leaks of 1mm at 6 bar, 50 leaks of 2mm at 6 bar, and 10 leaks of 6mm at 7 bar. Calculate the annual cost savings if these leaks were eliminated. Assume 7000 annual operating hours, an aggregate electric rate of $0.15/kWh, and compressed air generation requirement of approximately 6.3 kW/m$^3$/min.

Cost savings = # of leaks x leakage rate (m$^3$/min) x kW/ m$^3$/min x # of hours x $/kWh

Using values of the leakage rates from the above table at 0.61 sharp-edged orifices:

Cost savings from 1mm leaks = 100 x 0.04 $m^3$/min x 6.3 x 7000 x 0.15 = $26,460

Cost savings from 2mm leaks = 50 x 0.16 $m^3$/min x 6.3 x 7000 x 0.15 = $52,920

Cost savings from 6mm leaks = 10 x 1.62 $m^3$/min x 6.3 x 7000 x 0.15 = $107,163

**Total cost savings from eliminating these leaks = $186,543**

Note that the savings from the elimination of just 10 leaks of 6mm account for almost 60% of the overall savings. As leaks are identified, it is important to prioritize them and fix the largest ones first.

## 5.3 Artificial Demand- Increased Pressure Increases Air Demand

All compressed air systems have some minimum pressure that is necessary for proper function of the system. Frequently that pressure is not defined. However, as long as the pressure is above the minimum, for example 5.52 bar, there are no complaints. Running the compressors at 7.58 bar maintains 6.9 bar in the system and operations are satisfied. If 5.52 bar is acceptable, 6.9 bar is an improvement to operations. Correct?

Wrong! More pressure is not better. Supplying 1.4 bar more pressure to the system will force the system to consume 20% more air flow. The 20% of wasted airflow is called **Artificial Demand**. Artificial Demand

| Artificial Demand |
|---|
| ***Artificial Demand*** is the increased compressed air flow consumed by a compressed air system when the applied demand side pressure is increased above the lowest optimum pressure necessary to support productive air use. All unregulated compressed air use and unregulated leakages contribute to the system's total artificial demand. |

Applying increased air pressure (bar) to an orifice increases the flow rate ($m^3$/min) of compressed air through the orifice. A 8 mm diameter orifice with a 0.61 coefficient and 7 bar applied upstream, will pass 2.88 $m^3$/min of compressed air flow.

Suppose instead of 7 bar, the upstream pressure is controlled to be 6 bar. The same 8 mm diameter hole would only pass 2.52 $m^3$/min of compressed air. That's about 12% less air demand with 1.0 bar less pressure applied to the system.

Artificial Demand – Applying increased air pressure (bar) to an orifice, increases the flow rate ($m^3$/min) of compressed air through the orifice.

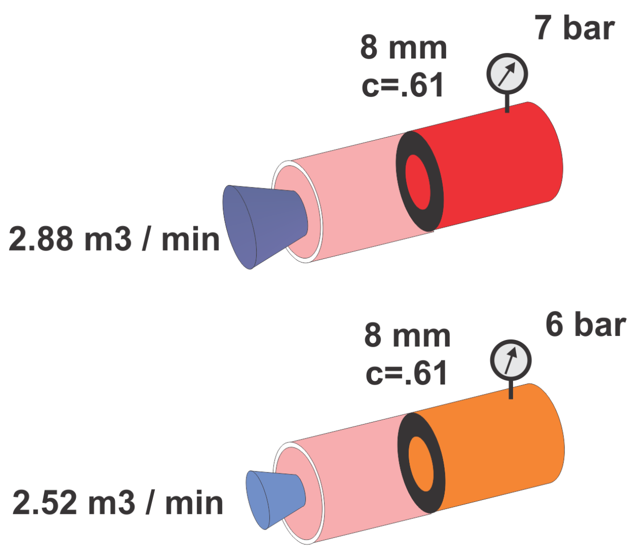



**Figure 5-2: Artificial Demand at Increased Pressure**

Any unregulated air demand will add to artificial demand as system pressure is increased above the minimum pressure to ensure proper functioning of the system. Just because an air application has a regulator on it doesn't mean it is regulated. Many times a quick check of the regulator finds that it is adjusted to maximum and is not regulating anything. A partially open valve is not a regulator. A partially open valve is simply a smaller hole in the pipe than a similar valve that is wide open. Leaks are unregulated air demands. Increasing system pressure causes every leak in the system to pass more air.

Calculating artificial demand using orifice air flow calculations (as in the example shown in Figure 5-2) if applied to an entire plant compressed air system, would assume that there is no effective pressure regulation at any point in the system. In any compressed air system, there is some amount of effective pressure regulation. Typically end uses involving automation such as robotics or automated assembly tend to be well regulated because good pressure regulation is necessary for proper performance. Many process applications of compressed air must also be well regulated. Generally, it can be estimated that 60% of air demand in a "typical" industrial plant is regulated.

## a. Artificial Demand Reduction

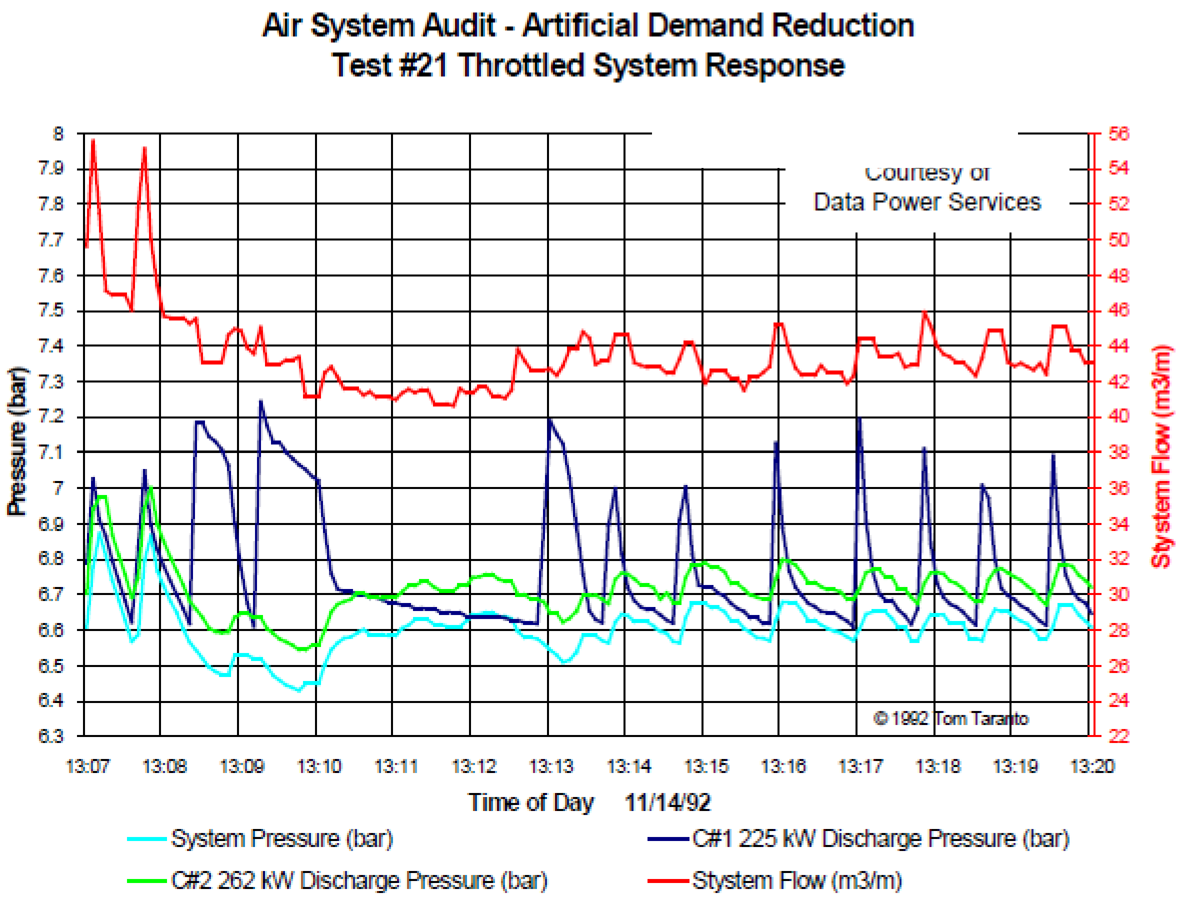



**Figure 5-3: Artificial Demand Reduction**

The data tracing above was taken during an air system audit and illustrates the benefit of pressure / flow control. In this case the pressure / flow control (a type of system control- see section b. Pressure/Flow Controllers, below) amounted to a manually operated multi-turn gate valve in the piping. Although the control is not too precise, and response time needs enhancement, system operation was improved.

The selected target pressure is 6.55 bar, since the normal compressor load cycles currently resulted in 6.55 bar minimum system pressure. System throttling began at about 13:08 hrs. Prior to pressure / flow control, system flow averaged slightly less than 51 $m^3$/min, and loads cycles were about 2 cycles per minute. Initial transients occurred between 13:09 and 13:14 hrs. The controlled system operation resulted in 42.5 $m^3$/min of controlled demand, and load cycles at 1 cycle per minute.

In the system above, even crude pressure / flow control reduced air demand by 16%, and cut compressor load cycles in half.

## b. Pressure/Flow Controllers

Flow controllers are optional system pressure controls used in conjunction with individual compressor or system controls. A flow controller does not directly control a compressor and is generally not included as a part of a compressor package. A flow controller is a device that serves to separate the supply side of a compressor system from the demand side. This may require compressors to be operated at an elevated pressure and therefore, increased horsepower, while pressure on the demand side can be reduced to a stable level to minimize actual compressed air consumption.

Storage, sized to meet anticipated fluctuations in demand, is an essential part of the control strategy. Higher-pressure supply air enters the primary storage tanks from the air compressors and is available to reliably meet fluctuation in demand at a constant lower pressure level.

A well-designed and managed system needs to include some or all of the following: overall control strategy, demand control, good signal locations, compressor controls, and storage. The goal is to deliver compressed air at the lowest stable pressure to the main plant distribution system, and to support transient events as much as possible with stored higher-pressure compressed air. Primary storage replacement should utilize the minimum compressor horsepower to restore the primary pressure to the required level.

Each compressed air system differs in supply, distribution and demand aspects that require proper evaluation of the benefits to the system of a flow/pressure controller. Additional primary and/or secondary air receivers may also address intermittent loads, which can affect system pressure and reliability, and may allow operating the compressor at the lowest possible discharge pressure and input power. The following should be considered:

- The primary function of a pressure/flow controller in a compressed air system is to stabilize pressure in the system.
- As demand changes in a compressed air system, so too will pressure.
- In a facility that is supplied by one air compressor, the change in pressure will occur relative to the control bandwidth of that air compressor. For example, if a single rotary screw air compressor with modulation control is the compressor in use, the compressor will progressively open its inlet to supply an increase in compressed air consumption. This modulation control, typically having a span of 20-70 kpa, will deliver the least amount of air at the highest pressure and the most amount of air at the lowest pressure in its span.

- This 20-70 kPa swing in system pressure may be undesirable to some facilities, especially those facilities that require very stable pressure for their instrumentation and processes.
- In a system that is supplied by more than one compressor, the various control bandwidths will overlap and cascade, and can magnify the variance in pressure.
- Pressure/flow controllers can stabilize plant pressure within tighter tolerances than compressor controls can. Typically, the set point pressure is held within ± 1% and the pressure delivered to the plant will not change regardless of the number of compressors that are on-line.
- Pressure/flow controllers require some knowledge of the system into which they are installed.
- Simply counting the total capacity of air compressors will not ensure a successful installation.
- Pressure/flow controllers respond very quickly to demand events in a compressed air system. That is why they are able to keep pressure stable.
- Some demand events can be larger in their rate of flow than the entire compressor capacity combined.
- Some demand events can change demand too rapidly for standard controls to react.
- These end-uses must be taken into consideration when sizing a pressure/flow controller or variations in pressure will still occur because of the inability to respond to system demand events.
- Some demand events are best serviced from local storage to isolate their influence from the system.
- Most events can be supplied from central storage located upstream of the pressure/flow controller faster than compressor controls can respond, and often will avoid the need to start a compressor if the storage is sized properly and the compressor controls are set up correctly.
- Pressure/flow controllers will not resolve a problem of inadequately sized piping, although distribution piping is rarely a significant cause of pressure problems in a compressed air system.

## 5.4 Inappropriate Use

Given the poor efficiency of energy conversion from electrical energy into compressed air energy; **Inappropriate Use** of compressed air is any productive work powered by compressed air energy that can be replaced with an alternative energy technology representing a more efficient conversion of energy to productive work.

Some common and typically inappropriate uses of compressed air include:

**Table 5-3: Inappropriate uses of compressed air**

| Inappropriate Use | Description of the Compressed Air Use |
|---|---|
| **Open Blowing** | Processes such as cooling, bearing cooling, drying, clean-up, draining compressed air lines, and clearing jams on conveyors. |
| **Sparging** | Sparging is aerating, agitating, oxygenating, or percolating liquid with compressed air. |
| **Aspirating** | Aspirating is using compressed air to induce the flow of another gas (such as flue gas) with compressed air. |
| **Atomizing** | Atomizing is where compressed air is used to disperse or deliver a liquid to a process as an aerosol. |
| **Padding** | Padding is using compressed air to transport liquids and light solids. |
| **Dilute Phase Transport** | Dilute Phase Transport is used in transporting solids such as powdery material in a diluted format with compressed air. |
| **Dense Phase Transport** | Dense Phase Transport used to transport solids in a batch format. |
| **Personnel Cooling** | Personnel cooling is operators directing compressed air on themselves to provide ventilation. (always inappropriate) |
| **Open hand held blowguns or lances** | Open hand held blowguns or lances are any unregulated hand held blowing and are a violation of most health and safety codes, and very dangerous. (always inappropriate) |
| **Diaphragm pumps** | Diaphragm pumps are commonly found installed without regulators and speed control valves. Those diaphragm pumps that are installed with regulators are found with the regulators adjusted higher than necessary. |
| **Vacuum Generation** | Vacuum generators are used throughout industry. Some applications for vacuum generators are shop vacuums, drum pumps, palletizers, depalletizers, box makers, packaging equipment, and automatic die cutting equipment. |
| **Vacuum Venturi** | Applications where compressed air is used with a venturi, eductor, or ejector to generate a negative pressure mass flow. When compressed air is forced through a conical nozzle, the velocity increases and a decrease in pressure occurs. Vacuum generators are used throughout industry. Some applications for vacuum generators are shop vacuums, drum pumps, palletizers, depalletizers, box makers, packaging equipment, and automatic die cutting equipment |

| **Inappropriate Use** | Description of the Compressed Air Use |
|---|---|
| **Cabinet cooling** | When first cost is the driving factor, open tubes, air bars (copper tube with holes drilled along the length of the tube) and vortex tube coolers are used to cool cabinets. Cabinet cooling should not be confused with panel purging (an explosion proof panel having an inert gas passed through it at positive pressure).<br>Cabinet cooling should not be confused with panel purging (an explosion proof panel having an inert gas passed through it at positive pressure). |

For any productive work process there are often several alternative methods the can accomplish the task. For many tasks, compressed air is the “traditional” energy source that is used.

**Table 5-4: Alternatives to compressed air energy use**

| Inappropriate Use | Alternative | Inappropriate Use | Alternative |
|---|---|---|---|
| Open Blowing | Fan, Blower, Broom, Electrically powered vacuum | Personnel Cooling | Electric Fan |
| Sparging | Blower, Mechanical Agitation | Open hand held blowguns or lances | Blower (low or medium pressure), Low pressure compressor, Electrically powered vacuum, Brush, broom or other mechanical device |
| Aspirating | Blower, Fan | Diaphragm pumps | Electrically driven pump |
| Atomizing | Blower, High Pressure Nozzle | Vacuum Generation | Electrically Driven Vacuum Pump, Centralized Vacuum System |
| Padding | Blower (low or medium pressure), Low Pressure Compressor | Vacuum Venturi | Low pressure venturi designed to be powered by a blower. |
| Dilute Phase Transport | Blower (low or medium pressure), Low Pressure Compressor | Cabinet cooling | Ventilation fan, heat pipes, liquid cooling, refrigerated cooler |
| Dense Phase Transport | Blower (low or medium pressure), Low Pressure Compressor<br>Mechanical Conveyor | Vacuum Generation | Electrically Driven Vacuum Pump, Centralized Vacuum System |

## 5.5 Irrecoverable Pressure Loss

**Irrecoverable Pressure Loss** in a compressed air system is the difference in pressure at two points in the system resulting from the interaction between compressed airflow and the fixed frictional resistance of components that compressed air is flowing through.

Productive end use air demands have an airflow requirement and minimum acceptable pressure that must be delivered to properly accomplish the productive task. At the supply side of the system air compressors must deliver compressed air at a sufficiently high pressure to overcome Irrecoverable Pressure Loss while still supplying the minimum acceptable pressure at each end use connection point in the system. Increasing the air pressure delivered at the discharge of a positive displacement compressor increases the compressor's energy consumption. It is therefore desirable to minimize irrecoverable pressure loss throughout the compressed air system.

## 5.6 Key Learning Points

1. Use compressed air only when other alternatives are not available

2. Compressed air systems should be operated at the lowest practical pressure.

3. Optimize compressor control with a properly implemented control strategy.

## 5.7 Key Energy Points

4. Eliminating inappropriate use of compressed air reduces air demand and saves energy.

5. Reducing system pressure eliminates Artificial Demand and saves energy.

6. Reducing leakage loss in the system eliminates waste and saves energy.

7. Minimize irrecoverable pressure loss and reduce compressor discharge pressure to save energy.

8. Greatest energy savings occur when the compressor control strategy optimizes the balance between supply and demand.

# 6. Distribution

Compressed air applications require a volume of air and a suitable supply pressure. The volume of air use can be considered as average air demand, or peak dynamic airflow rate. Air compressors are sized based on average air demand. Compressed air storage can be engineered to supply the peak airflow requirements.

Storage can be primary storage located in the compressor room, or secondary storage located near the end use requirements. All components located downstream of the storage receiver tank must be sized large enough to handle the peak airflow rate as it occurs.

For example, a dense phase transport system uses on average 150 l/s air demand. However, the operating cycle is 15 seconds on using compressed air, and 15 seconds off while the transport pot is refilled. The peak dynamic airflow rate used by the dense phase transport is 300 l/s.

## 6.1 Air System Point of Use Design

Consider, a clamping application requires 45,000 Newtons of clamping force. The clamp moves 250mm and must engage and disengage in 1.5 seconds. The clamp operates at 4 cycles per minute. A 300mm diameter pneumatic cylinder at 6.9 bar clamping pressure provides 45 kN of force.

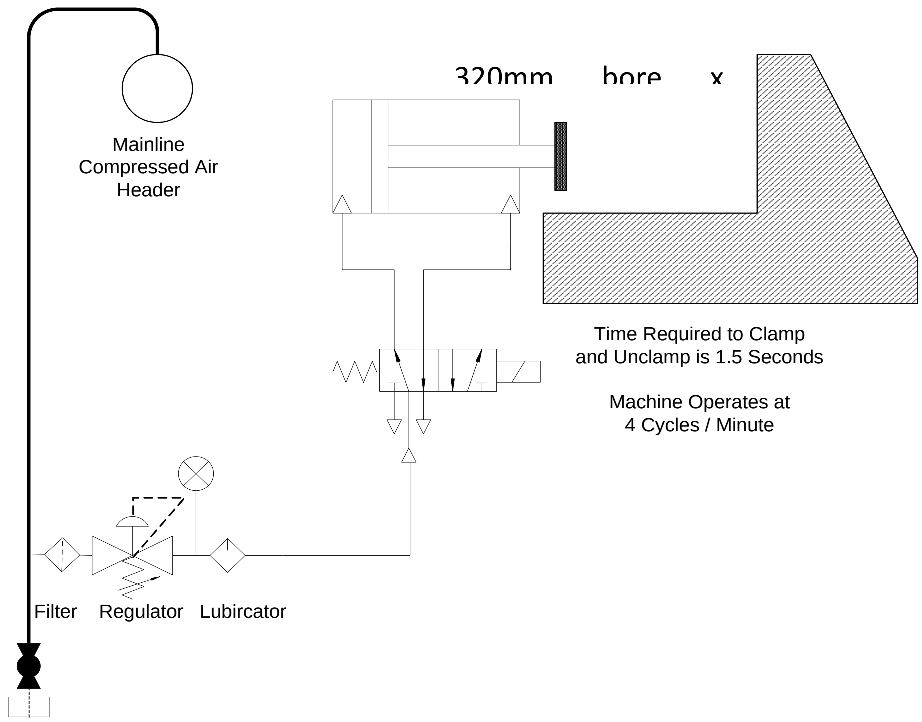



**Figure 6-1: Clamping Cylinder Block Diagram**

What is the average $m^3$ air use for the clamping cylinder shown in Figure 6-1?

**Equation 6-1: Solve for the Cylinder Volume (cubic meters)**

$$V = \frac{\Pi \ r^2 \times l}{(1000)^3} = \frac{\Pi \ (160)^2 \times 250}{(1000)^3} = 0.02 \ cubic \ meters$$

The volume of free air ($m^3$) required to pressurize 0.02 $m^3$ volume to 6.9 bar(g) is:

**Equation 6-2: Solve for m3 (free air) to Fill the Cylinder Volume**

$$m^3 = V \times \frac{\Delta P}{P_{atm}} = 0.02\, m^3 \times \frac{6.9bar(g) + 1bar}{1bar} = 0.158 m^3$$

(Note: the volume of the rod end of the cylinder is slightly less due to the volume of the piston rod).

If the cylinder strokes out and back 4 times per minute the volume must be filled 8 times during 1 minute of operation.

**Equation 6-3: Solve for Average Air Flow ($m^3$)**

$$Average\ Air\ Flow\left(m^3\right) = 8\,\frac{strokes}{minute} \times 0.158\,\frac{m^3}{stroke} = 1.264\ m^3\ \ (1\ minute\ average)$$

What size compressed air line, air valve, filter, regulator, lubricator, etc. should be used to supply the clamping cylinder?

Air Line Size ______________________________________________

Filter, Regulator, Lubricator _____________________________

Valve Size _______________________________________________

In order to properly apply the point of use piping, and other equipment, the designer must consider the dynamic peak airflow rate of the compressed air demand.

Dynamically, all of the air use occurs during the 8 times the cylinder is moved, and each move takes 1.5 seconds. The total time during which air is flowing is 8 x 1.5 sec. or 12 seconds. The rate of air demand is 1.264$m^3$ in 12 seconds or 6.32 $m^3$/min.

To solve for the peak airflow rate during one cycle of the air cylinder, given that the air use is 1.264$m^3$ / 12 seconds:

**Equation 6-4: Solve for Peak Dynamic Airflow Rate ($m^3$/min)**

$$Dynamic\ Airflow\ Rate\ (m^3/\min) = \frac{1.264\ \mathrm{m}^3}{12\ \text{seconds}} \times \frac{60\ \text{seconds}}{1\ \text{Minute}} = 6.32\ m^3/\min$$

Based on the dynamic airflow rate used by the large pneumatic clamping cylinder, what size point of use supply components do you want to use now?

Air Line Size _______________________________________________

Filter, Regulator, Lubricator _______________________________

Valve Size __________________________________________________

There is still an additional consideration when evaluating the clamping cylinder application. The peak dynamic airflow rate is 6.32 $m^3$/min and the required minimum pressure for clamping is 6.9 bar(g) (45 kN force). However, peak airflow occurs while the cylinder is moving and the necessary pressure requirement occurs when it is not moving.

"Flow Static" applications are applications where peak airflow does not occur simultaneously with the necessary point of use pressure required. The large pneumatic clamping cylinder shown in Figure 6-1 requires 1.264 $m^3$/min average, with peak airflow rate of 6.32 $m^3$/min and 6.9 bar(g) minimum pressure. The peak airflow and pressure requirement do not occur at the same time. When air is flowing the cylinder advances until the clamp engages, during that time the airflow rate is 6.32 $m^3$/min and the pressure requirement is simply the pressure necessary to move the cylinder and clamp fixture. When the clamp engages the airflow is essentially zero (leakage only) and the necessary clamp pressure is 6.9 bar(g).

Considering the demand is flow static, what will you select?

In "Flow Dynamic" applications, the peak airflow and the necessary point of use pressure must occur simultaneously. For example, if the cylinder in the application described here was used to push a steel billet rather than a clamping function, the dynamics of cylinder operation are completely changed. The force required must overcome friction and accelerate the heavy steel billet while accomplishing the movement in 1.5 seconds. Since the cylinder is now moving under load, the operation requires the necessary airflow rate and operating pressure to occur together as the pusher advances while moving the load of the steel billet.

When evaluating the pressure requirements of existing air demands, investigating the Distribution Pressure Gradient (more in Section 7.11 Distribution System Pressure Profile) and Point of Use Connections is important in order to:

1. Define peak versus average air demand, and their influence on pressure gradients and the system pressure profile.

2. Characterize air demands as Flow Static or Flow Dynamic to verify an acceptable pressure profile under actual dynamic conditions.

Demand characteristics and the peak airflow requirement ultimately determine the appropriateness of the system pressure profile. Demands should be characterized as Flow Static or Flow Dynamic. Flow Dynamic Demands are often perceived to require high pressure when in reality the pressure drop through point of use components causes extremely low pressure during peak airflow.

## 6.2 Perceived High Pressure Demands

Perceived high-pressure air demands frequently establish the minimum air pressure requirement for the system. Validating the pressure requirements is important. An accurate pressure profile of the air demand will help define the necessary supply pressure. Also important is to rule out excessive pressure drop at the point of use connection. Particularly in the case of Flow Dynamic air demands, occasional point of use pressure drop under peak airflow rate may lead to the perception that an application requires high supply pressure to function. Alternatively, decreasing resistance in the point of use piping can allow peak airflow to be delivered at the necessary use pressure without excessively high supply pressure.

Proper validation of perceived high-pressure air demands is necessary to evaluate the opportunity to save energy through the reduction of system air pressure.

Investigating Perceived High Pressure Demands is necessary to:

1. Validate the application pressure and flow requirements.

2. Evaluate the suitability of point of use piping and connection practice, particularly for Flow Dynamic applications.

3. Consider data logging point of use dynamic flow and pressure demand and documenting current application performance.

4. Evaluate solutions of connection or application modifications, dedicated storage, or pressure boosters where appropriate.

5. For valid high pressure uses, consider the impact of alternative supply configurations, and potential energy savings through reduced system air pressure in the remainder of the system.

### a. Pressure Gauges, Things Aren't Always What They Seem

The data tracing (Figure 6-2: Dynamic Pressure Tracing of Test MachineFigure 6-2) is from a pneumatically operated test machine. The air use is dynamic, which means, the work is accomplished while the air is flowing. The pressure gauge appeared to read about 6.21 bar. The machine was not performing reliably therefore higher supply pressure is required. Right? Wrong!

The perception that the machine's operating pressure is 6.21 bar is incorrect. The actual pressure as the machine cycles is 5.4 to 5.6 bar.

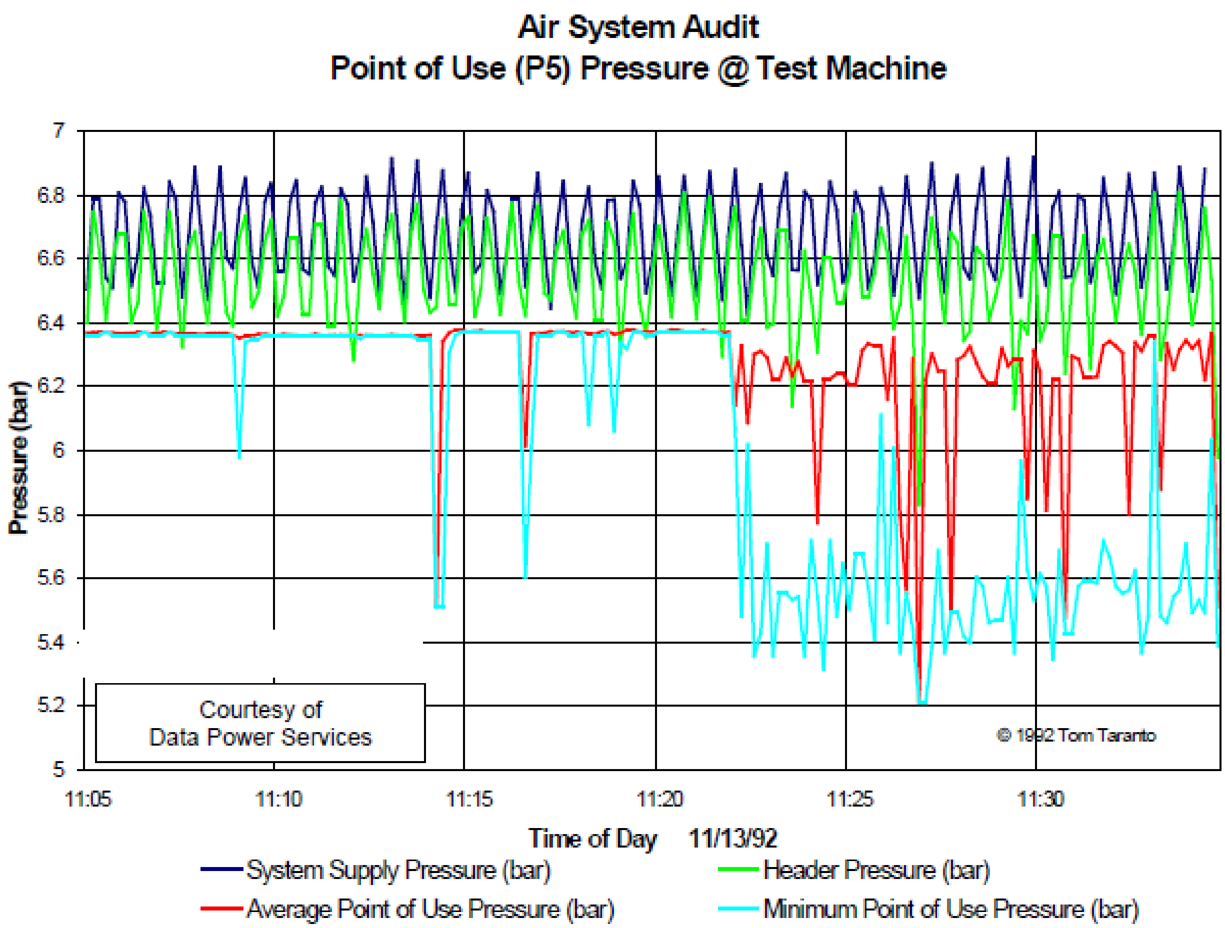



**Figure 6-2: Dynamic Pressure Tracing of Test Machine**

Mechanical damping in a pressure gauge does not allow a gauge to respond to rapidly changing dynamic conditions. Slow speed averaged data readings (Average Point of Use) also shows average supply pressure at around 6.21 bar. However, high speed pressure sampling to capture minimum pressure (Minimum Point of Use) shows dynamic supply pressure is actually closer to 5.52 bar.

Restrictions caused by the poor point of use piping construction shown in Figure 6-3, limit the dynamic supply pressure.

**Figure 6-3: Test Machine Compressed Air Point of Use Connection**

When evaluating the performance of perceived high pressure air demands, make accurate measurements of current operating dynamic pressure profiles. Review point of use connection practice and rule out restrictions that may be causing unnecessary pressure drop under peak airflow conditions.

## 6.3 High Volume Intermittent Demand Events

High volume intermittent demand events consume a large airflow rate for relatively short periods of time. The demand event is followed by a longer period of relatively low air use. These events are characterized with high peak airflow requirements and relatively low average air demand.

High volume intermittent demand events erratically affect the system pressure profile. They also create peak airflow requirements that are not characterized by average air demand.

Frequently, the entire system responds to High Volume Intermittent Demand Events. The effect of rapid airflow changes impacts the system pressure profile. In addition, compressor control signals, system supply pressure, distribution pressure gradient, and use point pressure are all briefly affected.

High volume intermittent demand events often initiate a compressor start-up. Considering that 15 to 20 seconds or more may pass for a compressor to start-up and begin delivering air to the system, a short duration event may be over before the compressor is online. The compressor response is ineffective and wastes energy. The air demand is better supported with the application of air from storage.

One common operational remedy is to increase overall system pressure. Excessive system operating pressure under normal operation to support the intermittent pressure drawdown to maintain an acceptable valley pressure can waste energy.

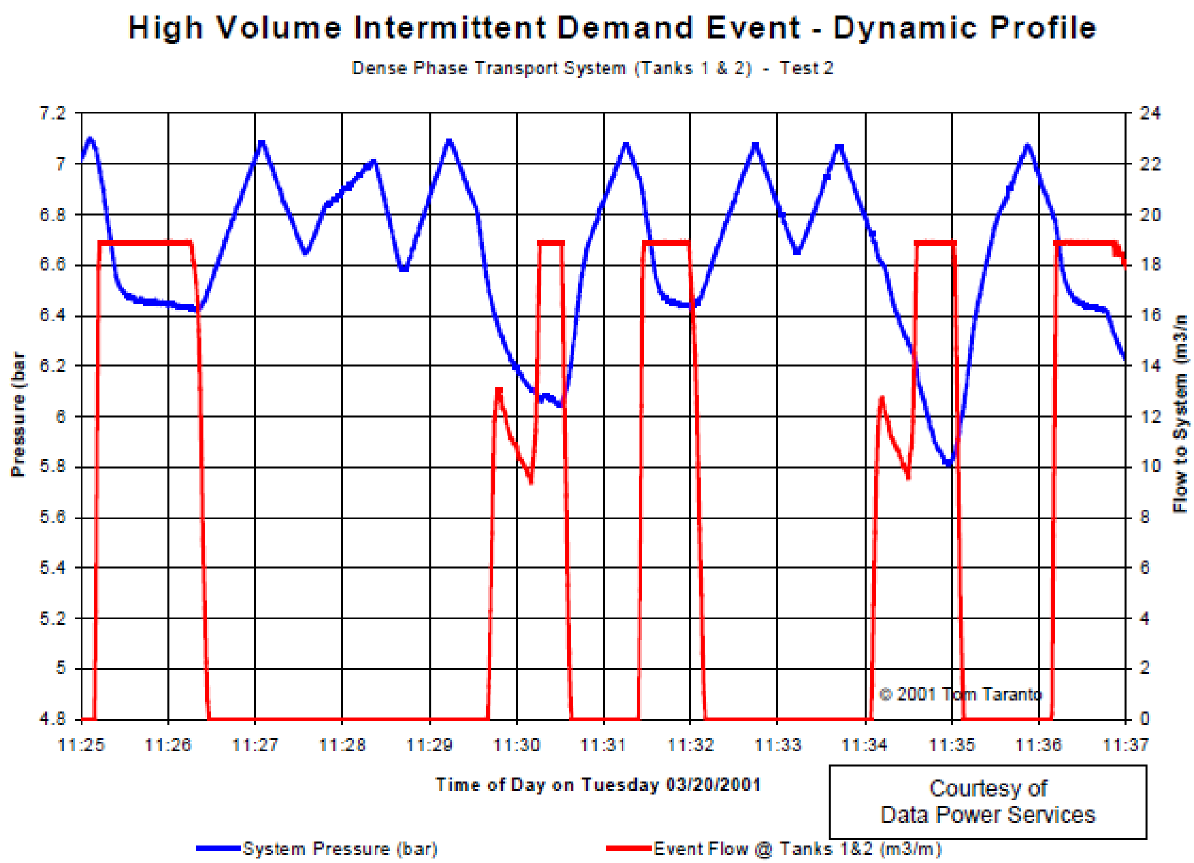



**Figure 6-4: High Volume Demand Event**

### a. Investigating High Volume Intermittent Demand Events

It is necessary to:

1. Define short-lived peak airflow rate, valley pressure, and rate of system pressure decay. Gather information necessary to calculate compressed air storage solutions.

2. Measure the duration of demand events and total air consumed.

3. Measure the delay time between demand events and the ability to refill storage during the available delay time.

4. Evaluate compressor control response and determine if compressors are running unnecessarily.

5. Consider that excessive system pressure may currently be an operational solution to inadequate air storage.

The characteristics of High Volume Intermittent Demand Events vary as widely as the applications that create them. The impact on system performance often affects everything in the compressed air system. The upset will cause control response from the air compressors. The pressure drop through treatment equipment and distribution piping will change during the surge of airflow. Other air demands throughout the entire air system will see an upset in their supply pressure. The potential solutions are as varied as the characteristics of the demand events that cause the upset.

Remember***, everything that happens in a compressed air system affects everything else in the compressed air system***. Take measures to control high volume demand events and minimize their impact to the air system.

## 6.4 Compressed Air Distribution System

The compressed air distribution system is a network of pipelines connecting the supply side of the system to the various compressed air use points on the demand side of the compressed air system. The function of distribution piping is to move compressed air from the supply to connected demands delivering the necessary compressed airflow rate and pressure to accomplish productive work.

Compressed air moves through the distribution network flowing from areas of high pressure to areas of low pressure following the path of least resistance. Every component in the system; pipelines, valves, fittings, filters, connectors, has some frictional resistance to compressed air flow. The interaction of airflow and pipeline resistance creates pressure loss.

## a. Pipe Layouts – Point of Use Piping

Point of use piping delivers compressed air from the main or branch header to the application point of use where pneumatic energy is applied. The energy delivered to the use point is a function of the airflow and pressure when the conversion from pneumatic to mechanical energy takes place. It is common to find pipe layouts that result in 1.4 to 2 bar or more pressure loss between the header and use point.

The result is a poor performing or unreliable pneumatic application. Frequently, operators observe that the application seems to works better when the air pressure is higher. Intuitively the application performs poorly under the condition of low pressure. What is the solution?? – Increase the air system pressure. The symptom goes away but is the illness cured?

Frequently, the root cause of poor performing applications is pressure loss in the connection piping. Which of the piping layouts shown would you expect to perform best? One with undersized filters, rubber hoses, and a redundant refrigerated dryer; or a system with properly sized filters, permanent piping, and no needless redundant components?

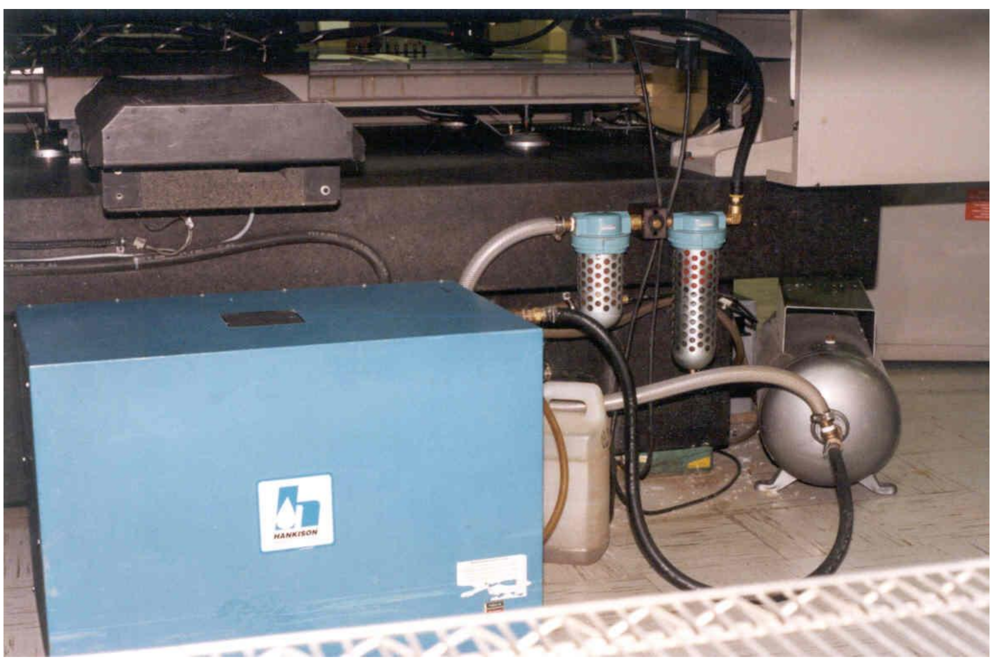


**Figure 6-5: Examples of Point of Use Piping**

## 6.5 Key Points – Air System Design

1. Identify dynamic airflow conditions of average –vs- peak airflow.
2. Classify air demands as Flow Static and Flow Dynamic.
3. Point of use connection practice has a significant effect on applications performance.
4. Review perceived high pressure air demands to validate their pressure requirements.
5. Pressure gauges have slow response to pressure changes. Pressure transducers and high-speed sampling may be required to capture pressure dynamics.

## 6.6 Key Energy Points – Air System Design

6. Supplying higher end use pressure requiring higher discharge at the compressor(s) increases compressor power (kW) by 6% per bar.
7. Poor piping design with excessive flow restriction can create a perception that the end use air demand requires higher pressure than is actually necessary.

8. Minimize the use of hose for connections. Hose has much smaller ID size (higher pressure drop) than similar diameter pipe.

9. Where hose must be used select the hose size based on the inside diameter and peak airflow rate. Avoid the use of hose, hose barbs and pipe clamps. They are very restrictive and frequently develop leaks.

10. Do not use redundant point of use dryers, filters, etc. as each component represents additional pressure drop.

11. Avoid over filtration, maintain an appropriate compressed air cleanliness class for the application requirements.

12. When components are improperly sized for average airflow rate rather than peak airflow requirements, system is pressure is often increased to accommodate the improperly sized components.

13. Size all connection equipment to the actual dynamic conditions associated with the application. Account for the peak airflow rate that must be supported, do not size equipment based on average airflow rate.

## 6.7 Balancing the Supply to Demand

In a compressed air system when supply exceeds demand the system's pressure will increase. Conversely when demand exceeds supply pressure will decrease. The rate of pressure increase or decrease depends on how large the supply excess or deficit is and the volume of the system. The larger the system volume the more slowly pressure will rise and fall given the same amount of supply excess or deficit.

When the compressed air supply and demand are balanced (the airflow rate entering the system is equal to the airflow rate leaving the system) pressure in the system will remain constant.

## 6.8 Stabilize System Operating Pressure

All compressed air systems have some minimum pressure that is necessary for proper function of the system. Frequently that pressure is not defined. However, as long as the pressure is above the minimum, for example 5.52 bar, there are no complaints. Running the compressors at 7.58 bar maintains 6.9 bar in the system and operations are satisfied. If 5.9 bar is acceptable, 6.9 bar is an improvement to operations. Correct?

Wrong! More pressure is not better. Supplying 1.0 bar more pressure to the system will force the system to consume 12% more air flow (if there is no effective pressure regulation within the system). It also causes the compressors to consume 7% more power for each 1 bar increase in discharge pressure. Total energy waste in the system is 19%.

As the system's pressure is increased all unregulated leakage and air demands will consume an increased amount of compressed airflow. It is common practice to operate compressed air systems at a higher pressure than necessary. This leads to increased compressed air demand in the system. This increased air demand is called Artificial Demand (refer: 5.3 Artificial Demand - Increased Pressure Increases Air Demand).

To eliminate waste to artificial demand it is desirable to operate the system with the demand side pressure set to the lowest optimum pressure necessary to support production requirements. Any higher pressure will create some amount of artificial demand; the greater the pressure, the greater waste to artificial demand.

In a system the overall waste to artificial demand depends not only on how high the demand side pressure is but also depends on how much of the compressed air use is regulated. For example, if the system pressure is 1.0 bar above the optimum target pressure, and 50% of the compressed air use is unregulated, then the component of artificial demand is approximately 6.0%.

Measurements made during a compressed air system assessment define the plant consumption profile and average air demand swings throughout the day. Peak air demand for system surges and transient events is also defined. Plant pressure instability is recorded and an initial target controlled system pressure is selected.

Action items typically include implementation of controls to balance the energy supply to the system with the compressed air demand. By balancing supply with demand, system pressure is stabilized at some selected initial target supply pressure. Over time the system should be tuned by lowering demand side pressure with the objective to ultimately control the system supply to the lowest acceptable pressure. The supply side pressure set point and compressor controls should then be adjusted to optimize their operation. There are ultimately 3 objectives.

1. Minimize the cost of generating compressed air.

2. Control air demand and reduce artificial demand.

3. Create controlled air storage to supply peak demand.

There are many benefits to initial stabilization of system supply pressure. A portion of Artificial Demand in the system is immediately controlled. Compressor control response is improved and load cycles are smoother and frequency reduced. Useable air storage of the existing receiver volume is improved by establishing differential pressure. Stored energy supplies dynamic, short-lived events and stabilizes system supply pressure.

## 6.9 Engineer Primary Storage Systems

Primary stored air volume upstream of the pressure / flow control is engineered to provide the necessary useable air volume from storage. Stored energy is an important component of system stability. When a machine or process starts, there is a rapid increase in air demand (energy) required by the system. The storage system provides immediate energy. In the case of large demands that require capacity increase from the compressors, storage provides energy during the time it takes for compressor controls to respond.

Receiver volume alone does not create useable air storage. Every compressed air system has a target supply pressure. System supply falling below the target pressure, compromises reliability. Therefore, storage pressure must be maintained higher than the target pressure. The useable air (energy) in storage depends on the receiver volume and the pressure difference between receiver pressure PS and the system supply target pressure PT.

However, if supply pressure is above the target energy waste through artificial demand, can be as much as 12% per bar (if all air use is unregulated). The target pressure should be the lowest optimal pressure to properly supply productive air demands.

When evaluating controlled storage, equipment specification should account for capacity to meet surge demands and transient events as defined in the Plant Demand Profile (see Module 8). Time-based response characteristics of the pressure / flow control depend on the nature and magnitude of airflow change during surge demand. In addition to dynamic response, the pressure / flow control must also maintain a controlled pressure differential between storage receivers and the system supply target pressure. The pressure / flow control must be capable of high airflow rate while maintaining control at a low pressure differential. Stable operation is necessary during low air demand, with a high degree of capacity turn down.

### a. Storage; A Lake – Vs. A Reservoir

The example illustrated below shows, an air receiver is like a lake, while air storage is more like a reservoir.

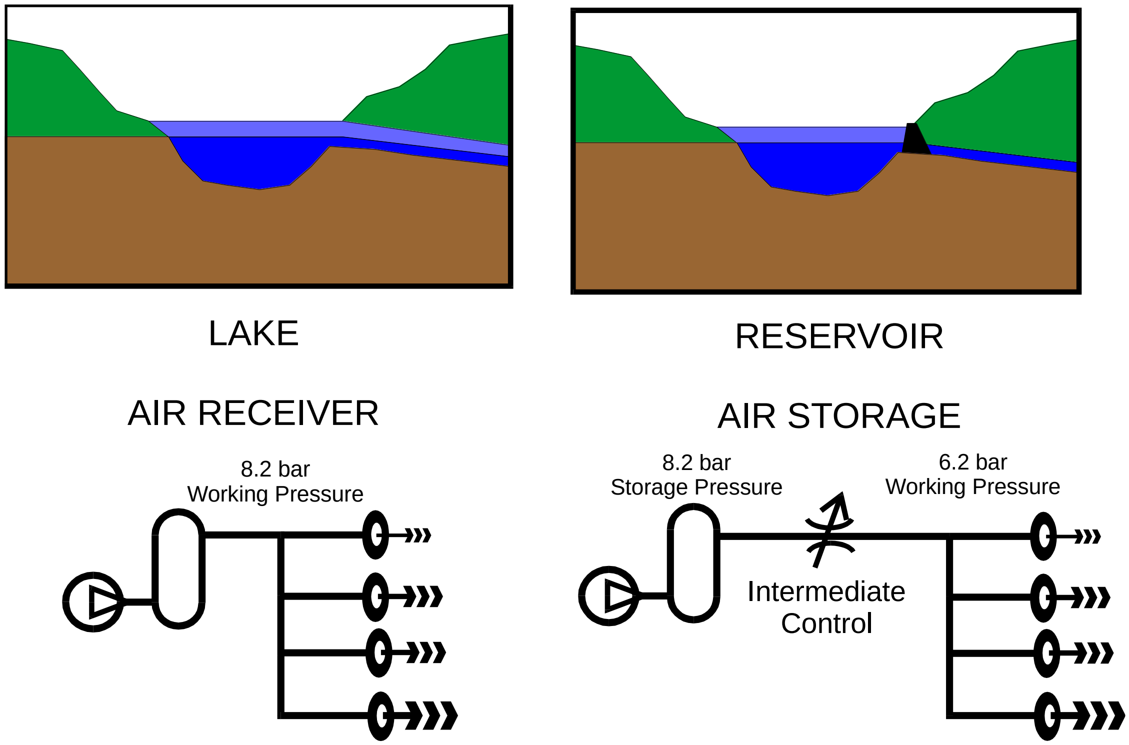



**Figure 6-6: Pressure / Flow Control of Storage Differential Pressure**

As the lake level rises, water runs out faster and everything downstream floods. In most compressed air systems compressor controls are set higher than target pressure. The receiver is maintained at this high pressure and useable air is available from storage. But the higher pressure causes everything downstream to flood, resulting in 13% waste for every 1.0 bar through artificial demand.

If on the other hand a dam is built at the outlet of the lake, the lake becomes a reservoir. As the water level rises, the outflow from the lake is controlled and the downstream flood is prevented. In a compressed air system, the pressure / flow control is the dam, and flood control gate. As receiver pressure increases, the pressure / flow control throttles the outflow of air and prevents downstream pressure rise. Artificial demand is prevented.

Industrial compressed air systems use the least amount of air when the pressure is lowest. However, when the receiver pressure is equal to the system target pressure, no air is available from storage.

### b. Calculating Compressed Air Primary Storage

Useable compressed air available from primary storage is a function of the storage volume and pressure differential between air pressure in storage (PS) and the target supply pressure (PT) for the system. The formula to calculate useable air for Primary Storage is shown below.

**Equation 6-5: Storage Capacity Calculation**

$$V_a = V_s \times \left(\frac{P_{max} - P_{min}}{P_{amb}}\right)$$

Where:

$V_a$ = Useable compressed air in storage

$V_s$ = Total volume of storage system

$P_{max}$ = Maximum storage or receiver pressure (cut-out pressure)

$P_{min}$ = Minimum storage or receiver pressure required (cut-in pressure)

$P_{amb}$ = Absolute ambient air pressure

As the formula demonstrates, changes in actual volume of the storage system OR changes in pressure differential can create useable storage. It also demonstrates that if there is no pressure differential, there is no useable storage.

You will note that the useable air in storage is directly proportional to the storage pressure differential. With 6 cubic meters of receiver volume; @ .33 bar = 2 $m^3$ storage; at twice the pressure differential, .66 bar = 4 $m^3$, twice the storage. At 2 bar, six times the original pressure differential (.33 bar x 6), there is six times (12 $m^3$) the useable air in storage. If the compressor control set point is equal to the target system supply pressure (PT) there is no useable air available from storage. It doesn't matter how large the receiver volume is, without pressure differential, there is no useable air storage.

Tuning the system is adjusting compressor control set points to optimize useable air storage. Optimal air storage depends on system dynamics and peak airflow demands. Creating storage costs money because increasing generation pressure increases the power investment in generating compressed air by 6% for each 1.0 bar increase. However, insufficient storage leads to system instability and running excess generation capacity to meet peak load (refer Figure 6-8).

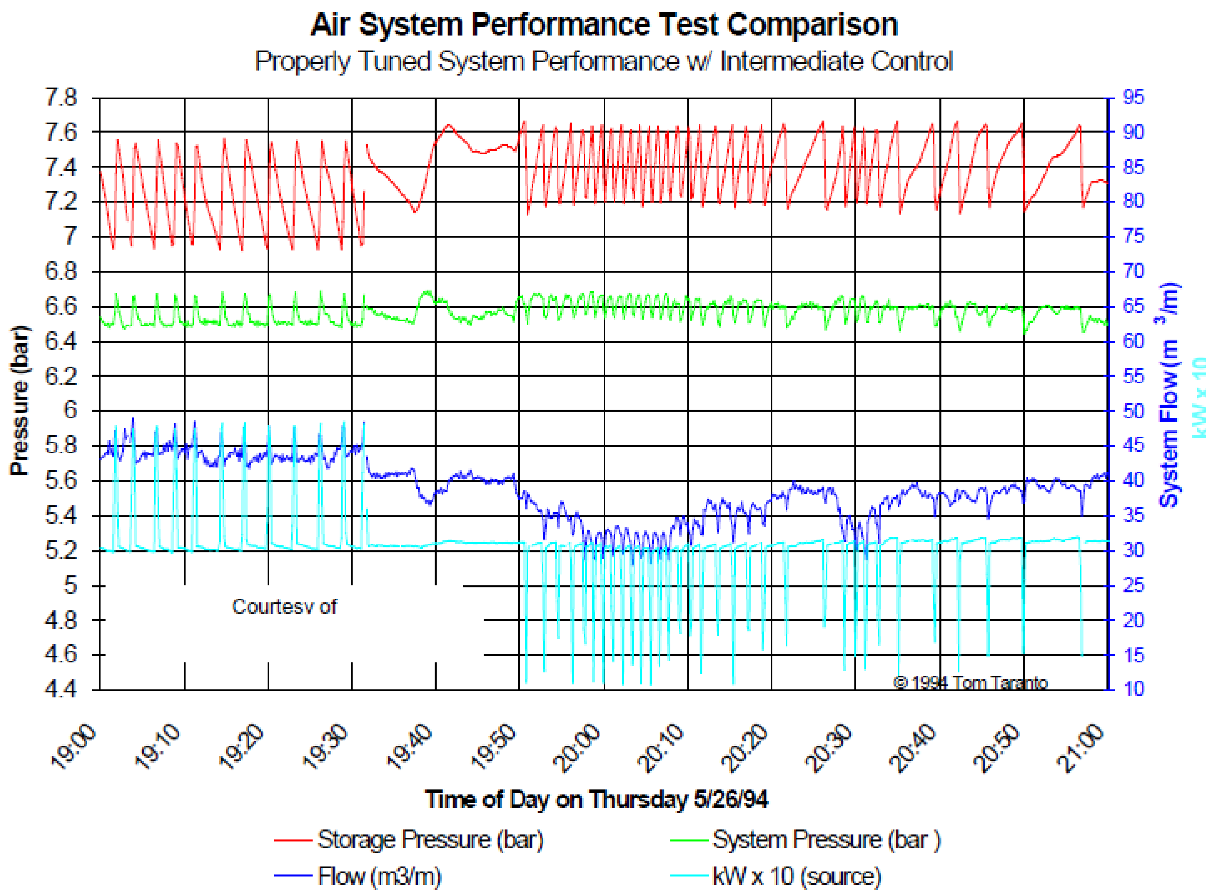



**Figure 6-7: Properly Tuned system with Pressure / Flow Control**

In Figure 6-7 above is a data tracing of a properly tuned system that provides between 0.34 and 1 bar storage differential, with a pressure / flow control set for target system supply pressure of 6.55 bar.

The chart below, Figure 6-8 shows measurements taken with the system improperly tuned. The pressure / flow control was by passed, the compressor set points were lowered to provide for source control without any storage pressure differential. The data below are considerably different from the properly tuned system.

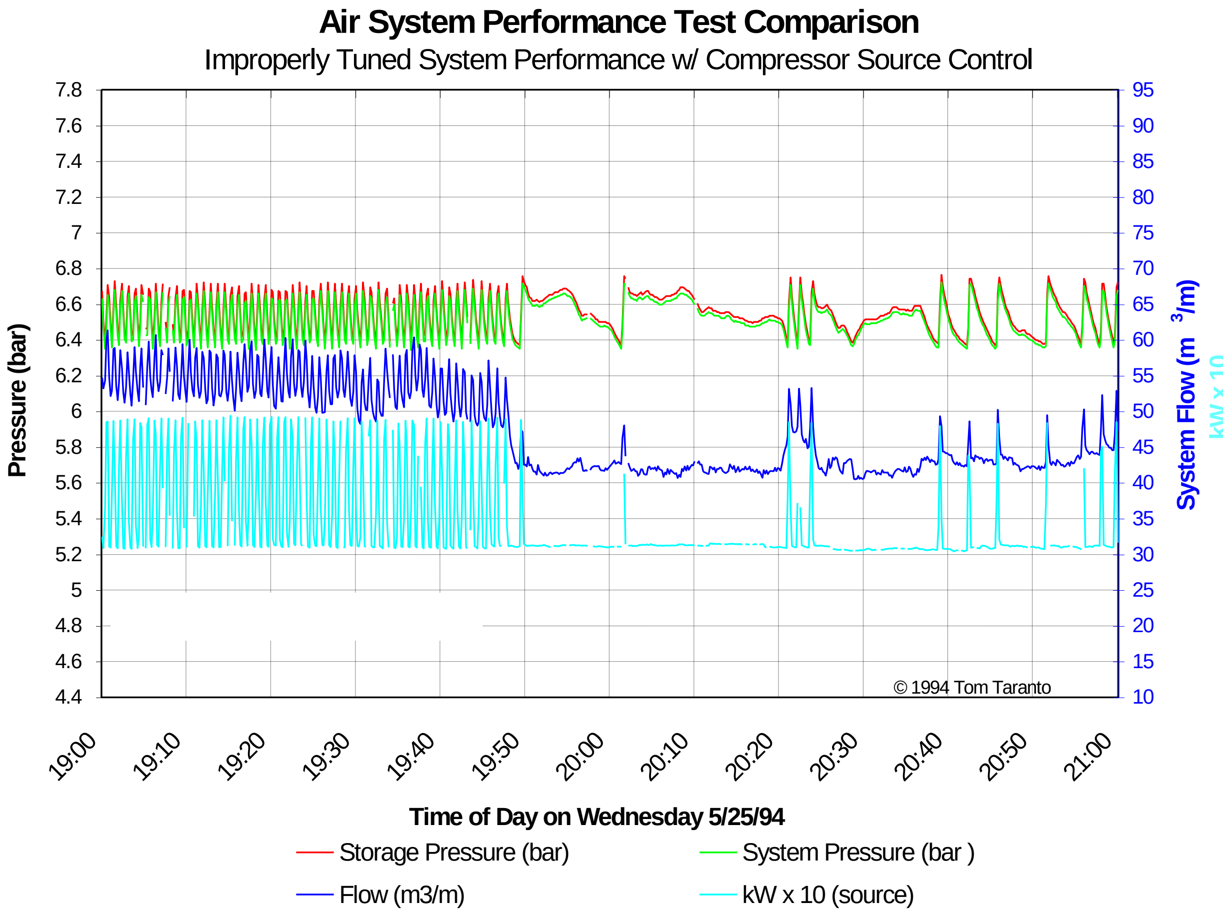



**Figure 6-8: System Performance without Pressure / Flow Control**

The improperly tuned system operating without pressure / flow control and storage pressure differential is poor performing. The system target pressure is less consistent. The trim compressor is operating with very rapid load cycles. The system is driven by compressor load cycles, and air demand is increased. The base load compressor is at 100% capacity all of the time. Finally, the lack of useable air from storage forces the trim compressor to operate continually to meet occasional air demand peaks. The improperly tuned system suffers poor performance and wastes energy.

## 6.10 Key Learning Points – Balancing the Supply and Demand

1. Stabilize system operating pressure.

2. The amount of energy in storage depends on storage volume and controlled pressure differential.

## 6.11 Key Energy Points – Balancing the Supply and Demand

3. Elevated air pressure increases compressed air demand at leaks and unregulated air demands.

4. Leakage can be reduced by controlling to a lower system pressure.

5. Artificial demand is a component of any unregulated leak or air demand.

6. Target pressure should be the lowest optimal pressure to supply productive air demands.

7. Air storage should be designed to supply surge demands, satisfy events defined in the demand profile, and improve compressor control response.

# 7. Pressure Profile

A compressed air system's pressure profile is a graphical description of compressed air pressure at various locations within the system. Pressure profile performance affects many aspects of air system operation and efficiency. Most obvious is the pressure at the end use point which must be acceptable to meet production requirements. The pressure profile should be measured at several important locations in the compressed air system.

Typical pressure measurement locations used to create a system pressure profile.

- Compressor maximum working pressure (MWP)
- Compressor control range
- Treatment equipment pressure drop
- Pressure differential reserved for primary storage

- Supply header pressure to the system
- Distribution header pressure in one or more demand side locations
- Point of use connection pressure
- End use pressure (not shown in the profile above)

The pressure profile chart below shows the air compressor's maximum working pressure (MWP), 9.0 bar. The compressor's operating control range is 8.0 to 8.7 bar. Pressure drop through treatment equipment, 0.5 bar thereby determines the header pressure leaving the supply side which is normally 7.5 bar.

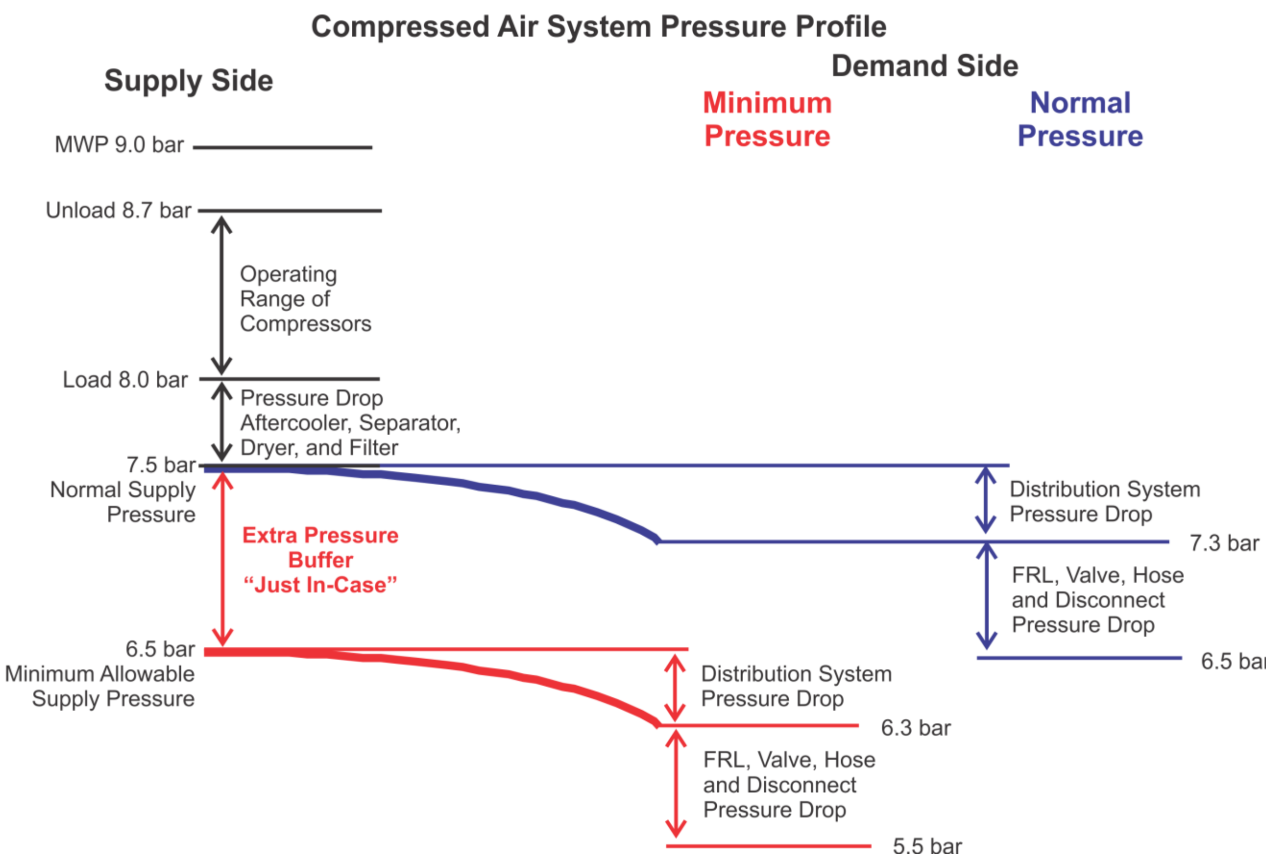



**Figure 7-1: Pressure Profile for Minimum and Normal Operating Pressure**

The minimum allowable supply pressure is 6.5 bar which will support the 5.5 bar end use pressure requirement. This allows 1 bar of extra pressure "just in-case" there is an upset to the system. It is common to set the operating pressure higher than the lowest pressure necessary to support the production process.

During normal operation the demand side pressure profile shows 2 bar pressure loss in distribution piping. The result is a normal distribution header pressure of 7.3 bar and 0.8 bar pressure loss in end use piping. The normal operating end use pressure is 6.5 bar for a requirement of 5.5 bar.

In fact the 1.0 bar extra pressure creates primary storage differential given that the lowest optimum target pressure for the supply header is 6.5 bar. (The lowest optimum target pressure is the minimum supply header pressure necessary to properly support production requirements.)

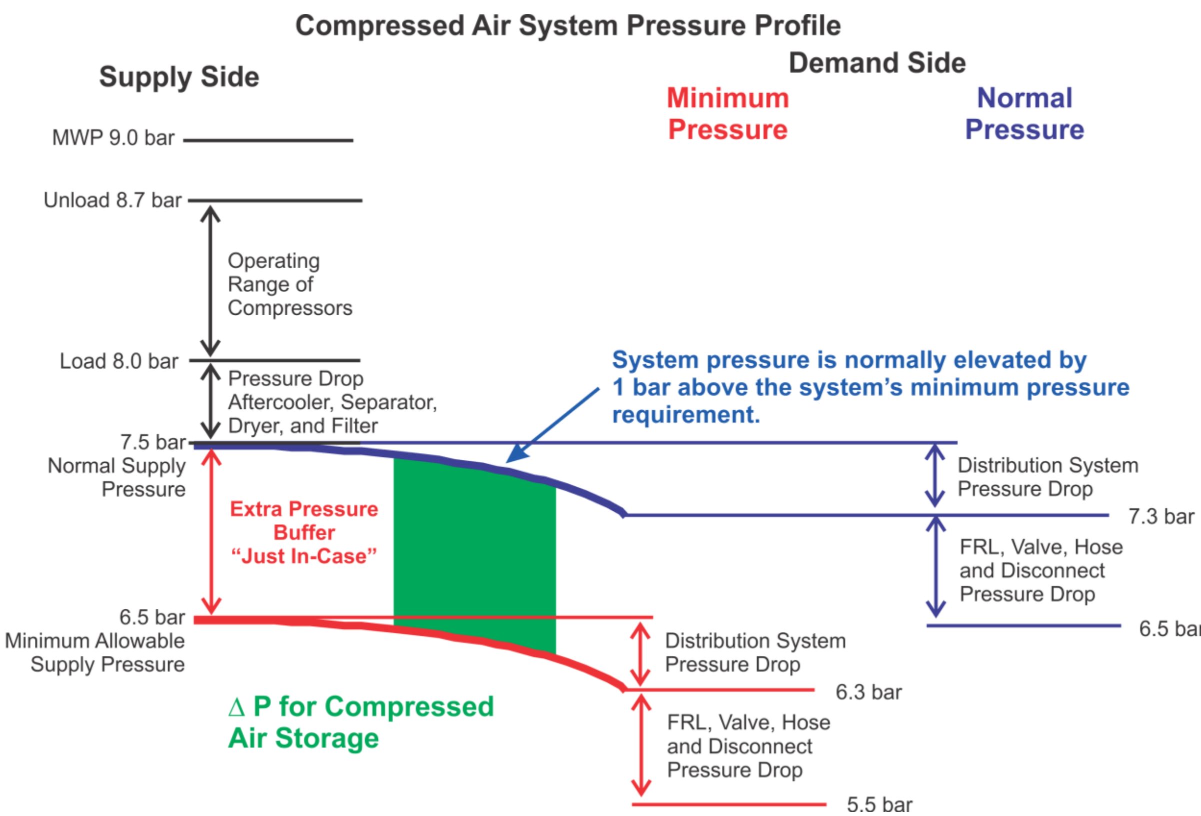



**Figure 7-2: Pressure Profile showing Differential Pressure for Storage**

While the 1.0 bar normal supply pressure is necessary to create useable storage, it also elevates the entire system pressure adding to artificial demand.

There is further pressure loss between the use point connection and the end use point. The end use pressure is the point at which compressed air energy is converted to productive work. It is the pneumatic cylinder, rotary actuator, gripper, nozzle, etc. Most often the pressure loss between the use point connection and the end use pressure is a result of design decisions made by the OEM that supplied the pneumatically powered equipment.

The pressure profile of an existing system is established by measuring the system's operating pressure. Data logging is often necessary to understand how the system's pressure profile changes throughout time. Data loggers are normally deployed in multiple locations. By synchronizing the time clock of multiple data loggers dynamic changes in the pressure profile can be assessed.

## 7.1 Practical application of pressure profiles

A compressed air system's pressure profile has many effects on system operation and efficiency. There are many practical considerations that the compressed air system engineer must evaluate. The lowest optimum supply header pressure depends on the required end use pressure, and pressure loss in the distribution and point of use connection piping. The supply side pressure profile is limited by the maximum working pressure of installed compressors. The control pressure range, pressure loss through treatment equipment, and primary storage pressure allowance must result in pressure equal to or greater than the lowest optimum supply header pressure.

To optimize the system's performance the compressed air system engineer must determine target pressure for the supply header. The target supply header pressure should be the lowest optimum pressure necessary to support production requirements. Operation at excessively high pressure wastes energy.

- Reducing System Pressure Decreases Energy Use
- Power at the air compressor drops by 6% per bar of pressure reduction (for positive displacement compressors).
- Air demand in the system drops by 6 % to 12% per bar of pressure reduction (assuming 50% to 100% of air use is unregulated).

The target pressure should be selected only after assessment of the pressure profile and elimination of excessive irrecoverable pressure loss. As a result of poor application of hose, fittings, filters, regulators, etc. at the point of use connection; it is common to find pressure loss of 1 bar(g) to as much as 3 bar(g) or more. Acceptable pressure drop should be less than 0.2 to 0.5 bar(g).

Before establishing a compressed air system's Target Supply Header Pressure, Irrecoverable pressure loss in the system should be minimized.

There are two basic types of pressure differentials in a system.

1. First is *irrecoverable pressure drop*- for example, through a filter- which is an energy loss to the system.

2. Second is *recoverable pressure differential,* such as the increased pressure of an air storage receiver. There is increased energy cost to the system when compressed air energy enters storage. However, this is recoverable energy that will be used as system demand requires stored energy.

## a. Air system components with irrecoverable pressure drop

- Pipe and pipe fittings, tees, elbows, valves, etc.
- Filter housing and filter element (finer filtration = higher pressure loss).
- After-coolers, air dryers, and moisture separators.
- Hose, fittings, quick disconnects, point of use filters, and lubricators
- Orifices, needle valves (restrictors), speed controls, tubing and fittings

## b. Air system components with recoverable pressure differential

- Air receivers, primary, secondary, point of use
- The recoverable energy is proportional to the receiver's operating pressure differential (final pressure minus initial pressure).
    - Receivers have a small component of irrecoverable loss through the piping nozzles in and out of the receiver.
    - Piping has a small component of recoverable energy. Pipe volume is typically small compared to total receiver volume. Systems with stable operation have a minimal pressure differential in the piping; it is desirable for distribution piping to operate with consistent pressure.

## c. Air system components with both recoverable pressure differential and irrecoverable pressure loss.

- Pressure regulator
    - Recoverable pressure differential; the pressure increase available if the regulator is adjusted from its set point to the highest pressure possible.
    - Irrecoverable pressure loss includes the off-set pressure; the pressure differential necessary to move the regulator element to a position where it is in control. Additional irrecoverable pressure loss is associated with the airflow rate through the regulator and resultant pressure loss when the regulator is fully open.

- Pressure/flow control

  - Recoverable pressure differential- the storage pressure differential (final pressure minus initial pressure) available to the system while maintaining the target set point pressure at the flow control outlet.
  - Irrecoverable pressure loss includes control pressure differential- the pressure differential necessary to move the flow control element to a position where it is in control. Additional irrecoverable pressure loss is associated with the airflow rate through the flow control and resultant pressure loss when the flow control is fully open.

An optimized pressure profile design minimizes irrecoverable pressure loss, and defines the minimum use point connection pressure. The compressor operating pressure at the high end of the pressure profile should be as low as possible while supporting the system's required pressure differentials.

## 7.2 Pressure Profile Design Criteria

Compressed air system pressure profile optimum design targets the objective to:

1. Operate compressor controls in as narrow a pressure band as possible while allowing:
   a. Unneeded compressors to automatically shutdown.
   b. All compressors, except one, to operate at full load capacity.
   c. Only one compressor to provide trim capacity, selecting the most efficient part load capacity control available.
2. Operate compressor discharge pressure at the lowest possible pressure.
3. Establish the delivered use point pressure at the lowest optimum pressure necessary to support productive air demand.
4. Create pressure differential (P final minus P initial) to create the necessary compressed air energy storage. Energy storage should serve normal demand events and cover permissive start-up time of reserve compressor capacity.
5. Use energy storage to prevent additional air compressors from starting in response to short duration peak demand events.
6. Minimize irrecoverable pressure loss throughout the system.
7. Control recoverable pressure differential of primary storage to eliminate artificial demand.

8. Control supply header target pressure to the lowest optimum pressure while accounting for irrecoverable pressure loss through distribution, and point of use piping.
9. Apply pressure regulation at use points where recoverable pressure differential is available. Eliminate pressure regulators that are set at maximum.

## 7.3 Supply Side Pressure Limits

A compressed air system must work within defined pressure limits. Normally, the upper limit is determined on the supply side and the lower limit on the demand side of the system.

Supply side upper limit is the maximum working pressure (MWP) of the system component with the lowest rated working pressure. For example, in a system with compressors rated for a maximum working pressure of 10 bar, and air receivers rated for 8.6 bar, the system pressure would be limited to 8.6 bar maximum.

Note that while an air compressor's maximum rated working pressure is the practical limit for the high end of the system's pressure profile, operating a positive displacement compressor at increased pressure increases the compressor's energy consumption. Therefore, it is desirable to operate compressors at the lowest possible pressure discharge pressure while supporting an optimized pressure profile.

Demand side lower limit is usually the minimum acceptable pressure to properly support productive air demands.

Both supply and demand side pressures should be operated at the lowest optimum pressure required by the system. Increasing supply side pressure increases the power used by compressors. Increasing demand side pressure increases the airflow consumed by the system as all unregulated air demands and leaks consume more air at higher pressure.

## 7.4 Supply Side Pressure Upper Limit

### a. Air Compressor Maximum Working Pressure (MWP)

Air compressor manufacturers provide performance data for every model of air compressor. Included in this data is MWP for the compressor package. All compressors have safety relief valves to protect from over pressure operation. Additionally, there is a motor overload safety shutdown which may activate when the compressor is operating above rated pressure. However, it is not desirable to rely on these safety devices for normal operation. The compressor and system controls should normally operate air compressors at or below their MWP rating.

There are two over pressure safeguards in the compressor package:

1. Internal components i.e., piping, coolers, oil sump / separator; of the compressor are pressurized. Those components are protected from over pressurization by a safety relief valve.
2. Operating the compressor at higher pressure increases the kW input and the power drawn on the drive motor. The motor is protected with electrical overload safety devices that shutdown the compressor if the motor is overloaded.

When operating multiple compressor systems there are frequently compressors with different MWP ratings. In that case, compressors with lower MWP can operate as trim capacity compressors and must be set to unload below their MWP rating. If this is the operating strategy to be used, it is necessary to ensure that the lower pressure compressors' pressure vessels and safety relief settings are safe for operation at the MWP of the highest pressure rated compressor(s) in the system.

## b. Modulation Control Compressor Maximum Working Pressures

For air compressors using modulating style control there are often two maximum working pressure ratings specified. The first is MWP which considers the pressure vessel and safety relief setting of the compressor. The second is maximum full flow working pressure which is related to the drive motor power rating. The shaft input kW of a compressor is a function of both the compressor's discharge pressure and the amount of delivered airflow.

Full Flow operation results in compressor power increasing as discharge pressure increases. Based on the drive motor's maximum rated power there is a pressure limit where further pressure increase at full rated airflow would cause a motor overload condition. This point is the maximum full flow working pressure rating. At this point, the compressor controls should cause the inlet valve to modulate closed which reduces delivered airflow and also reduces input power.

While the compressor is operating within its modulation range, the relative power decrease due to less delivered airflow allows the compressor to operate at higher discharge pressure without overloading the drive motor. The compressors MWP rating is given assuming the controls are modulating to minimum delivered airflow.

## c. Maximum Working Pressure of Other Components

Other components in the system such as air receivers, filters, dryers, drain valves, etc. must be evaluated as to their MWP rating. All components must be protected from overpressure operation. Therefore, the high pressure limit for the system's pressure profile is the rating of the component having the lowest MWP.

The compressed air system engineer must assess the allowable working pressure of every component part of the compressed air system and ensure the remains safe during all operating scenarios.

## 7.5 Supply Side Low Pressure Limit

As stated above, the low pressure limit for system operation is normally a function of the required demand side pressure. However, it is important to consider the performance rating of all equipment and the effect of low pressure operation.

### a. Minimum Pressure Rating – Air Velocity

For a constant mass flow rate ($Nm^3$/min) of compressed air, the air velocity through component parts of the system will increase as pressure decreases. This is due to the expansion of air volume with decreasing pressure. Two supply side components most often affected by high compressed air velocity resulting from low pressure operation are coalescing filters and air dryers.

Coalescing filters bring together small aerosols and droplets of lubricant as they pass through the filter media. At the outside surface of the media, a drop is formed that falls out of the compressed air stream by gravity. If the compressed air velocity is too high, the drop of lubricant formed may become re-entrained in the air stream and carried downstream. The air / lubricant separator element found in lubricant injected air compressors is a coalescing style filter. Therefore, compressor rating data will specify a minimum working pressure rating. It is important to operate at or above the compressor's minimum rated working pressure to prevent excessive lubricant carry-over from the compressor.

Air dryer performance, particularly desiccant style units, is also affected by compressed air velocity. High velocity resulting from operation below the minimum rated working pressure can reduce the contact time of compressed air in the dryer and compromise dew point performance. For example, an air dryer operating at constant mass flow rate ($Nm^3$/min) and rated at 7 bar working pressure will experience an air velocity increase of about 20% when the dryer operating at 5.5 bar. Air dryer manufacturers have performance rating correction factors for operating at other than standard design conditions. Consult the dryer manufacturer if an air dryer will be operating below the standard rated pressure.

Performance of air system components such as after-coolers, particulate filters, moisture separators, and others can also be adversely affected by lower than design operating pressure.

## 7.6 Demand Side Pressure Limits

### a. Demand Side High Pressure Limit

High pressure limit of the compressed air system pressure profile is usually determined by MWP rating of supply side equipment. However, it is possible that a particular component or application on the demand side has a MWP limit lower than the maximum operating pressure of the system. In this case, proper protection of the low pressure component or application must be installed. This protection may include pressure reducing regulators, over-pressure safety-relief valves, blow-out plugs, flow restrictors, velocity fuses or other components. The compressed air system engineer must provide proper protection ensure safe operation.

### b. Demand Side Low Pressure Limit

Compressed air systems have a wide variety of compressed air demands that must be properly supplied for efficient operation of production equipment and processes. These air demands typically have different compressed airflow and pressure requirements. If a particular compressed air demand is interrupted repeatedly, it is often a consequence of low supply pressure to the use point. This results in the perception that the air system's pressure is too low and often leads to increasing the operating pressure of the compressors and resultant pressure increase across the entire system.

In many compressed air systems, one or two isolated compressed air use points that may account for a small fraction of the total airflow demand of the system are used to establish the demand side low pressure limit of the system's pressure profile.

It is common practice to increase overall system pressure to satisfy small air demands which require (or are perceived to require) higher pressure than the majority of air demands. If these air demands can be modified to operate at lower pressure energy savings may be possible.

## 7.7 Key Learning Points – Pressure Limits

Both the supply and demand sides of a compressed air system have pressure limits within which the system must operate.

1. Pressure limits form the operating envelope of the pressure profile
2. Supply maximum working pressure (MWP) is the high limit of the pressure profile
3. Demand side point of use pressure target is the low limit of the pressure profile
4. Consider minimum design pressure (velocity) rating of supply components
5. Protect demand side components from exceeding their MWP

## 7.8 Point of Use Pressure Profile

The compressed air system engineer should assess various use point applications and determine the actual lowest optimum working pressure to appropriately supply productive air demands. Often the perceived pressure requirements are much higher than the actual use point pressure necessary for proper operation of the connected air demand.

There are two physical locations that are sometimes referred to as the point of use. One is the supply connection to a machine or process and the second is the actual pneumatic device i.e. pneumatic cylinder, actuator, or process connection. The use point device is where the conversion of compressed air energy to useful work takes place. These points are referred to here are the *point of use supply connection* and the *end use pneumatic device*.

## 7.9 Perceived –vs– Actual Required Pressure

The pressure supplied to use points is often perceived to be too low for proper operation of the connected air demand. The compressed air system engineer must test and validate assumptions of required compressed air pressure at the point of use.

It is often assumed that the pressure supplied to the actual pneumatic device i.e. cylinder, tool, actuator, etc. is equal to the pressure at the supply connection to the machine or process. In fact, between the air supply connection point and the end use pneumatic device there are often control valves, tubing, speed adjustment restrictor valves and other components designed by the OEM (original equipment manufacturer) of the machine or process. There can be a large pressure loss through these components. Therefore, the actual end use pressure at the pneumatic device must be measured to validate the application's actual required pressure.

### a. Perceived Pressure Example:

During an air system audit one concern was the unreliable operation of a pneumatic cylinder used to lift the product at a particular point in the process. The production engineer and machine operator stated "If the system pressure gets below about 5.8 bar the 550 line can't operate." To measure performance, a pressure transducer was mounted directly on the pneumatic cylinder lifting the product. Another transducer measured the supply pressure to the machine. The data tracing below shows one lift cycle of the process.

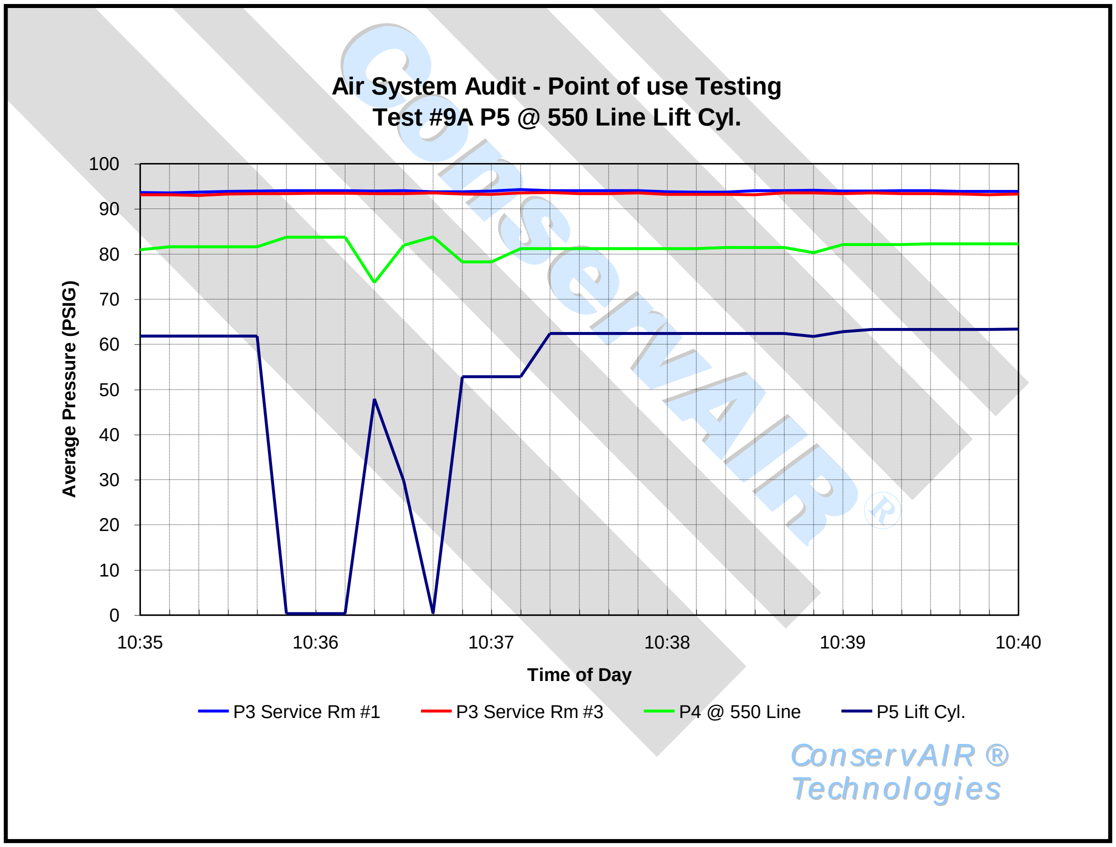



**Figure 7-3: Pressure Profile for Pneumatic Lift Cylinder**

Figure 7-3 shows the compressed air pressure leaving two of the plant's three compressor rooms (Service Rm #1 & #3) is 93 psig. The data labeled P4 @ 550 Line is the machine's supply point connection pressure. The data labeled P5 Lift Cyl. is the use pressure of the pneumatic device, which in this instance is a pneumatic cylinder.

When analyzing the pressure profile, the first observation is that the machine supply connection pressure is 5.5 bar (80 psig) when the system pressure is 6.4 bar (93 psig). A pressure gradient of 0.9 bar (13 psig) from the compressor station to the machine connection is unacceptable. The reported minimum acceptable system target pressure of 5.9 bar (85 psig) at the compressor station will result in the machine's supply connection being less than 4.9 bar (72 psig).

To evaluate the dynamics of the lift cylinder operation, observe that each minor division of time on the chart's X-axis is 10 seconds and the point of use pressure (P5) is taken and the pneumatic cylinder's connection port which is pressurized during the lift stroke. At 10:35 in Figure 7-3, the lift cylinder is raised and in the "home" position. After 40 seconds the cylinder begins lowering to pick up the part, this takes 10 seconds. Then the cylinder remains down for 20 seconds as the fixture rotates in position to engage the part to be lifted. The actual lift of the part occurs during the next 10 seconds followed by 20 seconds as the fixture turns and the part is lowered into the next production step. The part is then released and it takes 10 seconds for the part to release and the cylinder move away. After 30 more seconds, the cylinder is parked in the home position.

The dynamic supply pressure (P4 the green line) droops by another 0.4 to 0.5 bar (6 or 7 psig) when air is actually lifting the cylinder. The data shows actual work pressure on the cylinder of slightly less than 3.4 bar (50 psig) during a lift cycle that was successful. Therefore, the assumption that this application requires 5.9 bar (85 psig) system supply pressure at the compressor station is invalid. Only 3.4 bar (50 psig) dynamic supply pressure is necessary to successfully accomplish the production operation. The problem is not insufficient supply pressure to the air system. The problem is an inappropriate pressure profile and the resulting irrecoverable pressure loss between the compressor station and the point that compressed air energy is converted to work.

### b. Flow Static –vs– Flow Dynamic

When assessing point of use pressure profile performance, the compressed air system engineer must assess both the static and dynamic profiles. In the lift station example in Figure 7-3, the static supply pressure is 0.4 to 0.5 bar (6 or 7 psig) higher than the dynamic pressure as airflow occurs. The pressure variation from the static to dynamic condition must be measured and quantified. The impact of this variation will affect various end use applications in different ways. End use applications can be divided into two general classifications, Flow Static and Flow Dynamic.

**Flow Static** the low or no flow condition that exists when an end use application's work function is being performed. For flow static applications peak airflow does not occur simultaneously with the minimum pressure required.

In **Flow Dynamic** the high flow condition that exists when and end use application's work function is being performed. For flow dynamic applications peak airflow rate and minimum acceptable pressure must occur simultaneously.

The lift cylinder application discussed here is a flow dynamic application. As the compressed air energy is doing work, the pneumatic cylinder must overcome both static and dynamic forces. Static forces include the weight of the fixture and the part which the cylinder is lifting. As the cylinder begins moving it must accelerate the mass of the fixture and part which is a dynamic force. Other dynamic forces include the internal friction force of the cylinder, and the external mechanical forces of the fixture mechanism. Therefore, the most important aspect of the pressure profile data in the lift cylinder application is the dynamic supply pressure available as the air is flowing during the lift operation.

One example of a Flow Static end use is a simple pneumatic clamping cylinder. While the cylinder is advancing or retracting, it must simple move the clamp mechanism. The actual end use work is accomplished after the clamp engages and the cylinder stops moving. This static condition is when the critical highest pressure is necessary to develop the necessary clamping force. In this application the most critical pressure profile performance occurs during the static condition.

The compressed air system engineer must be aware of the potential interaction of both flow static and flow dynamic end use requirements. If a flow dynamic application impacts the connected supply pressure to a machine or process, the dynamic condition will often also affect flow static applications within the machine. Dynamic pressure variations can also occur in localized sectors of the plant air distribution piping, potentially affecting all end use applications in the sector.

When assessing the point of use pressure profile, and validating perceived use point pressure requirements, the compressed air system engineer must study the dynamic performance of the system's pressure profile.

### c. Use Point Piping Pressure Loss

The compressed air system engineer will frequently find point of use piping and connection practice which results in excessive irrecoverable pressure loss. This leads to invalid perceived pressure requirements of the connected air demand. Remedial measures should include modifying point of use piping to support peak airflow rate to the connected air demands while operating with minimal dynamic pressure loss. The design of production machinery and processes may result in large pressure drop between the compressed air supply connection point and the end use pneumatic device. OEM designs should be reviewed and modified as necessary to achieve proper operation without requiring excessively high compressed air pressure at the compressed air supply connection point.

### d. Use Point Target Pressure

The compressed air system engineer must establish the pressure profile's use point target pressure. Avoid establishing an excessively high target pressure for production equipment compressed air supply connection.

## 7.10 Key Learning Points – Use Point Pressure

1. Evaluate use points that require high system pressure.
2. Validate perceived high pressure requirements.
3. Eliminate poor point of use piping causing excessive pressure loss.
4. Check dynamic supply pressure to end use pneumatic devices.
5. Review OEM designs to identify excessive pressure loss within machines.
6. Establish an appropriate target pressure for point of use supply connection.

## 7.11 Distribution System Pressure Profile

Compressed air distribution piping extends throughout the entire manufacturing facility. In some cases a central compressor station may serve multiple buildings at a single plant site. The system pressure profile must account for distribution piping pressure loss (pressure gradient) throughout the entire facility.

### a. Distribution System Performance

When assessing distribution performance, it is necessary to check peak airflow and pressure gradient. The system dynamics driven by high volume intermittent demand events can create transient velocity surges that increase the gradient. In Figure 7-4, the pressure "Orifice Downstream" is the supply pressure to the system. Notice how the distribution pressure in Building 13 tracks with the supply pressure. The figure also shows some widening of gradient from the "Orifice Downstream to Bldg 13" just before 15:30 hrs when airflow peaks at 4,250 scfm (120 $m^3$/min).

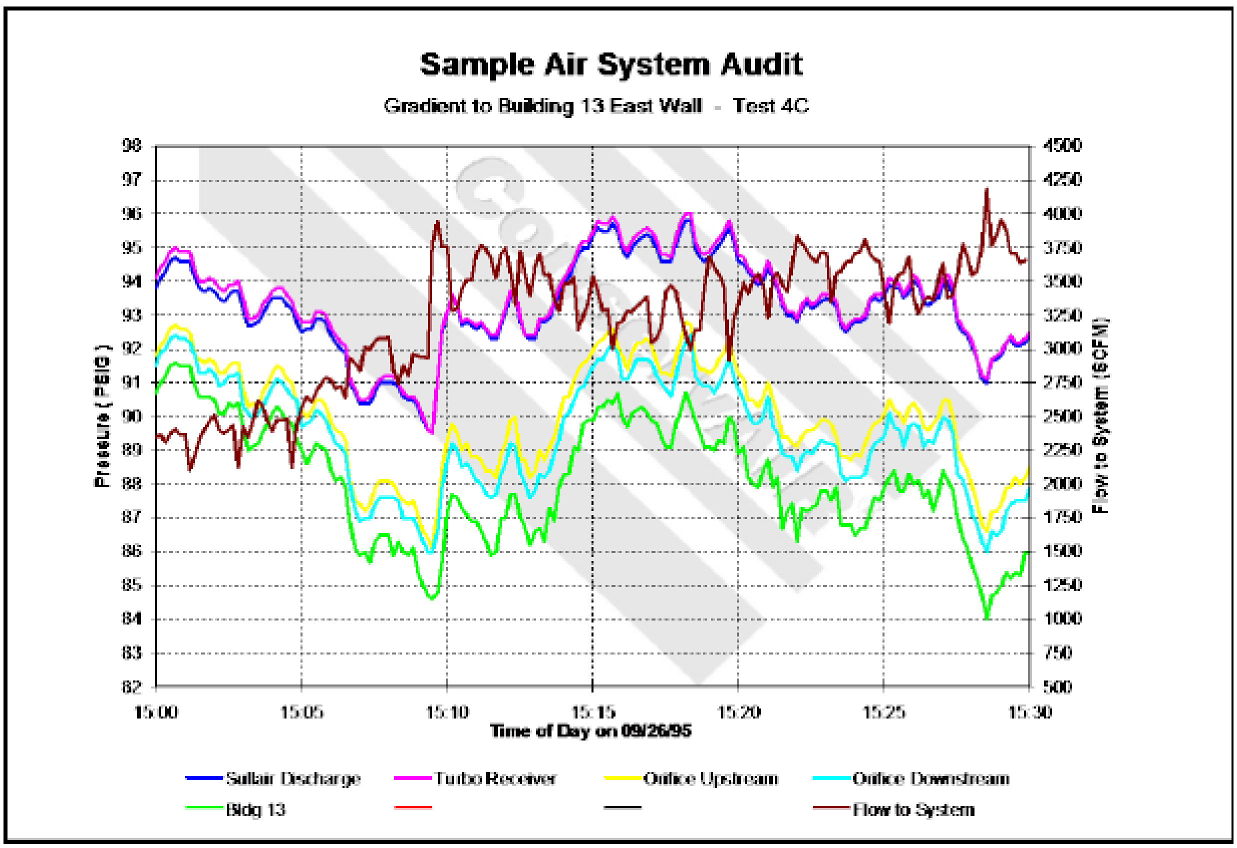



**Figure 7-4: Showing Normal, Acceptable Distribution Pressure Gradient**

The pressure gradient from supply to Building 13 is 1 to 2 psig and through time, the pressure in distribution directly tracks the supply pressure. The pressure profile represented here is normal system performance with an acceptable pressure loss in the distribution system.

A pressure gradient check to another location (Building 1) in the same system is shown in the Figure 7-5. This figure shows extreme gradient excursions affecting distribution pressure at the Bldg. 1 air receiver with relatively small flow peaks of 3,750 scfm (106 $m^3$/min).

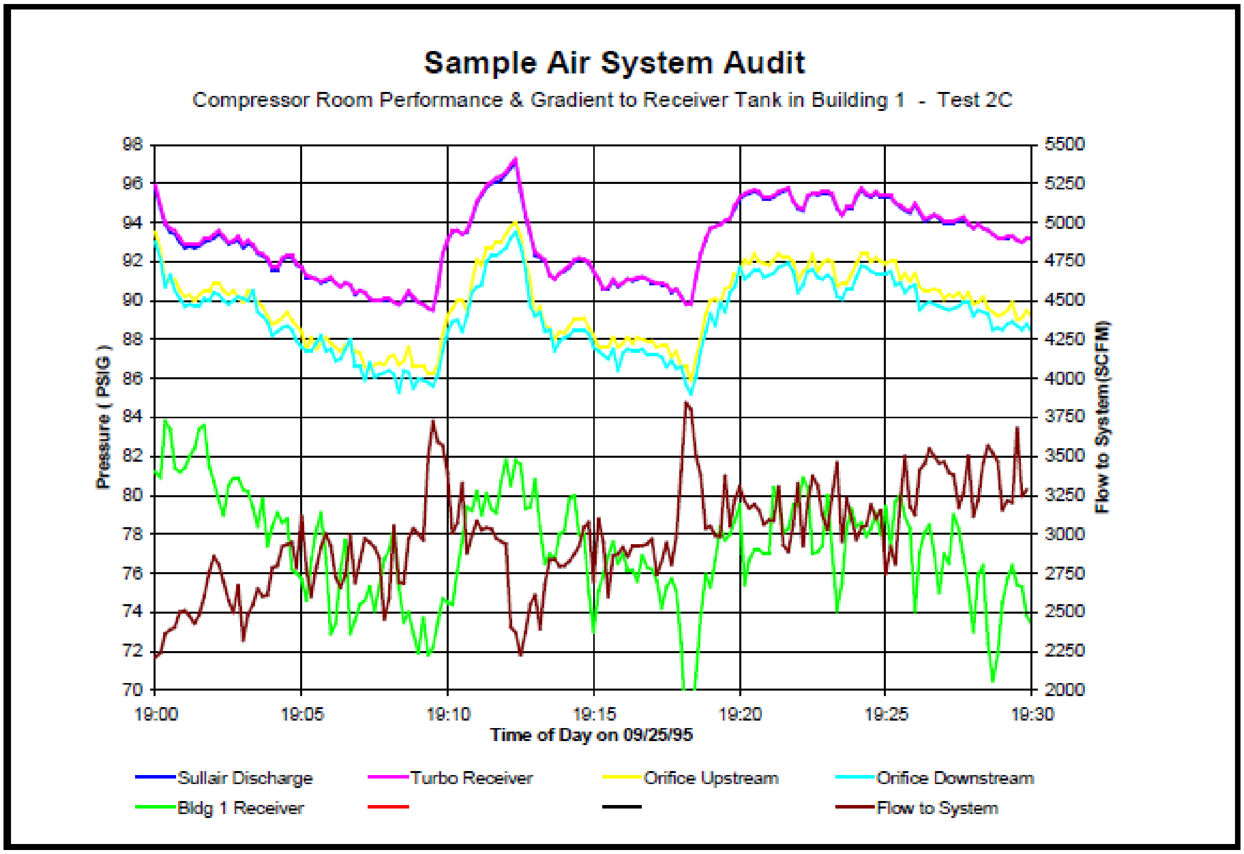



**Figure 7-5: Showing Extreme, Unacceptable Pressure Gradient**

When analyzing the data, first notice that the pressure gradient from supply operates at a sustained differential of about 0.7 bar (10 psig) to Bldg 1. The pressure in Bldg 1 generally tracks with the system supply pressure. However, under conditions of peak airflow the pressure gradient increases to over 1.1 bar (16 psig).

The compressed air system engineer must measure and assess distribution system performance. The system pressure profile must account for the pressure gradient that exists. ***Normal pressure gradient should be limited to 0.15 bar (***2 psig) maximum and should directly track supply pressure changes. When extreme pressure gradient and inconsistent performance exists, the compressed air system engineer must further investigate distribution piping layout. Remedial measures must be implemented to correct distribution pressure gradient before a workable pressure profile can be achieved.

### b. Effect of A Sustained Pressure Gradient

A sustained pressure gradient can act like a large leak in an air system. A leak in an air system essentially connects the air system to atmosphere. You could think of the atmosphere as a very large volume which is at low pressure. Air flows from high pressure to low pressure following the path of least resistance. Therefore, air flows out of the air system to atmosphere.

A pressure gradient inside the volume of the air system causes compressed air to do the same thing. Within an air system compressed air flows from where pressure is high to places where that air pressure is lower. The system in **Error! Reference source not found.** has two volumes at d ifferent pressure which are connected by a closed valve. When the valve is opened what will happen to the pressure in the two volumes? The pressure will equalize with the high pressure getting lower, and the low pressure increasing. How quickly the pressures change depends on how far the valve is opened or how much resistance there is in the piping between the two volumes.

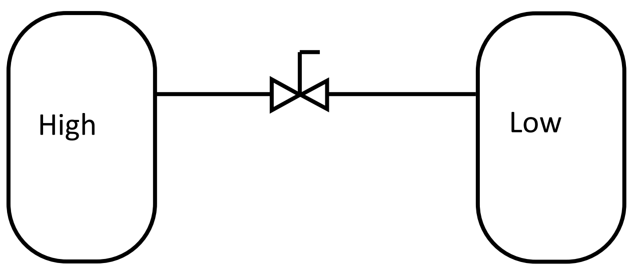



**Figure 7-6: System with Pressure Difference between Two Volumes**

In Figure 7-6, the valve is replaced with a piping restriction and a load / unload air compressor is added to the system. When the compressor is loaded, the piping restriction causes a pressure differential or gradient to exist between the two volumes. When the compressor unloads what will happen to the pressure in the two volumes?

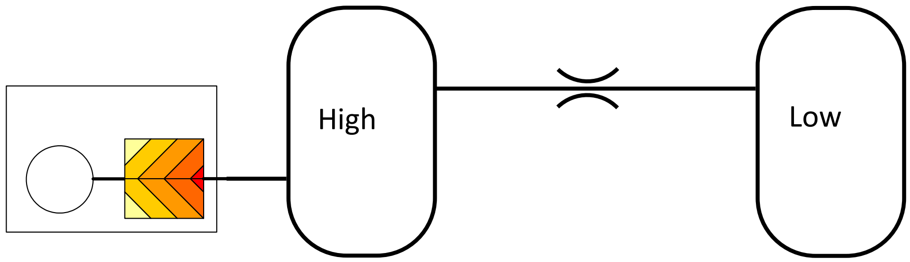



**Figure 7-7: Air System with Sustained Gradient**

The pressure will equalize, the higher pressure will lower and the lower pressure will rise. How will the compressor controls respond to the pressure decrease in the high pressure volume? The compressor will load and the energy from the compressor will re-establish the pressure gradient. When sustained pressure gradient exists in an air system, it can cause one or more compressors to cycle (load / unload) to maintain the pressure gradient. Using compressor power to overcome poor piping is very expensive.

### c. Flow –vs– Delta-P

From fluid mechanics it is known that the pressure drop in a fluid system changes as the square of the flow rate through the system. For example, doubling the flow rate through any portion of the system increases the pressure drop by four times. If piping and components are marginally sized, small airflow increases or decreases can result in a large pressure changes.

Compressed air flows throughout the system by moving from where there is high pressure (point A) to where the pressure (point B) is low. The pressure gradient between any points in the system and the resistance of pipe between the two points determines how fast the air will move from point A to point B.

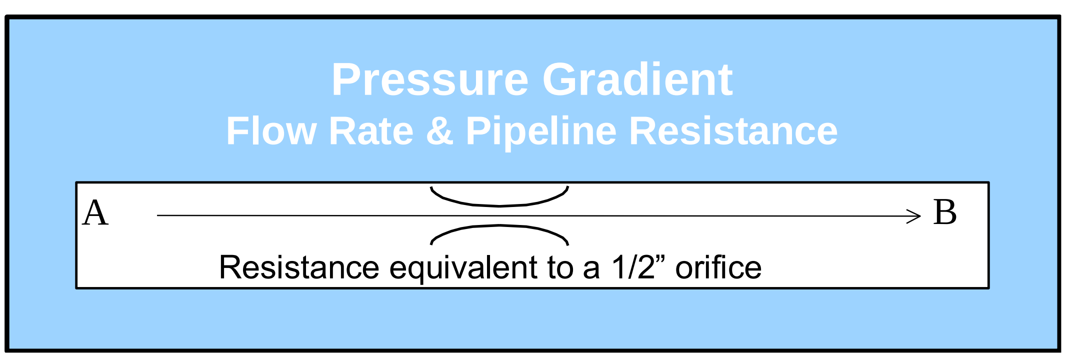



**Figure 7-8: Pressure Gradient in a Pipeline**

Pressure gradient must exist for air to flow through a compressed air system. Pressure gradient in fluid mechanics is expressed as ***dp/dx***- that is, the change in pressure (p) along distance (x).

Compressed air flows from high pressure to low pressure. The fluid velocity and the pipeline resistance determine the pressure gradient. Increasing the pressure gradient, with constant pipeline resistance, creates a corresponding velocity increase. Increase the resistance of the pipe, and the velocity will decrease for the same pressure gradient.

### d. Compressors Create Flow – System Resistance Creates Pressure

Air compressors pump flow, not pressure. When flow meets resistance pressure is created.

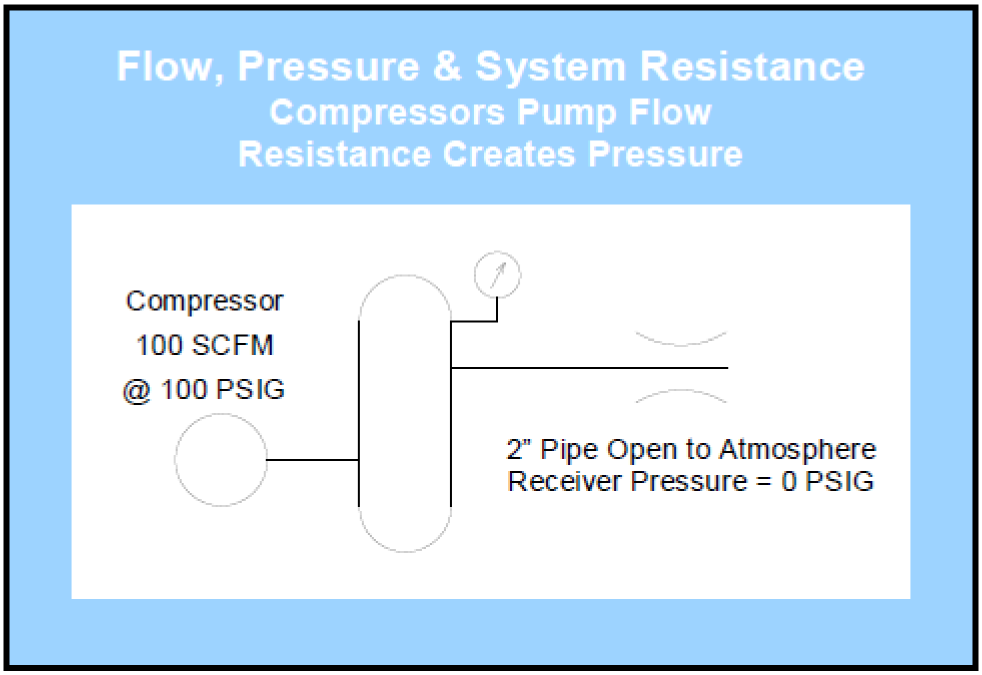



**Figure 7-9: Airflow and System Resistance Determine System Pressure**

The installation above would result in a very low pressure reading on the gauge. A 50 mm (2 inch) pipe has extremely low resistance with only 2.8 $m^3$/min (100 scfm) flowing. The air compressor can be operating and delivering its full rated flow output of 2.8 $m^3$/min (100 scfm) however, the pressure is essentially 0 psig. Since the resistance to flow is very low, no pressure is created.

If a valve is added to the outlet of the pipe in the system above, the system resistance can be changed by adjusting the valve. Closing the valve increases the resistance to flow and opening the valve will decrease the resistance. As the valve is throttled closed what would happen to the pressure reading on the gauge? Closing the valve would cause the pressure to increase. If the valve were closed enough to have resistance equivalent to a 1.3 cm (½ inch) orifice, the pressure would be just under 2 bar (30 psig). Closing the valve further to create a resistance equal to a 0.8

cm (5/16 inch) orifice would result in 6.7 bar (100 psig). Note an orifice coefficient of 0.61 is assumed.

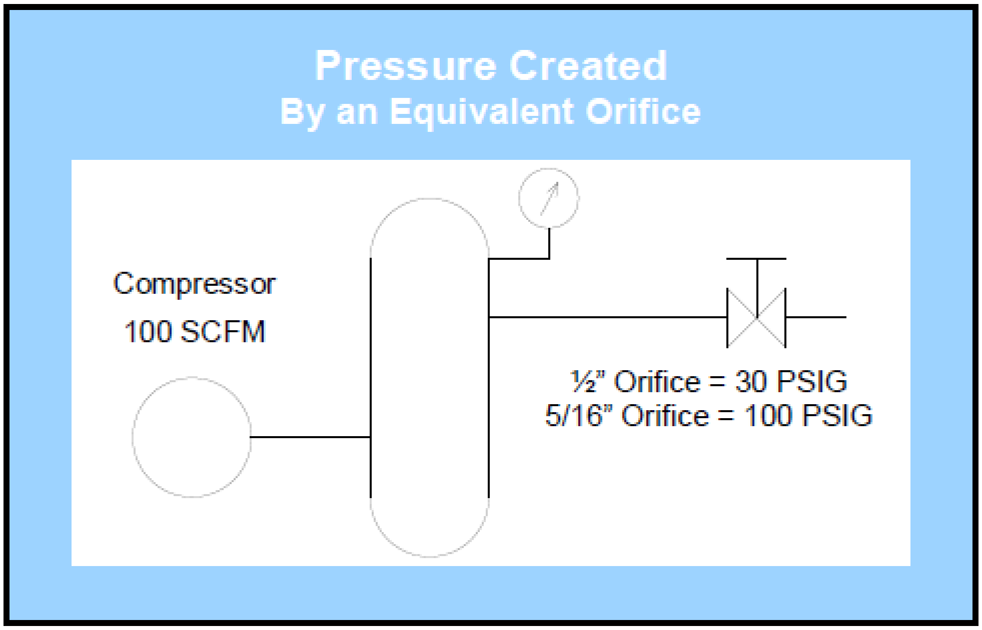



**Figure 4 -** Error! No text of specified style in document.**.1 Resistance to Flow Creates Pressure**

This example uses the equivalent orifice method of evaluating system performance to provide an approximation of system performance. The formula is valid for critical flow. That is to say, the upstream pressure must be greater than two times the downstream pressure. What is demonstrated here is that the pressure in the air receiver depends on the available airflow from the compressor and the amount of resistance to flow that exists in the air system.

## 7.12 Pressure Loss in Fluid Flow

The pressure drop through the system increases as the square of airflow rate (velocity). High volume intermittent demands can create peak airflow rates causing significant pressure excursions.

Figure 7-10 shows the relationship of pipeline velocity (airflow) to pressure drop. It is for a 30 meters (100 ft.) of DN 50 STD (2" schedule 40) Steel Pipe. The characteristic curve represents an exponential increase in pressure drop with increasing velocity. The result is that if the pipeline velocity doubles, the pressure drop increases by 4 times (2 raised to the power of 2 [$2^{(2)}$]). If the velocity triples what happens to the pressure drop?

At 5 m/s (20 ft / sec). pressure drop is 0.03 bar (0.4 psi). If the velocity triples, the pressure drop increases by $3^{(2)}$ or 9 times the original pressure drop 0.27 bar (9 x 0.4 psid = 3.6 psid). If the airflow rate in 90 meters of DN 50 STD pipe reaches 18.5 $m^3$/min, the pressure drop is 0.81 bar. (*If the airflow rate in 300 ft. of 2" pipe reaches 654 scfm (60 fps), the pressure drop is over 10 psig (3 x 3.6 = 10.8 psid)).*

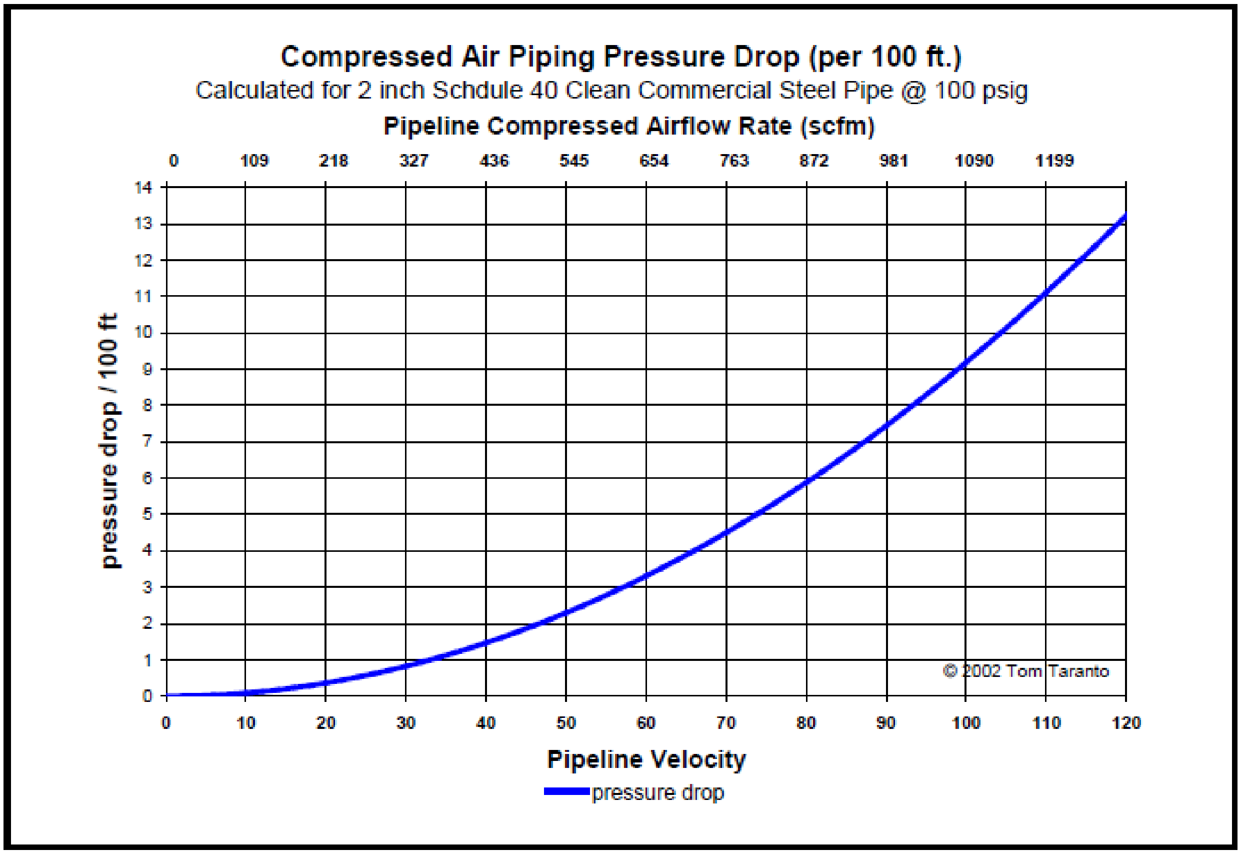



**Figure 7-10: Relationship Between Velocity and Pressure Drop**

For new piping installations, the design pipeline velocity should be 5 to 7.5 m/s (15 to 25 ft./sec). for mainline distribution headers. Branch velocities should not exceed 10 m/s (30 ft./sec). If the system is small in terms of length, higher velocities can be tolerated. The total pressure drop across distribution should not exceed 0.1 to 0.2 bar (1.5 to 2.0 psig) maximum. Keeping the pipeline velocity under 10 m/s. allows the system to support transient peak surge flow without causing a large increase in pressure loss.

## 7.13 Compressed Air Pressure Drop

Pressure drop is a term used to characterize the reduction in air pressure from the compressor discharge to the actual point of use. Pressure drop occurs as the compressed air travels through the treatment and distribution system. A properly designed system should have a pressure loss of much less than 10% of the compressor's discharge pressure, measured from the receiver tank output to the point of use.

Excessive pressure drop will result in poor system performance and excessive energy consumption. Flow restrictions of any type in a system require higher operating pressures than are needed, resulting in higher energy consumption. Minimizing differentials in all parts of the system is an important part of efficient operation. ***Pressure drop upstream of the compressor signal requires higher compression pressures to achieve the control settings on the compressor*.** The most typical problem areas include the aftercooler, lubricant separators, and check valves. This particular pressure rise resulting from resistance to flow can involve increasing the drive energy on the compressor by 6% of the connected power for each 1 bar of differential.

An air compressor capacity control pressure signal normally is located at the discharge of the compressor package. When the signal location is moved downstream of the compressed air dryers and filters, to achieve a common signal for all compressors, some dangers must be recognized and precautionary measures taken. The control range pressure setting must be reduced to allow for actual and potentially increasing pressure drop across the dryers and filters. Provision also must be made to prevent exceeding the maximum allowable discharge pressure and drive motor amps of each compressor in the system.

Pressure drop in the distribution system and in hoses and flexible connections at points of use results in lower operating pressure at the points of use. If the point of use operating pressure has to be increased, try reducing the pressure drops in the system before adding capacity or increasing the system pressure. Increasing the compressor discharge pressure or adding compressor capacity results in significant increases in energy consumption.

Elevating system pressure increases unregulated uses such as leaks, open blowing and production applications without regulators or with wide open regulators. The added demand at elevated pressure is termed "artificial demand", and substantially increases energy consumption. Instead of increasing the compressor discharge pressure or adding additional compressor capacity, alternative solutions should be sought, such as reduced pressure drop, strategic compressed air storage, and demand/intermediate controls. Equipment should be specified and operated at the lowest efficient operating pressure.

### a. What Causes Pressure Drop?

Any type of obstruction, restriction or roughness in the system will cause resistance to air flow and cause pressure drop. In the distribution system, the highest pressure drops usually are found at the points of use, including in undersized or leaking hoses, tubes, disconnects, filters, regulators and lubricators (FRLs). On the supply side of the system, air/lubricant separators, aftercoolers, moisture separators, dryers and filters are the main items causing significant pressure drops.

The maximum pressure drop from the supply side to the points of use will occur when the compressed air flow rate and temperature are highest. System components should be selected based upon these conditions and the manufacturer of each component should be requested to supply pressure drop information under these conditions. When selecting filters, remember that they will get dirty. Dirt loading characteristics are also important selection criteria. Large end-users that purchase substantial quantities of components should work with their suppliers to ensure that products meet the desired specifications for differential pressure and other characteristics.

The distribution piping system often is diagnosed as having a high pressure drop because a point of use pressure regulator cannot sustain the required downstream pressure. If such a regulator is set at 6 bar and the regulator and/or the upstream filter has a pressure drop of 1.3 bar, the system upstream of the filter and regulator would have to maintain at least 7.3 bar. The 1.3 bar pressure drop may be blamed on the system piping rather than on the components at fault. The correct diagnosis requires pressure measurements at different points in the system to identify the component(s) causing the high pressure drop. In this case, the filter/regulator size needs to be increased, not the piping.

### b. Minimizing Pressure Drop

Minimizing pressure drop requires a systems approach in design and maintenance of the system. Air treatment components, such as aftercoolers, moisture separators, dryers, and filters, should be selected with the lowest possible pressure drop at specified maximum operating conditions. When installed, the recommended maintenance procedures should be followed and documented. Additional ways to minimize pressure drop are as follows:

- Properly design the distribution system.
- Operate and maintain air filtering and drying equipment to reduce the effects of moisture, such as pipe corrosion.
- Select aftercoolers, separators, dryers and filters having the least possible pressure drop for the rated conditions.
- Reduce the distance the air travels through the distribution system.
- Specify pressure regulators, lubricators, hoses, and connections having the best performance characteristics at the lowest pressure differential.

### c. Controlling System Pressure

Many plant air compressors operate with a full load discharge pressure of 7 bar and an unload discharge pressure of 7.7 bar higher. Many types of machinery and tools can operate efficiently with an air supply at the point of use of 5.5 bar or lower. If the air compressor discharge pressure can be reduced, significant savings can be achieved. Check with the compressor manufacturer for performance specifications at different discharge pressures.

Demand controls require sufficient pressure drop from the primary air receiver into which the compressor discharges, but the plant header pressure can be controlled to a much narrower pressure range, shielding the compressor from severe load swings. Reducing and controlling the system pressure downstream of the primary receiver can result in a reduction in energy consumption of up to 10% or more, even though the compressors discharge pressure has not been changed.

Reducing system pressure also can have a cascading effect in improving overall system performance, reducing leakage rates, and helping with capacity and other problems. Reduced pressure also reduces stress on components and operating equipment. However, a reduced system operating pressure may require modifications to other components, including pressure regulators, filters, and the size and location of compressed air storage.

Lowering average system pressure requires caution since large changes in demand can cause the pressure at points of use to fall below minimum requirements, which can cause equipment to function improperly. These problems can be avoided with careful matching of system components, controls, and compressed air storage capacity and location.

For applications using significant amounts of compressed air, it is recommended that equipment be specified to operate at lower pressure levels. The added cost of components, such as larger air cylinders, usually will be recouped quickly from resulting energy savings. Production engineers often specify end-use equipment to operate at an average system pressure. This results in higher system operating costs. Firstly, the point of use installation components such as hoses, pressure regulators, and filters will be installed between the system pressure and the end-use equipment pressure.

Secondly, filters will get dirty and leaks will occur. Both result in lower end-use pressure. This should be anticipated in specifying the available end-use pressure.

If an individual application requires a higher pressure, instead of raising the operating pressure of the whole system it may be best to replace or modify this application. It may be possible to have a cylinder bore increased, gear ratios may be changed, mechanical advantage improved, or a larger air motor may be used. The cost of the improvements probably will be insignificant compared with the energy reduction achieved from operating the system at the lower pressure.

It is also important to check if manufacturers are including pressure drops in filters, pressure regulators, and hoses in their pressure requirements for end-use equipment, or if the pressure requirements as stated are for after those components. A typical pressure differential for a filter, pressure regulator, and hose is 0.5bar, but it could be much higher in poorly designed and maintained systems.

When demand pressure has been successfully reduced and controlled, attention then should be turned to the compressor control set points to obtain more efficient operation, and also to possible unloading or shutting off a compressor to further reduce energy consumption.

## 7.14 Key Learning Points – Distribution System Pressure Profile

Distribution pressure gradient requires measurements throughout the system.

1. Check pressure gradient at peak airflow rate.
2. Normally pressure should track supply at < 0.15 bar (< 2 psig) pressure differential.
3. High pressure gradient leads to unstable performance.
4. High pressure gradients in distribution piping must be corrected.
5. Compressors create airflow, system resistance creates pressure.

## 7.15 Key Energy Points – Distribution System Pressure Profile

6. Sustained pressure gradient will drive inefficient compressor load cycles.
7. Pressure drop increases as a function of airflow change squared.
8. Mainline distribution header pipeline design velocity should be less than 10 m/s (30 ft/sec).

## 7.16 Compressor Control Signals

Air compressors are most efficient when operating at full load capacity. However, there are periods of time when the compressed air supply exceeds demand. In that situation, the system’s pressure increases. To rebalance supply and demand, the compressor(s) in the system must reduce the amount of compressed air they are generating. Therefore, compressors are supplied with some type of capacity control allowing, their air delivery to increase or decrease in response to pressure changes in the system.

When the compressed air system’s pressure decreases compressor controls react to increase the amount of air the compressor is delivering.

When the compressed air system's pressure increases compressor controls respond by decreasing the amount of air that the compressor generates.

In the earlier section "3. Supply Side: Compressors and Their Application" several types of capacity control were discussed. One thing all of these controls have in common is that there is a sensor (pressure switch, transducer, etc.) that monitors the air system pressure and provides an input signal to the control system. This pressure measurement is commonly referred to as the "Control Signal Pressure".

Control signal pressure can be measured at different locations in the compressed air system. For standard factory packaged rotary screw air compressors, the Control Signal Pressure is sensed with-in the package, usually near the discharge connection.

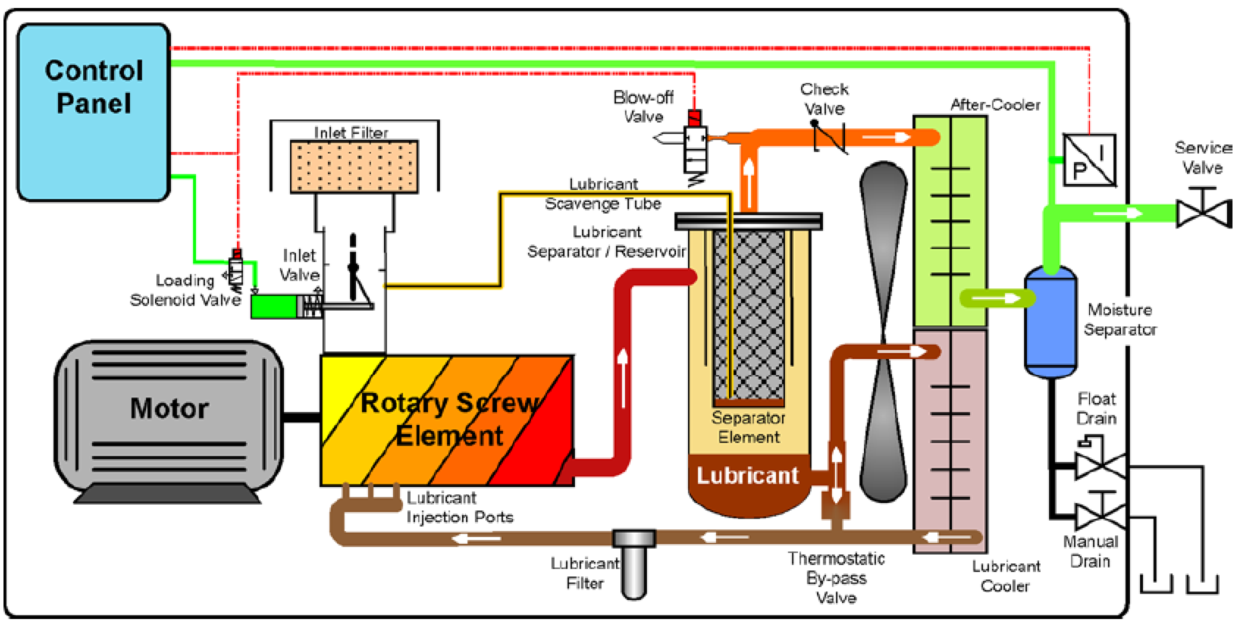



**Figure 7-11: Package rotary screw compressor typical control arrangement**

## a. Control signal shift due to treatment delta-P

When the compressor's control signal is sense inside the compressor package, downstream air treatment equipment (dryers and filters) affects the control pressure signal. Consider the system shown in the block diagram in Figure 7-12 below.

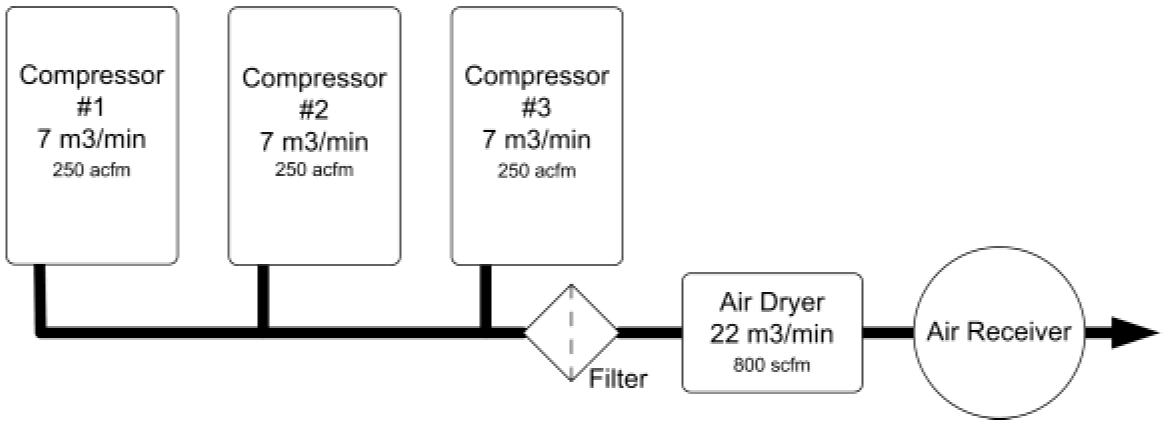



**Figure 7-12: System Block Diagram & Pressure Profile**

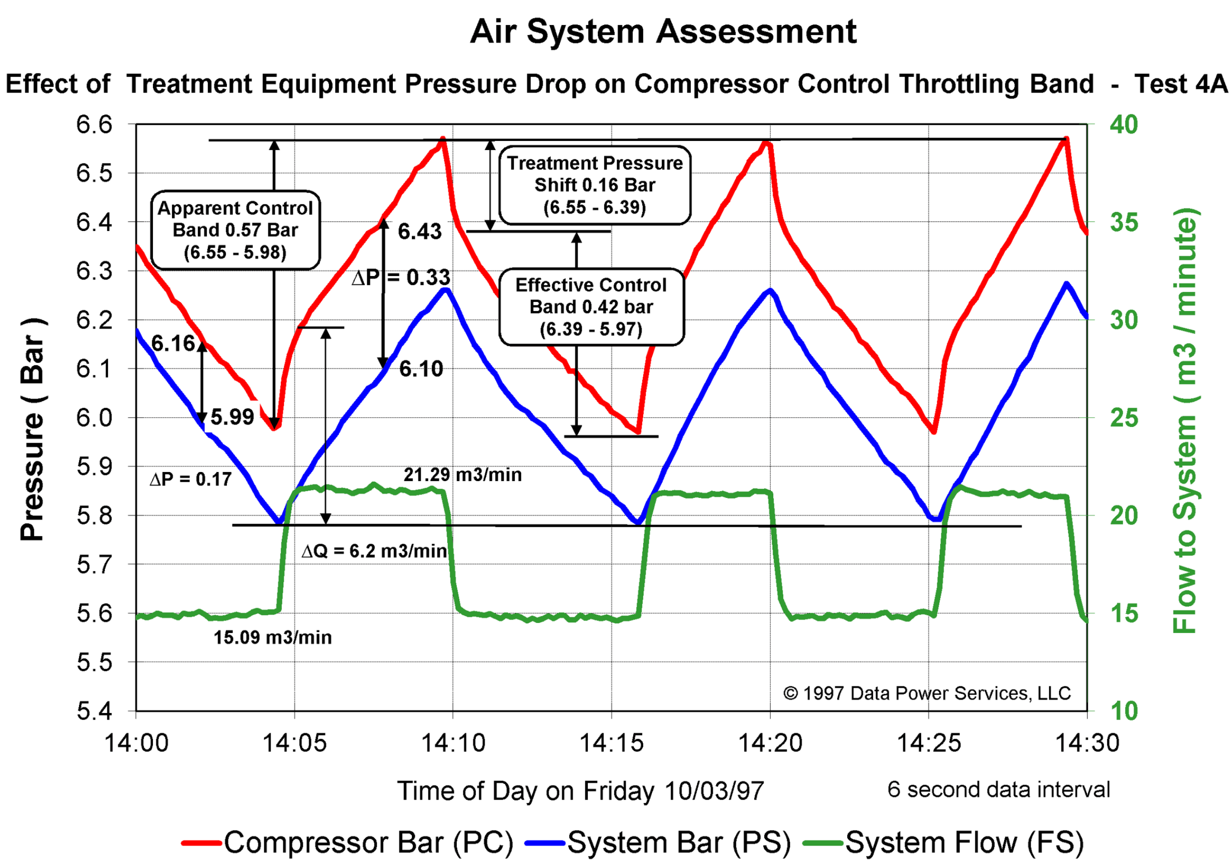



**Figure 7-13: Pressure profile dynamics with compressor load cycles**

In the system shown here, three compressors are served by a single filter and air dryer with a dry air receiver. The air treatment equipment has some resistance to airflow. The resultant pressure drop is dependent on the amount of airflow being delivered from the compressors. With two of the three compressors loaded, the airflow is 15 m3 / min and the pressure drop through the dryer and filter is 0.19 bar.

When viewing the measured pressure profile it appears that the control pressure band of the third compressor is 0.58 bar. However, as the third compressor loads the airflow increases to 21.3 $m^3$ / min. As a result of the increased airflow rate, the pressure drop through the dryer and filter increases from 0.19 bar to 0.35 bar. The 0.16 bar shift in signal pressure is sensed at the compressor discharge. As a result, the effective control pressure band is reduced to 0.42 bar.

Over time the pressure drop through the filter will continue to increase. If the pressure drop with two compressors increases to 0.5 bar the control pressure shift with loading of the third compressor will be even greater. The calculation shown below solves for the control pressure shift that would occur with 0.5 bar initial pressure drop.

$$\Delta P_2 = \left(\frac{Q_2}{Q_1}\right)^2 \times \Delta P_1$$

$$\Delta P_2 = \left(\frac{21.29\ m3/\min}{15.09\ m3/\min}\right)^2 \times 0.5\ bar = 0.995\ bar$$

The resultant control pressure shift would be $\Delta P_2 - \Delta P_1$ , or 0.495 bar

Given that the apparent control band of the 3rd compressor is 0.57 bar, and that the control pressure shift would be 0.5 bar; the remaining effective control band for the third air compressor would be reduced to 0.07 bar. When operating at this extremely low effective control pressure band, the 3rd compressor would operate with very short rapid load / unload cycles an operating condition known as short cycling. In addition to the mechanical wear-and-tear when lubricant injected rotary screw compressors are “short cycling”, the part load energy efficiency is extremely poor.

## b. Other Components Can Affect Compressor Control Pressure Signals

The system should be reviewed to ensure that there are no other components in the pipeline that could negatively affect the compressor's control signal. In addition to components such as air dryers and filters any valve or component between the control sensing location and the system can affect performance. For example, line mounted check valves can prevent the compressor controls from properly sensing system pressure.

Some compressed air dryers or other components may have built-in check valves that are not immediately obvious.

### c. Remote control signal pressure sensing

One common method to eliminate the impact of control signal pressure variation is to remotely sense the system pressure at a point downstream of the treatment equipment, such as a dry air receiver tank. It is important to remember that remote sensing does not change or eliminate the pressure drop that will exist in the system. Therefore, the compressor must be capable of operating at the actual discharge pressure.

When installing compressor control sequencers, or other multiple compressor automation systems, it is common for these controls to use a remote pressure sensing location.

Any time that remote pressure sensing is used to operate an air compressor's capacity control system it is essential that some local over pressure protection be maintained on the compressor package. If for example someone closes a service valve in the compressed air line that is located between the compressor discharge and the control pressure sensing location the compressor will continue to deliver air until the over pressure safety relief valve opens, or motor overload protection actuates to shutdown the compressor.

## 7.17 Key Learning Points – Compressor Control Signals

1. Air compressor capacity controls react to pressure sensed by its control system.

2. As pressure decreases compressor air delivery will increase until its maximum output is being produced.

3. As pressure increases compressor air delivery is reduced.

## 7.18 Key Energy Points – Compressor Control Signals

4. Restrictions in the system such as air dryers and filters can impact compressor control.

5. Remote sensing or external sequencing of compressor controls can improve control response. If remote sensing is used, over-pressure protection should sense pressure within the compressor package.

# 8. Air Storage and System Energy Balance


## 8.1 Balancing Supply and Demand

Compressed air system performance is constantly changing. Compressed air uses typically start and stop, operate intermittently, or are sometimes cyclic in nature. Some compressed air consumption is relatively constant, such as leakage and open blowing (due to artificial demand unregulated leaks and blowing use will increase and decrease as pressure changes). As a result, a compressed air system's demand side airflow rate is constantly changing.

System controls must adjust the airflow delivered from the supply side of the system in an attempt to match the compressed air demand. Generally, supply side controls monitor the system's air pressure and respond by increasing airflow when pressure is low and decreasing it when pressure is high.

The system response is such that when supply exceeds demand system pressure increases, and when supply falls short of demand system pressure decreases. When the airflow rate of supply and demand is balanced (supply equals demand) system pressure is stable, neither increasing nor decreasing.

When considering the supply / demand balance in a compressed air system it is necessary to consider the dynamic performance of the system.

## a. Dynamics

Dynamics is the study of the effect of time variant parameters on system performance.

Compressed airflow rate is normally quantified in $Nm^3$ / minute. For example a pneumatically powered dense phase material transport system is rated by the manufacturer to consume 8.5 $Nm^3$ / min of compressed air. However, further investigation of the transport equipment reveals a 4 minute operating cycle. During the first half of the cycle material is filling the transport pot and no compressed air is consumed. During the second 50% of the cycle material is actually being transported. It is only during the actual transport time that compressed air is being consumed. Considering the dynamics of the transport system, the flow rate during times when compressed is consumed is 17.0 $Nm^3$ / min, twice the rate of flow specified. This example demonstrates the difference between the average and peak compressed airflow rate of a compressed air demand.

## b. Average Air Demand

For an individual compressed air use point, average air demand is the compressed airflow rate (Nm3 / min) consumed by the use as considered during the time duration of one or more full cycles of operation. For example the average demand of the transport system described above is 8.5 Nm3 / min.

For a system, average air demand is the cumulative airflow rate ($Nm^3$ / min) of the system's combined air demands considered during a time period of several minutes to an hour or more.

### c. Peak Air Demand – Demand Event

For an individual compressed air use point peak air demand is the highest compressed airflow rate (Nm3 / min) necessary to support the air use. A Demand Event is described as a peak air demand along with duration of time during which that airflow rate must be sustained. For example the transport system described above has 17.0 Nm3 / min peak air demand and it represents a demand event of 17.0 Nm3 / min for 2 minutes duration.

For a compressed air system, peak air demand is the highest compressed airflow rate ($Nm^3$ / min) of the system's combined air demand which is a detectable airflow rate greater than the continuous steady demand. Peak demand duration may be a few seconds or minutes of time.

### d. Shift in Air Demand – Demand Shift

A shift in air demand or demand shift; is similar to a demand event where-by air demand quickly increases or decreases. However, operation at the new airflow rate (Nm3 / min) lasts for a long period of time. While a demand event may last for a few seconds or minutes; a demand shift will operate at the new airflow rate for several minutes, an hour, or more.

## 8.2 Maintaining an Efficient Supply / Demand Energy Balance

To provide stable consistent performance, compressed air system controls must maintain a real time energy balance between supply and demand. There are four possible sources of compressed air energy in the system.

- Rotating Capacity – Compressed air energy generated by operating air compressors.
- Rotating Reserve Capacity – Potential compressed air energy in operating air compressors which are operating a less than their full load capacity.
- Storage Capacity – Potential compressed air energy stored in an air receiver tank.
- Stand-by Capacity – Potential compressed air energy in air compressors that are shutdown.

Rotating capacity in a system must be equal to or greater than the system's average air demand. Peak air demand is most efficiently supplied from storage capacity.

Rotating reserve capacity reduces system efficiency. Air compressors are most efficient when delivering their full load capacity. Rotating reserve capacity requires one or more compressors to be operating at part load capacity. During part load operation decreased efficiency of the compressor results in an increase in specific power (kW / $Nm^3$). The magnitude of increased specific power depends on the compressor’s capacity control method and the amount of airflow being generated by the compressor.

Stand-by capacity consumes no energy because the compressor is stopped. Note some types of compressors may have oil pumps, heaters or other auxiliary systems that use a minimum amount of energy even when the compressor’s main drive motor is not running. For stand-by capacity to add to the rotating capacity of the system the compressor must start. All compressors have some permissive start-up time which might range from 10 or 15 seconds to a minute or more.

Storage capacity serves several functions in a compressed air system.

- Storage capacity serves to buffer the rate of pressure rise and decay as compressor capacity control interacts with changing air demand.
- When a system is optimized, the rotating generation capacity should be equal to or slightly greater than the average demand. As a result, peak air demand of short time duration frequently exceeds the rotating capacity. Storage capacity adds to rotating capacity to supply peak air demand.
- The unanticipated shutdown of an air compressor results in the loss of rotating capacity that must be immediately replaced to maintain the supply / demand balance. Stand-by capacity is unavailable during the compressor’s permissive start-up time. Storage capacity is necessary to support the system until the stand-by’s rotating capacity is available.
- Demand shifts occur in many compressed air systems. Start-up or shut-down of a particular production line or process may create a demand shift in the system. In many plants production shift changes and the resultant change in production rates may cause compressed air demand shift. Storage capacity supplies needed airflow during permissive start-up time of stand-by capacity as it comes on-line in response to a demand shift.
- Rotating reserve capacity is frequently used when insufficient storage capacity is available to support, demand events, the unanticipated shutdown of rotating capacity, or demand shifts. Operating rotating reserve capacity decreases system efficiency.

## 8.3 System Supply / Demand Control Strategy

Control strategy to optimize system efficiency should accomplish three basic objectives.

- Operate rotating capacity equal to or slightly greater than the system’s average air demand. Shutdown any rotating capacity that is not needed.

- Operate all compressors at full load with only one compressor operating at part load to provide trim capacity
- Serve demand events from storage capacity. Eliminate the use of rotating reserve capacity and prevent stand-by capacity from coming on-line in response to short duration demand events.

## 8.4 Storage Capacity Calculation

Receiver volume alone does not create useable storage capacity. Useable storage capacity is the product of pressure differential and available storage volume. The formula for determining the amount of useable air storage is:

**Equation 8-1: Storage Capacity Calculation**

$$V_a = V_s \times \left( \frac{\left(P_{\max} - P_{\min}\right)}{P_{amb}} \right)$$

*Where:*

$V_a$ = *Volume of Air: Useable compressed air storage capacity*

$V_s$ = *Volume of Storage: Total volume of storage system*

$P_{max}$ = *Pressure Maximum: Storage or receiver pressure (cut-out pressure)*

$P_{min}$ = *Pressure Minimum: Storage or receiver pressure required (cut-in pressure)*

$P_{amb}$ = *Absolute ambient air pressure*

As the formula demonstrates, changes in actual volume of the storage system –OR– changes in pressure differential affect the useable storage capacity. It also demonstrates that if there is no pressure differential, there is no useable storage capacity.

To determine the receiver size required to supply a particular demand event, a slightly different formula can be used.

**Equation 8-2: Storage Volume Calculation**

$$V_s = \frac{T \times C \times P_{amb}}{P_{\max} - P_{\min}}$$

*Where:*

$T$ = *Time duration of the event (minutes)*

$C$ = *Air demand of the event*

$V_s$ = *Total volume of storage system*

$P_{max}$ = *Maximum storage or receiver pressure (cut-out pressure)*

$P_{min}$ = *Minimum storage or receiver pressure required (cut-in pressure)*

$P_{amb}$ = *Absolute ambient air pressure*

*(Note any consistent units of measure; flow rate, volume, & pressure can be used)*

This formula does not consider the compressed air being supplied by compressors during the event. It only considers the air that is in storage when the event occurs. If storage is being supplied with compressed air during the event, the amount of supply can be subtracted from the air demand of the event in the above formula.

## 8.5 Calculate Storage for Unanticipated Shutdown Event

To illustrate how this formula works, assume that a large manufacturing plant has several centrifugal compressors with capacities of 80 $m^3$/min each. How much air receiver storage volume would be required to maintain 6.5 bar if one 80 $m^3$/min compressor failed and a stand-by compressor was required to start?

Given that the stand-by compressor's permissive start-up time is 1.5 minutes. If the normal storage pressure is 8.5 bar and the minimum system supply pressure is 6.5 bar, what is the required receiver storage volume ($m^3$) to provide the necessary storage capacity? (Ambient Pressure = 1 bar)

*Permissive start time is; the time from the when the stand-by compressor receives a signal to start until the time the compressor begins to supply additional compressed air capacity into the system.*

With this information, the formula would be:

$$60\ m^3 = \frac{1.5\,minutes \times 80\,m^3\ /\ minute \times 1.0\,bar}{8.5\,bar - 6.5\,bar}$$

With 60 cubic meters of storage at 8.5 bar, the production area of the plant would not fall below minimum acceptable pressure during this event.

## 8.6 Storage

Storage can be controlled or uncontrolled.

Controlled storage uses pressure / flow controls to separate the demand side of the system from the supply side. The object of controlled storage is to maintain a reserve of compressed air that is at a pressure higher than the pressure in the distribution system. In a controlled system, pressure in the distribution system is maintained at a low pressure in order to minimize artificial demand and to provide a stable pressure regardless of air use or compressor control responses.

Uncontrolled storage In an uncontrolled system, the pressure throughout the plant rises and falls over the full control range of the compressors. In many cases, the plant air pressure can fall significantly below the lowest desired pressure because the compressor cannot react to changes in demand as quickly as they occur.

Both controlled and uncontrolled storage can benefit system performance with adequate and useable storage.

### a. Engineer Primary Storage Systems

Primary stored air volume upstream of the pressure / flow control is engineered to provide the necessary useable air volume from storage. Stored energy is an important component of system stability. When a machine or process starts, there is a rapid increase in air demand (energy) required by the system. The storage system provides immediate energy. In the case of large demands that require capacity increase from the compressors, storage provides energy during the time it takes for compressor controls to respond.

Receiver volume alone does not create useable air storage. Every compressed air system has a target supply pressure. System supply falling below the target pressure, compromises reliability. Therefore, storage pressure must be maintained higher than the target pressure. The useable air (energy) in storage depends on the receiver volume and the pressure difference between receiver pressure PS and the system supply target pressure PT.

Air demand in the system drops by 6 % to 12% per bar of pressure reduction (assuming 50% to 100% of air use is unregulated)

However, if supply pressure is above the target, energy waste through artificial demand, is 6 % to 12% per bar of pressure reduction (assuming 50% to 100% of air use is unregulated). The target pressure should be the lowest optimal pressure to properly supply productive air demands.

When evaluating controlled storage, the equipment specification should account for the capacity to meet surge demands and transient events as defined in the Plant Demand Profile. Time based response characteristics of the pressure / flow control depend on the nature and magnitude of airflow change during surge demand. In addition to dynamic response, the pressure / flow control must also maintain a controlled pressure differential between storage receivers and the system supply target pressure. The pressure / flow control must be capable of high airflow rate while maintaining control at low pressure differential. Stable operation is necessary during low air demand, with a high degree of capacity turn down.

## 8.7 Compressed Air System Energy Balance

The energy delivered in a compressed air system is a function of the mass of air delivered. The mass of compressed air depends on pressure and temperature. Increasing pressure increases the density and therefore the mass of air. Increasing air temperature will decrease the air's density, thus decreasing the mass of air. This relationship is stated in Amonton's law the pressure of gas is directly proportional to its absolute temperature.

Compressed air is often measured in terms of its volume, for example cubic meters ($m^3$). The volumetric measure of air is irrelevant with respect to the air mass unless the temperature and pressure of the air volume is also known. Therefore, standards are adopted to express the mass of air under "Standard" or "Normal" Conditions resulting in the definition for a Normal Cubic Meter of air (Nm3). Normal conditions according to DIN 1343 (normal physical state are: Temperature 0°C = 273.15K; Pressure 1.01325 bar; and Relative Humidity = 0%; where Density = 1.294 kg/$m^3$.

Compressed air energy transfer can be expressed as the mass flow rate of air at a given operating pressure in $Nm^3$/minute. This is both a measure of volumetric flow rate and the mass or weight flow rate of compressed air. Higher flow rate $Nm^3$/minute delivers higher power rate and the time duration of flow determines the energy transferred.

Compressed air power enters the system from the compressors and exits the system through air demands, including productive demand, leaks, and all points where compressed air leaves the system back to atmosphere. Energy from the compressors is measured as mass flow rate $Q$ ($Nm^3$/minute) from generation or $Q_{gen}$, and energy leaving the system also measured as mass flow rate of air demand $Q_{dmnd}$.

Definition: $Q_{gen}$ – is the compressed air mass flow rate (Nm3/minute) produced by the rotating on-line compressor capacity at any moment in time.

Definition: $Q_{dmnd}$ – is the compressed air mass flow rate (Nm3/minute) escaping from the compressed air system to atmosphere at any moment in time.

The ideal balance between generation and demand is achieved when $Q_{gen} = Q_{dmnd}$.

**Equation 8-3: Energy Balance of Compressed Air Systems**

Ideal Air System Energy Balance

$Q_{gen} = Q_{dmnd}$

Only when system pressure = constant

The first law of thermodynamics states that energy is not created or destroyed. This implies that the airflow rate of generation must equal the airflow rate of demand. From practical experience it is observed that generation and demand are not always equal, resulting in changing system pressure. When *Qgen* is greater than *Qdmnd* system, pressure increases, and when *Qgen* is less than *Qdmnd* system pressure decreases. The energy imbalance between generation and demand is either absorbed into, or released from storage.

**Equation 8-4: Energy Imbalance Between Generation and Demand**

$$\boxed{Q_{gen} \pm Q_{sto} = Q_{dmnd}}$$

Definition: $Q_{sys}$ – is the compressed air mass flow rate (Nm$^3$/minute) produced by the rotating on-line compressor capacity ($Q_{gen}$) at any moment in time minus the air flow absorbed into storage (– $Q_{sto}$ for increasing pressure) or plus airflow released from storage (+ $Q_{sto}$ decreasing pressure).

**Equation 8-5: Practical Air System Energy Balance**

Practical Air System Energy Balance

$$Q_{sys} = Q_{dmnd}$$

$$Q_{sys} = Q_{gen} \pm Q_{sto} = Q_{dmnd}$$

accounts for changing system pressure

The compressed air system engineer must analyze the dynamics of compressed air energy storage. This section develops the mathematical relationships necessary to study the relationship between air system generation, storage, and demand. To optimize the system operation, system energy must remain balanced with air demand. Furthermore, system energy must be optimized between generation (rotating on-line energy) and storage.

### a. Application of the Combined Gas Law

The combined gas law states that the pressure of an ideal gas multiplied by volume divided by temperature is a constant.

**Equation 8-6: Combined Gas Law**

$$P_1 \times \left(\frac{V_1}{T_1}\right) = P_2 \times \left(\frac{V_2}{T_2}\right)$$

The compression process usually begins with air at ambient conditions of pressure and temperature (assuming no moisture content). A larger volume of air is compressed to a reduced volume increasing the pressure of the air. During the compression process, the air's temperature increases. Assuming the air's temperature is ultimately returned to ambient, the resulting end pressure of the compression process is equal to the initial pressure times the ratio of beginning volume to ending volume (compression ratio).

**Equation 8-7: Compression Ratio**

$$P_1 \frac{V_1}{V_2} = P_2$$

$$where\ \frac{V_1}{V_2} = r\ (Compression\ Ratio)$$

$$P_1\, r = P_2$$

If during compression 8 cubic meters of air are reduced to a volume of one cubic meter, the resultant pressure is 8 times the initial pressure or r = 8 times.

Assume the compressor intakes 8 cubic meters of dry atmospheric air at 1 bar. Multiplying by r = 8 results in a pressure of 8 bar absolute or 7 bar gauge.

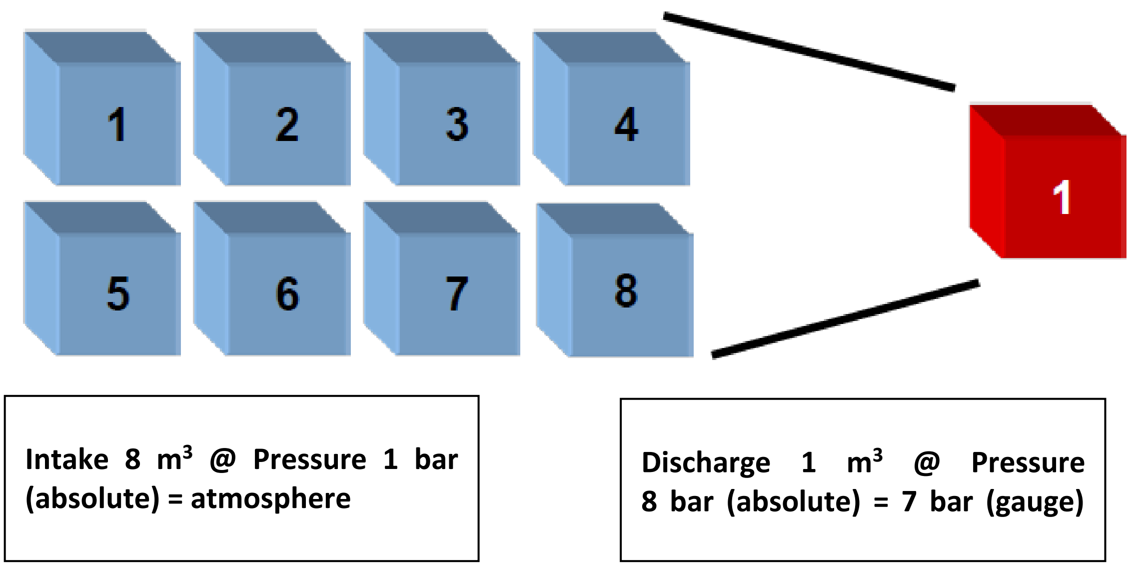



**Figure 8-1: Compression of air from atmospheric pressure to 7 bar gauge (8 bar absolute)**

The application of the combined gas law is the basis of compressed air storage calculation. More advanced forms of the calculation can be used to assess many aspects of compressed air system performance.

If an air receiver of 1 $m^3$ volume is pressurized to 8 bar absolute and its discharge valve is opened to atmosphere, how many cubic meters of air (at atmosphere) will be discharged from the receiver? The form of the ideal gas law use to solve this problem is:

**Equation 8-8: Gas Volume - Receiver Volume Relationship**

$$\boxed{V_{gas} = V_{rec} \times \frac{\Delta P_{rec}}{P_{atm}}}$$

The volume (at atmosphere) of air released from the receiver is equal to the air receiver's volume times the pressure change of the receiver divided by atmospheric pressure. Furthermore the receiver's pressure change can be expressed as the final pressure minus initial pressure ΔP = Pf – Pi. Substituting:

**Equation 8-9: Gas Volume Released from a Receiver**

$$V_{gas} = V_{rec} \times \frac{\left(P_f - P_i\right)}{P_{atm}}$$

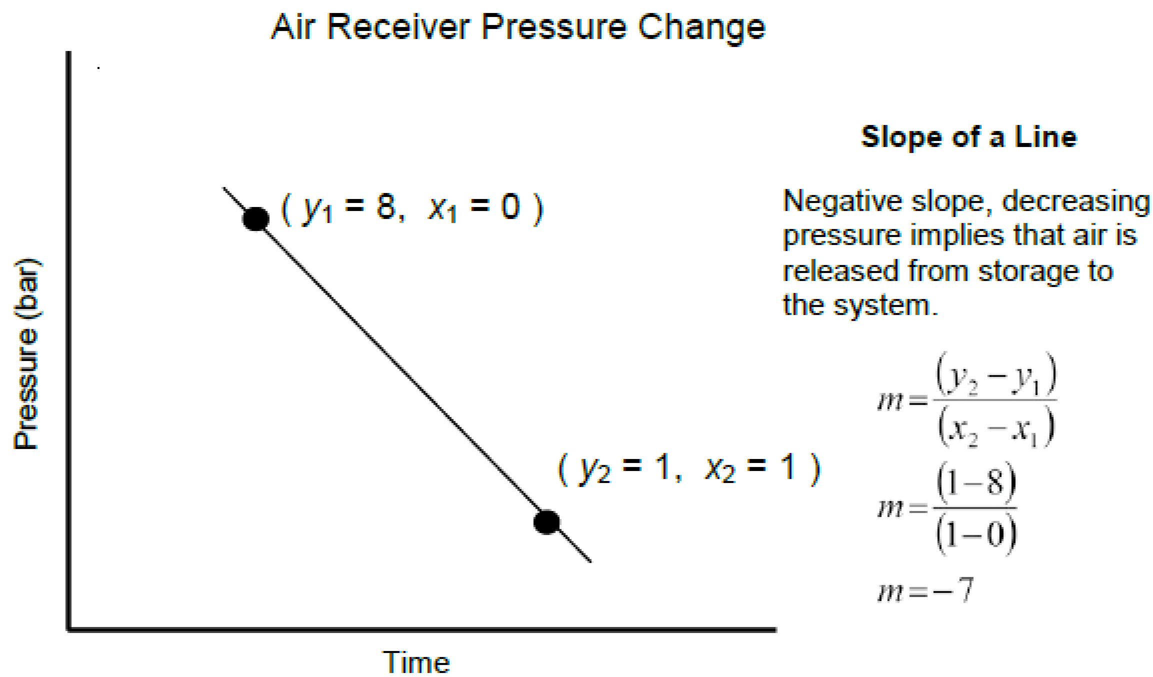
$$V_{gas} = 1\,m^3 \times \frac{\left(1\,bar - 8\,bar\right)}{1\,bar} = V_{gas} = -7\,m^3$$

In the previous discussion of air compression, it was shown that to reach a pressure of 8 bar (abs) requires compressing 8 $m^3$ of air. However, the receiver calculation above shows only –7 $m_3$ of air are released from the 1 $m^3$ receiver above.

What happened to the other cubic meter of air? The eighth cubic meter of air is still inside the air receiver since the receiver's pressure remains at 1 bar (abs).

It is important to note that the receiver's pressure change is the slope of the line calculated from final pressure minus initial pressure. Looking at the receiver pressure throughout time, when pressure is falling (negative slope) air is flowing from the receiver to the system. The xy plot below shows pressure (y) and time (x).




**Negative Slope = Energy is Released from Storage to the System**
**Positive Slope = Energy is Absorbed to Storage from the System**

**Figure 8-2: Air Receiver Pressure Change**

## b. Adding Time to Storage Calculations

Compressors and compressed air system airflow is usually expressed in m$^3$/unit of time. The volume / pressure relationship discussed above does not factor time in the relationship. Dividing each side of Equation 8-9 by time ( T ) and substituting *Q* for *V / T* gives the result below.

**Equation 8-10: Time based Airflow Rate from a Receiver**

$$V_{gas} = V_{rec} \times \frac{\left(P_f - P_i\right)}{P_{atm}}$$

$$\frac{V_{gas}}{T} = \frac{V_{rec} \times \frac{\left(P_f - P_i\right)}{P_{atm}}}{T}$$

$$Q_{gas} = \frac{V_{rec} \times \left(P_f - P_i\right)}{T \times P_{atm}}$$

When time is considered, the gas volume $V_{gas}$ ($m^3$) becomes gas flow rate $V_{gas}$ ($m^3$) / T (minutes) = Qgas ($m^3$/minute). This compressed air storage calculation has many uses for compressed air systems.

### c. Receiver Pump-up Test

Air storage calculations are used in a common method to check the airflow delivery of an air compressor, the "Receiver Pump-up Test". First isolate an air compressor and receiver from the rest of the air system.

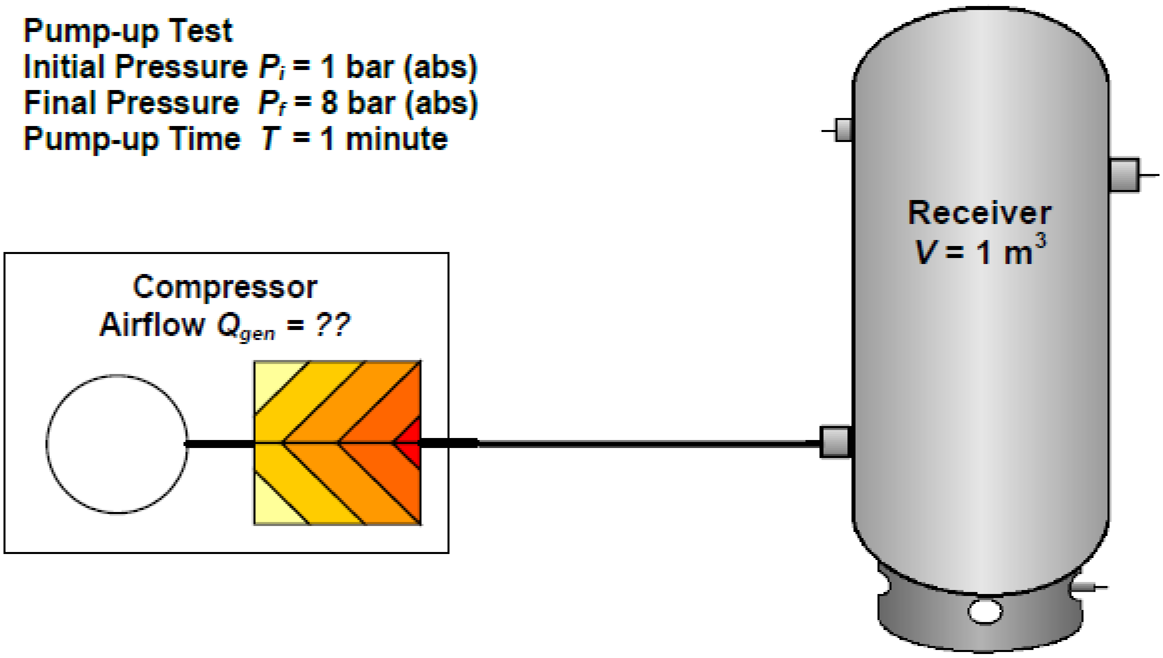



**Figure 8-3: Compressor - Receiver for Receiver Pump-up Test**

Begin operating the compressor at full load capacity and measure the time ( $T$ ) necessary to pump the receiver up from some initial pressure ( $P_i$ ) to a final pressure ( $P_f$ ). If the system volume ( $V_{sys}$ ) being pressurized in known, the air delivery generated by the compressor ( $Q_{gen}$ ) can be calculated as follows. For the example in Equation 8-11 assume the receiver volume $V_{rec}$ = 1 $m^3$; and time $T$ = 1 minute.

**Equation 8-11: Receiver Pump up Test Calculation**

$$Q_{gen} = \frac{V_{rec} \times (P_f - P_i)}{T \times P_{atm}}$$

$$Q_{gen} = \frac{1 m^3 \times (8\, bar - 1 bar)}{1 \text{minute} \times 1 bar}$$

$$Q_{gen} = +7\, m^3 / \text{minute}$$

It is important to note that the airflow for storage is positive which means that compressed air is entering the air receiver. For the calculation shown in Equation 8-9 the air volume $V_{gas}$ calculated is negative indicating air is leaving the receiver. If initial pressure is less than final pressure (as in Equation 8-11) the sign for pressure change ( $\Delta P = P_f - P_i$ ) is positive, and air is flowing from the system into the air receiver.

### d. Total air in receiver

In the previous discussion we have seen the total air in the receiver was 8 $m^3$ while the air released from the receiver was 7 $m^3$. A 1 $m^3$ air receiver when pressurized to 8 bar (abs) contains 8 $Nm^3$ of air expressed at normal ambient conditions. When removing air from the receiver the receiver's pressure remained at 1 bar (abs) therefore only 7 $m^3$ of air is released from the receiver. In most compressed air system applications there is some minimum pressure must be maintained for proper function of the air use points.

### e. Useable air in receiver

Useable compressed air energy in storage is limited by minimum system pressure. The final receiver pressure must be equal to the minimum system pressure plus any pressure drop losses between the air receiver and use point. The useable compressed air energy depends on the receiver volume $V_{rec}$ and the available storage pressure differential ( $\Delta P = P_f - P_i$ ).

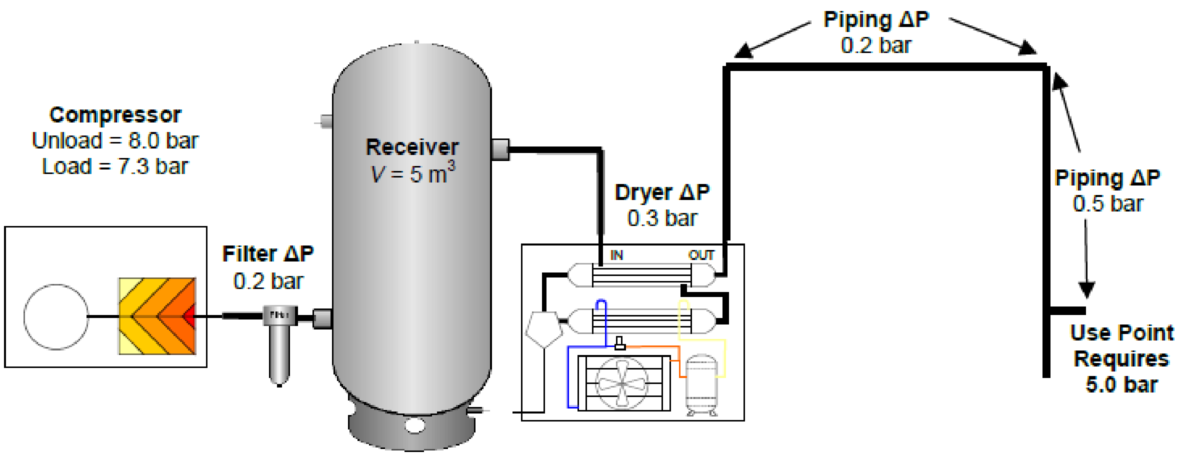



**Figure 8-4: System Diagram - Pressure Profile**

For the air system shown in Figure 8-4, the minimum point of use pressure is 5.0 bar and the compressor control range is; Load at 7.3 bar; and Unload at 8.0 bar. The system pressure profile includes; pressure drop through the filter 0.2 bar and dryer of 0.3 bar, piping loss 0.2 bar, and point of use connection is 0.5 bar loss. Evaluating air storage requires the available storage pressure differential to be calculated ( $\Delta P = P_f - P_i$ ).

What is the minimum air receiver pressure ( $P_f$ ) that can be used for calculating useable air storage? The pressure profile downstream of the receiver determines the minimum pressure. In the system shown use pressure is 5.0 bar, point of use pressure drop is 0.5 bar, distribution piping 0.2 bar, and air dryer is 0.3 bar. Adding these together we find the minimum air receiver pressure is $P_f$ = 6.0 bar ( 5.0 + 0.5 + 0.2 + 0.3 ).

What is the initial pressure that should be used to calculate useable air storage? The compressor control range is 7.3 / 8.0 bar Load / Unload settings. The most conservative approach is to assume that the compressor is at its lowest pressure in the range when calculating the air storage differential. The initial pressure ( $P_i$ ) is 7.3 bar.

Does the filter located upstream of the air receiver affect the initial storage pressure?

No it does not. In this situation with a single load / unload compressor the filter has no effect on initial receiver pressure. The lowest pressure occurs just before the air compressor loads. When the compressor is unloaded, there is no flow through the filter and therefore no pressure drop. Note in multiple compressor systems, the pressure drop through components upstream of the receiver must be subtracted from the minimum pressure of the compressor control band. This assumes the control signal pressure is sensed at the air compressor discharge.

For the system shown in Figure 8-4 what rate of airflow can be supported from storage $Q_{sto}$ for 1 minute of time before useable air storage is depleted? Assume Normal atmospheric pressure $P_{atm}$ = 1.013 bar and ignore temperature and relative humidity.

**Equation 8-12: Airflow storage calculation**

$$Q_{sto} = \frac{V_{rec} \times \left(P_f - P_i\right)}{T \times P_{atm}}$$

The given air receiver volume is 5 m$^3$, as determined above $P_f$ = 6.0 bar, $P_i$ = 7.3 bar and time is given as T = 1 minute. Substituting:

$$Q_{sto} = \frac{5\,m^3 \times \left(6.0\,bar - 7.3\,bar\right)}{1\,\text{minute} \times 1.013\,bar}$$

$$Q_{sto} = -6.42\,Nm^3 / \text{minute}$$

The calculated storage flow rate is $Q_{sto}$ = – 6.42 Nm$^3$ / minute, note the sign is negative indicating that air is flowing from storage to the system.

## f. Pneumatic Capacitance of Compressed Air Systems

Pneumatic Capacitance of a Compressed Air System ($C_{pn}$) represents the compressed air energy absorbed into or released by a compressed air system as its pressure increases or decreases. It is expressed in terms of the "volume of air / unit changes in pressure", for example Standard Cubic Foot / Atmosphere.

**Table 8-1: Definition of Variables and Units of Measure**

| | |
|---|---|
| $C_{pn}$ | = *Pneumatic Capacitance* ($m^3$ / *kPa*) |
| $V_{rec}$ | = *Receiver Volume* ($m^3$) |
| $V_{pipe}$ | = *Piping Volume* ($m^3$) |
| $V_{sys}$ | = *System Volume* ($m^3$) |
| $P_a$ | = *Atmospheric Pressure (kPa)* |
| $P_i$ | = *Initial Receiver Pressure (kPa)* |
| $P_f$ | = *Final Receiver Pressure (kPa)* |
| $\Delta P$ | = *Storage Pressure Delta* ($P_f - P_i$) |
| *rs* | = *Storage Pressure Ratio (Pf – Pi) / Pa* |
| $V_{gas}$ | = *Compressed Air Volume* ($Nm^3$) *Normal cubic meters* |
| $P_{load}$ | = *Compressor Load Pressure (kPa)* |
| $P_{unload}$ | = *Compressor Unload Pressure (kPa)* |
| $Q_{sys}$ | = *Airflow rate for the system (kPa)* |
| $Q_{gen}$ | = *Airflow rate from Generation compressor(s)* ($m^3$/*min*) |
| $Q_{sto}$ | = *Airflow rate of storage* ($m^3$/*min*) |

**Equation 8-13: Pneumatic Capacitance**

$$C_{pn} = \frac{V_{sys}}{P_a} = \frac{V_{rec} + V_{pipe}}{P_a}$$

For every 100 kPa (1 atmosphere) change in system pressure increase or decrease; the system will absorb or release 1 times its volume ($m^3$) of compressed air. Equation 8-13 above gives the Capacitance of the System in terms of cubic meters per atmosphere of pressure. Assume that the atmosphere is at standard conditions of pressure, and temperature, and the piping volume is negligible.

**Equation 8-14: Capacitance $C_{pn}$ = $m^3$ / atm**

$$C_{pn} = \frac{V_{sys}}{P_a} \times \Delta P$$

$$C_{pn} = \frac{3m^3}{1atm} \times 1\ atm$$

$$C_{pn} = 3m^3/atm$$

In Equation 8-14 above, a 3 $m^3$ volume air receiver changing pressure by 1 atmosphere, will displace 3 $m^3$ Normal cubic meters ($Nm^3$) of air into or out of the vessel. The Pneumatic Capacitance of the System is 3 $Nm^3$ / atm. If the $\Delta P = (P_f - P_i)$ is positive; i.e. initial pressure is lower than the final pressure, the air displaced is 3 $Nm^3$, and air is absorbed into the air receiver tank. If the ΔP is negative; i.e. initial pressure is higher than the final pressure the air displaced; – 3 $Nm^3$, is delivered from the air receiver tank.

Because compressed air system pressure is often measured in kPa, it is desirable to express the capacitance of compressed air systems in terms of Normal Cubic Meters per kilopascal ($Nm^3$ / kPa).

For the system above with $V_{sys}$ = 3 $m^3$, the capacitance is 0.03 $m^3$ / kPa. If the System Pressure delta is 100 kPa (1 atm.) then, the stored Gas Volume ( $V_{gas}$ ) is 3 $m^3$ (see Equation 8-15 below). Since atmosphere is the condition defined for Normal Air Conditions $V_{gas}$ = 3 $Nm^3$.

**Equation 8-15: Calculating Gas Volume**

$$\mathrm{C_{pn}} = \frac{V_{sys}}{P_a} = \frac{3\,m^3}{100\,kPa} = 0.03\,m^3/kPa$$
$$\mathrm{V_{gas}} = \mathrm{C_{pn}} \times \Delta P$$
$$For\ \Delta P = 100\,kPa\,(1\,atm.)\,the\,Gas\ Volume\,is:$$
$$\mathrm{V_{gas}} = \mathrm{C_{pn}} \times \Delta P = 0.03\frac{m^3}{kPa} \times 100\,kPa$$
$$\mathrm{V_{gas}} = 3\,m^3$$

## g. Using System Capacitance to Calculate Dynamic Airflow Rate

Capacitance is a useful tool for calculating dynamic performance of compressed air systems. Applying rate of change of system pressure to system capacitance, it is possible to calculate the amount of airflow being released from the system as pressure is decaying, or conversely the amount of airflow being absorbed by the system as air pressure increases.

For example, suppose a compressed air system has a total volume (receivers plus piping) of 4 cubic meters, the compressor's capacity is 4 $m^3$/min and there is an event load which causes a drawdown of system pressure. Data logging of system performance shows that the system pressure falls from ($P_i$) 655 kPa to ($P_f$) 600 kPa during the event which lasts for 1 minute. What is the peak dynamic airflow rate for the system which occurs during the demand event?

The peak dynamic air demand of the system ( $Q_{sys}$ ) is the sum of the base load compressor capacity; $Q_{gen}$ = 10 $m^3$/min, plus the airflow rate being released from storage ( $Q_{sto}$ ) during the drawdown of system pressure.

**Equation 8-16: Calculate the System Capacitance**

$$\mathrm{C_{pn}} = \frac{V_{sys}}{P_a}$$
$$\mathrm{C_{pn}} = \frac{4\,m^3}{100\,kpa} = 0.040\,m^3/kPa$$

**Equation 8-17: Calculate Air Released from Storage**

$$V_{gas} = C_{pn} \times \Delta P = C_{pn} \times (P_f - P_i)$$
$$V_{gas} = 0.04\, m^3/kpa \times (600 - 655) = -2.2\, Nm^3$$

The air delivered from storage (negative slope) during the demand event is 2.2 $Nm^3$. Since the event duration is 1 minute, the rate of airflow from storage is $Nm^3$ / minute. The peak dynamic demand is calculated by adding the base load compressor capacity to the airflow rate from storage.

**Equation 8-18: Calculate Peak Dynamic Demand Airflow Rate**

$$\text{Peak Dynamic Demand}: \quad Q_{sys} = Q_{gen} + Q_{sto}$$
$$Q_{sys} = 4Nm^3/\min + 2.2\, Nm^3/\min$$
$$Q_{sys} = 6.2\, Nm^3/\min$$

## h. Time Based Dynamic Performance Calculations

The dynamic performance of most compressed air systems includes pressure changes in time that occur more rapidly than the one-minute interval in the example above. To assess and calculate the real time performance of compressed air systems it is necessary to take into account the time derivative of pressure *dP/dt*. This analysis allows the compressed air system engineer to account for the volumetric airflow rate *dQ/dt* and the effect of system capacitance interacting with the continual change of system pressure.

Pneumatic Capacitance of an air system interacts with continual pressure changes to effectively absorb or release compressed airflow within the system. The time derivative or rate of change of system pressure *dP/dt* is directly related to the airflow of storage ( *Qsto* ). If *dP/dt* is positive, pressure is increasing- airflow is being absorbed into storage. If on the other hand if *dP/dt* is negative, pressure is decreasing and airflow is being released from storage into the system.

**Equation 8-19: Airflow rate calculated with Pneumatic Capacitance**

$$Q_{sto} = C_{pn} \times \frac{dP}{dt}$$

Suppose in the example above the drawdown event lasted only 25 seconds instead of one minute. The compressed air system has the same total volume (receivers plus piping) of 4 cubic meters, the compressors capacity is 4 $m^3$ / min and there is an event load which causes a drawdown of system pressure. Data logging of system performance shows that the system pressure falls from ( $P_i$ ) 655 kPa to ( $P_f$ ) 600 kPa during the event which lasts for 25 seconds. What is the peak dynamic airflow rate for the system which occurs during the demand event?

**Equation 8-20: Calculate the System Capacitance**

$$C_{pn} = \frac{4\ m^3}{100\ kpa} = 0.040\, m^3 / kpa$$

Using Pneumatic Capacitance of the system and the time derivative or rate of change of system pressure *dP/dt*, calculate the peak dynamic system air demand. Assume that the air compressor is fully loaded throughout the duration of the demand event. The peak airflow rate is airflow rate from the compressor ( $Q_{gen}$ ) plus the airflow rate from storage ( $Q_{sto}$ ). You should note that airflow to the system is the inverse of airflow to storage. If air is leaving storage (negative) it is entering the system as positive airflow. Therefore, in the system equation (Equation 8-21) storage flow ( $Q_{sto}$ ) is taken as the inverse being multiplied by (– 1).

**Equation 8-21: Calculate Peak System Air Demand (Nm³ / minute)**

$$Q_{sys} = Q_{gen} + \left(-1 \times Q_{sto}\right)$$

$$Q_{sto} = C_{pn} \times \frac{dP}{dt}$$

$$substuting:$$

$$Q_{sys} = Q_{gen} + \left(-1 \times C_{pn} \times \frac{dP}{dt}\right)$$

$$Q_{sys} = 4\frac{Nm^3}{\min} + \left[-1 \times \left(0.04\frac{Nm^3}{kpa} \times \frac{(600-655)}{25}\frac{kpa}{\sec} \times 60\frac{\sec}{\min}\right)\right]$$

$$Q_{sys} = 9.28\, Nm^3/\min$$

When the demand event ends, the system pressure is observed to increase from 600 kPa back to 655 kPa in the same 25 seconds of time. Assuming the compressor remains fully loaded, what is the air system demand during this time?

**Equation 8-22: Calculate System Air Demand During Storage Refill**

$$Q_{sys} = Q_{gen} + Q_{sto}$$

$$Q_{sto} = C_{pn} \times \frac{dP}{dt}$$

$$Q_{sys} = 4\,\frac{Nm^3}{\min} + \left[ -1 \times \left( 0.04\,\frac{Nm^3}{kpa} \times \frac{(655-600)}{25}\,\frac{kpa}{\sec} \times 60\,\frac{\sec}{\min} \right) \right]$$

$$Q_{sys} = 4\,\frac{Nm^3}{\min} + \left( -5.28\,\frac{Nm^3}{\min} \right) = -1.28\,Nm^3 \;/\; \min$$

Since the system pressure is decreasing, the pressure change ($P_f - P_i$) is negative. This indicates that the storage volume is releasing compressor capacity and the net system air demand $Q_{sys}$ is more than the compressor output. Again storage flow is the inverse to the system, multiplied by (– 1).

## i. Using Pneumatic Capacitance to Evaluate Compressor Load Cycles

In the preceding event demand analysis, it is assumed that the airflow from generation is constant at the full airflow delivery of the compressor. Using the method in Equation 8-23, the load / unload cycle of a compressor will result in the actual system flow ( $Q_{sys}$ ), while accounting for the airflow which is alternately absorbed into and released from air storage as the compressor loads and unloads.

Given the following information on the system shown in Figure 8-5, use the Pneumatic Capacitance approach to calculate the System Airflow rate ( $Q_{sys}$ ).

*Given:*

- $V_{sys}$ = 6.1 $m^3$
- $C_{pn}$ = 0.061 $m^3$ / kPa
- $Pa$ = 100 kPa
- $P_{load}$ = 600 kPa
- $P_{unload}$ = 655 kPa
- $Q_{sys}$ = ?? $m^3$ / minute
- $Q_{gen}$ = 15 $m^3$ / minute
- $T_L$ = 29 seconds (Compressor Load Time)
- $T_{NL}$ = 25 seconds (Compressor Unload Time)

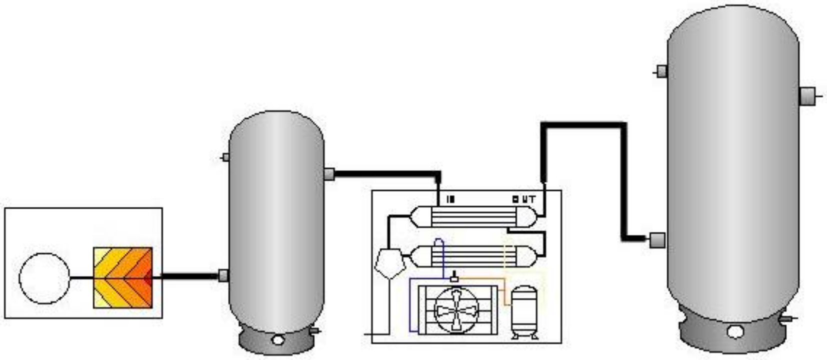



**Figure 8-5: Block Diagram 15 m³ / minute Compressor System**

One method of calculating the system airflow rate for the load / unload cycle operating period is to calculate the percent load for the compressor, and multiply it by the compressor capacity. The percent load of the compressor is the loaded time divided by the total time of the load cycle (loaded time plus unloaded time).

**Equation 8-23: Part Load Capacity of a Load / Unload Style Compressor**

$$Q_{sys} = Q_{gen} \times \%\, load = Q_{gen} \times \left( \frac{T_L}{T_L + T_{NL}} \times 100 \right)$$

$$\%\ load = \left( \frac{T_L}{T_L + T_{NL}} \times 100 \right) = \left( \frac{29}{25 + 29}\, x\, 100 \right) = 53.7\ \%$$

$$Q_{sys} = 15\ Nm^3\ /minute\ \times 53.7\% \ = 8.06\, Nm^3\ /minute$$

Using the system capacitance method provides a means of accounting for system dynamics and the effect of system pressure changes that occur with time.

During the load cycle as system pressure is increasing the pneumatic capacitance of the system absorbs a portion of the compressor capacity. System pressure increases from the compressor load pressure 600 kPa to the unload pressure of 655 kPa.

**Equation 8-24: System Airflow Rate during Compressor Load Cycle**

$$Q_{sys} = Q_{gen} + Q_{sto} = Q_{gen} + \left[ -1 \times \left( C_{pn} \times \frac{dP}{dt} \right) \right]$$

$$Q_{sys} = 15 \frac{Nm^3}{min} + \left[ -1 \times \left( 0.061 \frac{Nm^3}{kPa} \times \frac{(655 - 600)}{29} \frac{kPa}{sec} \times 60 \frac{sec}{min} \right) \right]$$

$$Q_{sys} = 15 \frac{Nm^3}{min} + \left[ -6.94 \frac{Nm^3}{min} \right] = 8.06\, Nm^3 \; / minute$$

The result of Equation 8-24 shows that when compressor is loaded delivering 15 $Nm^3$ of airflow, 6.49 $Nm^3$ is entering storage, and 8.06 $Nm^3$ is supplying air demand.

During the unload cycle system pressure decreases and the pneumatic capacitance of the system delivers airflow to the system. System pressure decreases from the compressor unload pressure of 655 kPa to the load pressure of 600 kPa.

**Equation 8-25: System Airflow Rate During Compressor Unload Cycle**

$$Q_{sys} = Q_{gen} + Q_{sto} = Q_{gen} + \left[ -1 \times \left( C_{pn} \times \frac{dP}{dt} \right) \right]$$

$$Q_{sys} = 0 \frac{Nm^3}{min} + \left[ -1 \times \left( 0.061 \frac{Nm^3}{kpa} \times \frac{(600 - 655)}{25} \frac{kPa}{sec} \times 60 \frac{sec}{min} \right) \right]$$

$$Q_{sys} = 0 \frac{Nm^3}{min} + \left[ 8.05 \frac{Nm^3}{min} \right] = 8.05\, Nm^3 \; / \; minute$$

All of the calculations above result in the System Airflow rate being equal to 8 $Nm^3$ / minute. Data logging compressed air system performance for load / unload controlled systems results in airflow and pressure data which reflects the compressor load cycles. Factoring the air system's pneumatic capacitance with measured performance data using the method described here will give the air system engineer the true dynamic air demand.

## j. Calculating the Pneumatic Capacitance of a Compressed Air System.

To use the dynamic performance analysis described here, it is necessary for the air system engineer to establish the pneumatic capacitance of the system. Pneumatic capacitance can be calculated from the system's volume (cubic meters) divided by atmospheric pressure (100 kPa).

For example, for the system shown in Figure 8-5, suppose the wet receiver is 2 m$^3$, the dry receiver is 3 m$^3$, and there is 60 m of 100 mm dia., along with 265 m of 50 mm dia. of ISO 65 medium piping throughout the system. Assuming the I.D. of 100 mm medium pipe is 105 mm and 50 mm medium pipe is 52.8 mm the system volume can be calculated.

**Equation 8-26: Calculate Pneumatic Capacitance from System Volume**

$$V_{REC} = \left(2\,m^3 + 3\,m^3\right) = 5\,m^3$$

$$V_{50mm\,Pipe} = \frac{\left(\pi D^2\right)}{4} \times 10^{-6} \times Length(m) = \frac{\left(3.14 \times 52.8^{\,2}\right)}{4} \times 10^{-6} \times 265\,m = 0.58\ m^3$$

$$V_{100mm\,Pipe} = \frac{\left(\pi D^2\right)}{4} \times 10^{-6} \times Length(m) = \frac{\left(3.14 \times 105^{\,2}\right)}{4} \times 10^{-6} \times 60\,m = 0.52\ m^3$$

$$V_{sys} = 5.0 + 0.58 + 0.52 = 6.1\,m^3$$

$$C_{pn} = V_{sys} \Big/ P_{atm} = 6.1 \Big/ 100 = 0.061\ \frac{m^3}{kpa}$$

Practically speaking, it can be very time consuming, and rather difficult to accurately calculate the volume of an entire compressed air system. A relatively simple empirical "system capacitance field test" can be performed to allow the compressed air system engineer to establish the pneumatic capacitance of a compressed air system.

While the compressed air system is running, manually unload a test compressor of known generation capacity ( $Q_{test}$ ) for some time period. Measure and record the unload time and rate of change of system pressure. Then at some point reload the known increment of generation capacity and continue to log the load time and dynamic pressure response ( *dP/dt* ) for the system pressure to recover. Assuming that during the test, system air demand ( $Q_{sys}$ ) is constant, and if other compressors are operating, their air delivery is constant during the load / unload cycles of the test compressor, the system's pneumatic capacitance can be calculated.

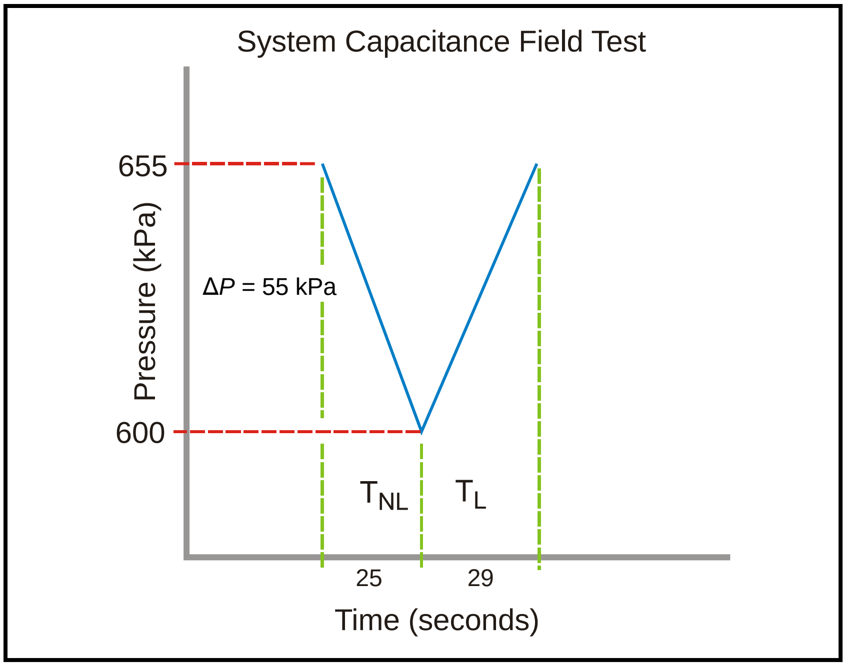



**Figure 8-6: System Capacitance Field Test**

With the test data above the compressed air system capacitance and effective storage volume can be calculated using Equation 8-27.

**Equation 8-27: System Capacitance Calculated from Test Data**

$$C_{pn} = Q_{test(L)} \times \frac{T_L}{T_L + T_{NL}} \times \frac{T_{NL}}{\Delta P}$$

$$C_{pn} = 15\frac{m^3}{min} \times \frac{29\ sec}{(29+25)\ sec} \times \frac{25\,sec}{(600-655)kPa} \times \frac{1\,min}{60\,sec}$$

$$C_{pn} = 0.061\frac{m^3}{kPa}$$

*To solve for the system's volume, multiply by atmospheric pressure:*

$$V_{sys} = C_{pn} \times P_{atm} = 0.061\frac{m^3}{kPa} \times 100\,kPa = 6.1\ m^3$$

## 8.8 Key Learning Points- Air Storage and System Energy Balance

1. System dynamics and the types of compressed air end use applications will determine the nature of the compressed air demand profile.

2. There can be a significant difference between average air demand (what compressors supply) and peak airflow rate driven by real air demand.

3. There are 4 sources of compressed air, rotating on-line capacity, rotating reserve capacity, storage capacity, and stand-by capacity.

4. The amount of useable energy in storage depends on receiver volume and available pressure differential.

## 8.9 Key Energy Points- Air Storage and System Energy Balance

5. The key to consistent, stable, and efficient operation of a compressed air system is maintaining balance between supply and demand.

6. Rotating on-line capacity must be equal to or greater than average air demand.

7. Peak demand is best supplied from storage. However, when air is used from storage there needs to be time and extra compressed air capacity to refill storage before the next event occurs.

8. Compressor controls should shut off compressors that are not needed, operate all compressors at full load, and trim with only 1 compressor operating at part load capacity.

9. Select a trim compressor with efficient part load capacity control.

10. There are many different applications for compressed air storage, engineer storage based on system requirements.

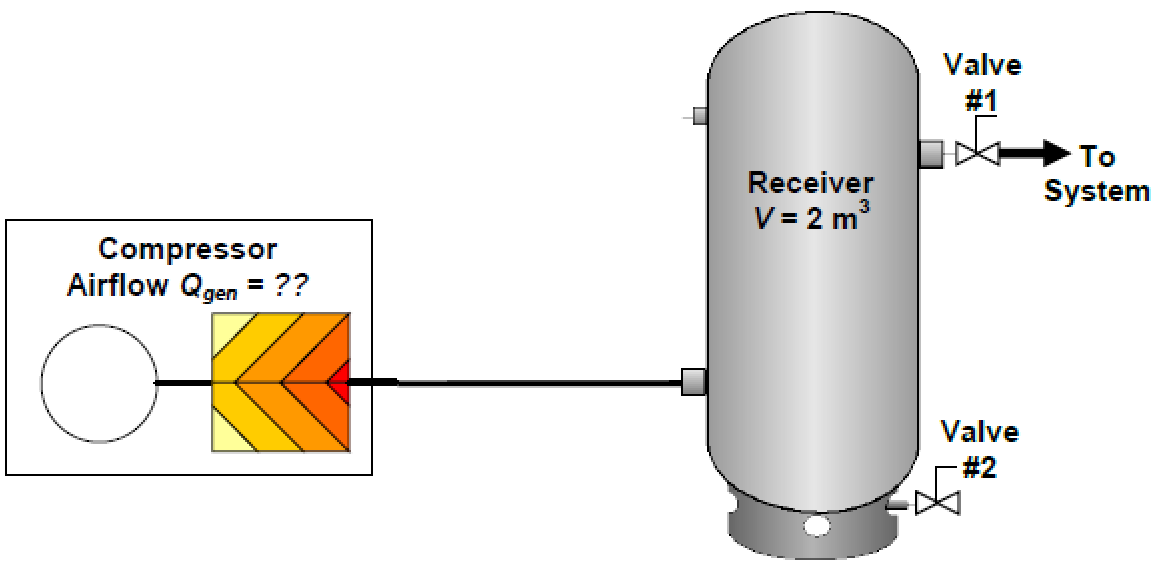
11. In many systems the single largest event requiring storage is the unanticipated shutdown of an operating air compressor.

## 8.10 Exercises

1. An air receiver has 5 $m^3$ volume and its pressure changes from 525 kPa to 650 kPa in 1 minute and 12 seconds. What volume of air ($Nm^3$) is displaced, is the air entering or leaving the receiver, and what is the air flow rate ($Nm^3$ / minute)?

2. During a demand event, the pressure in a 3 $m^3$ air receiver drops from 820 kPa to 600 kPa in 24 seconds. What is the air storage flow rate, is the air flow negative or positive and what is the significance of the air flow's sign?

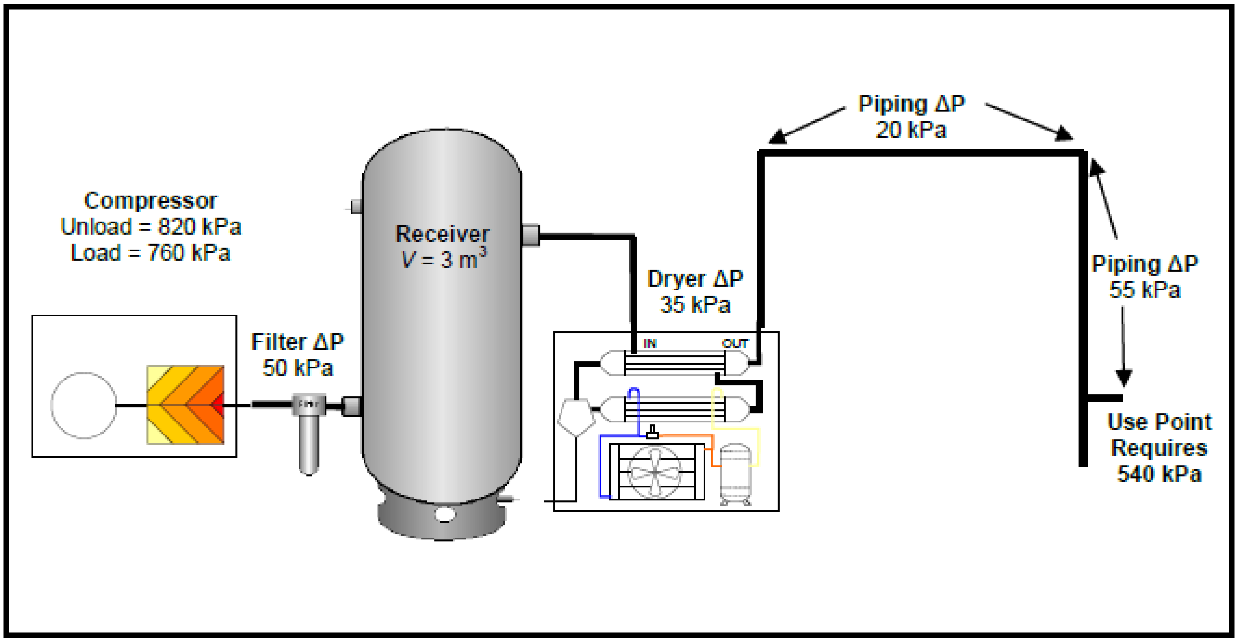
3. An air compressor of unknown capacity shown in the diagram below is field tested to determine its air delivery. Describe the test procedure that you would use. The compressor can be isolated from the system and tested off-line. There is load / unload control with a manual unload / normal switch on the control panel. The compressor manufacturer recommends 550 kPa minimum working pressure, and 750 kPa maximum working pressure. The compressor controls are set to load at 700 kPa and unload at 720 kPa. With these settings the compressor is observed to have a very short load cycle. Why are the load cycles so short?



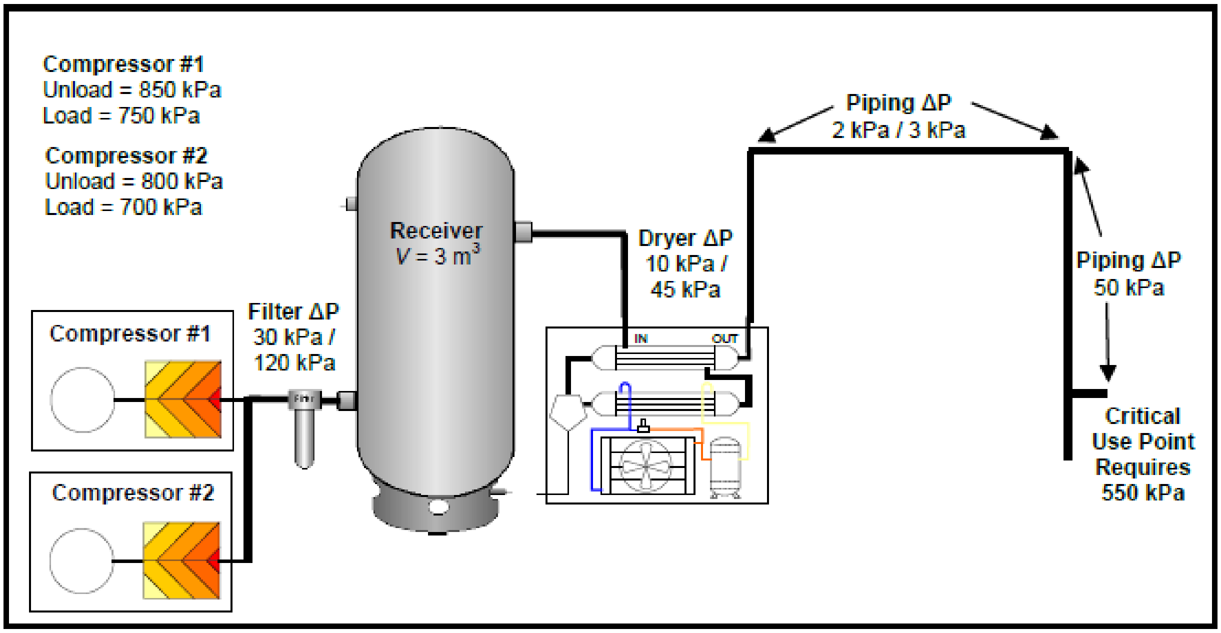

4. The test for the above compressor has shown that the compressor is loaded at 550 kPa and it takes 26 seconds to pump the receiver up to a pressure of 700 kPa. What is the air compressor's delivery ($Nm^3$ / minute)? Using the compressor air delivery ($Nm^3$ / minute) just calculated, how long is the compressor load time using the compressor's load / unload settings of (700 kPa and 720 kPa)?

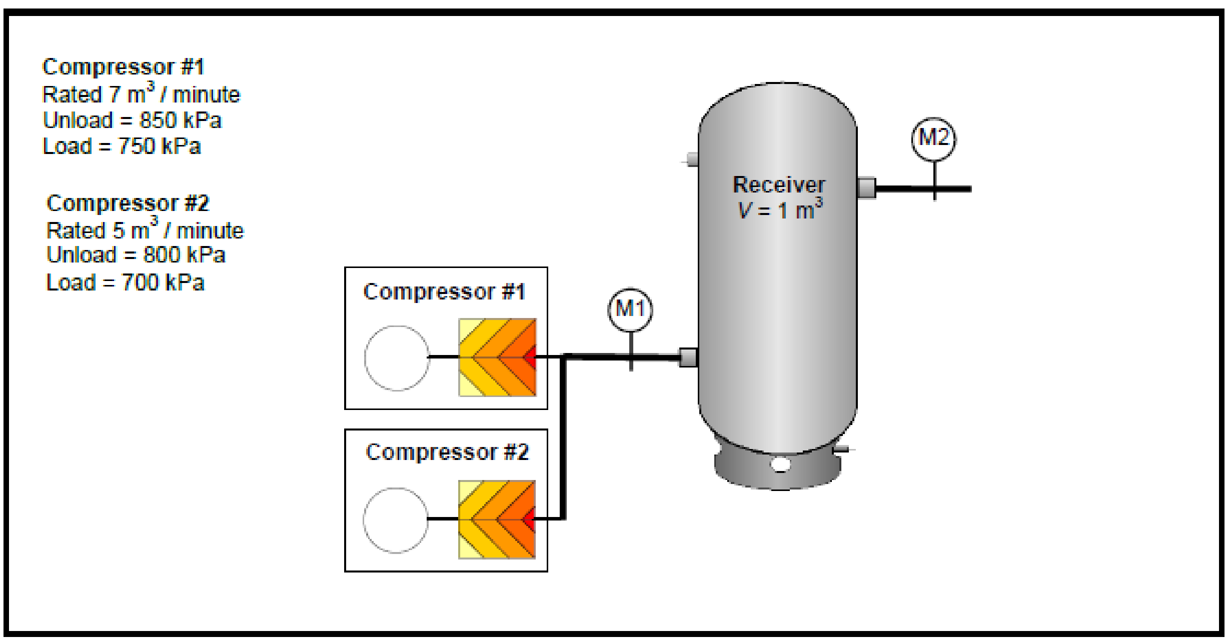
5. A compressed air system operates with the pressure profile shown below. The minimum use point pressure is 540 kPa, what is the useable compressed air ($m^3$) available from storage?




6. A compressed air system operates with the pressure profile shown below. A critical use point demand has a minimum pressure requirement of 550 kPa. An event demand causes system pressure to drop which fully loads both compressors. The pressure profile ΔP's shown are for compressor #1 loaded / both compressors #1 & #2 loaded. For example the filter pressure drop is 30 kPa when only Compressor #1 is loaded and 120 kPa when both Compressors #1 & #2 are loaded. What is the useable compressed air ($m^3$) available from storage to support the demand event?

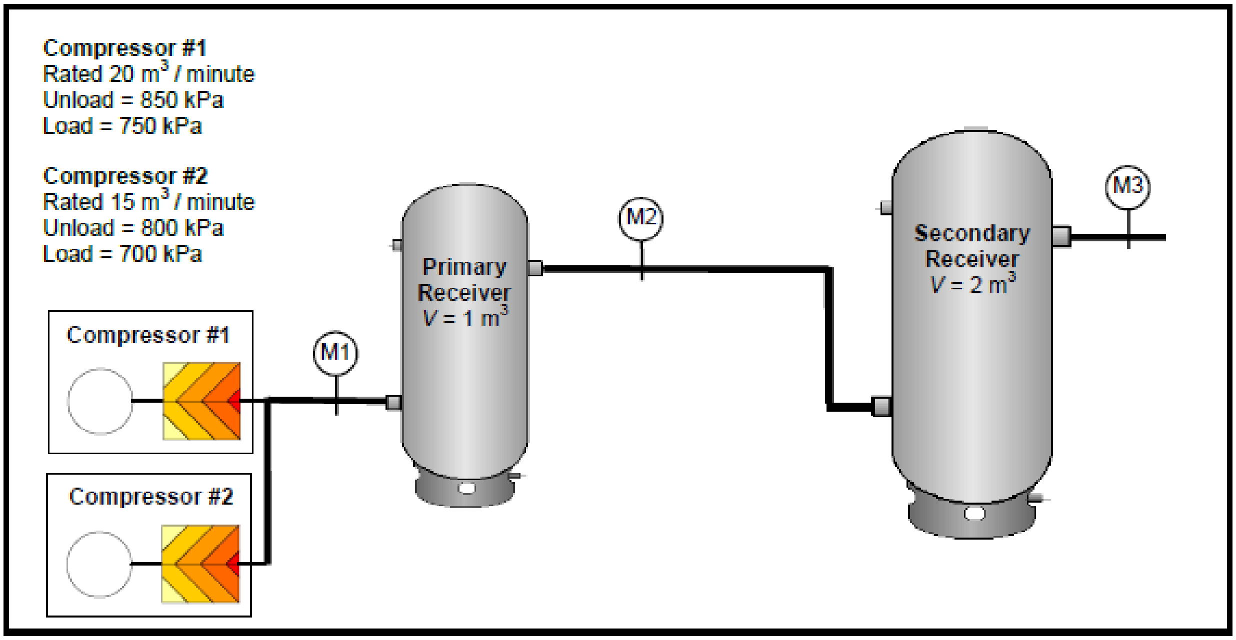



7. A compressed air system includes two compressors and receiver tank as shown below. Compressor #1 is fully loaded and compressor #2 is operating loaded for 15 seconds and unloaded for 60 seconds. What is the airflow rate measured by the two flow meters shown; while compressor #2 is loaded, and while compressor #2 is unloaded?



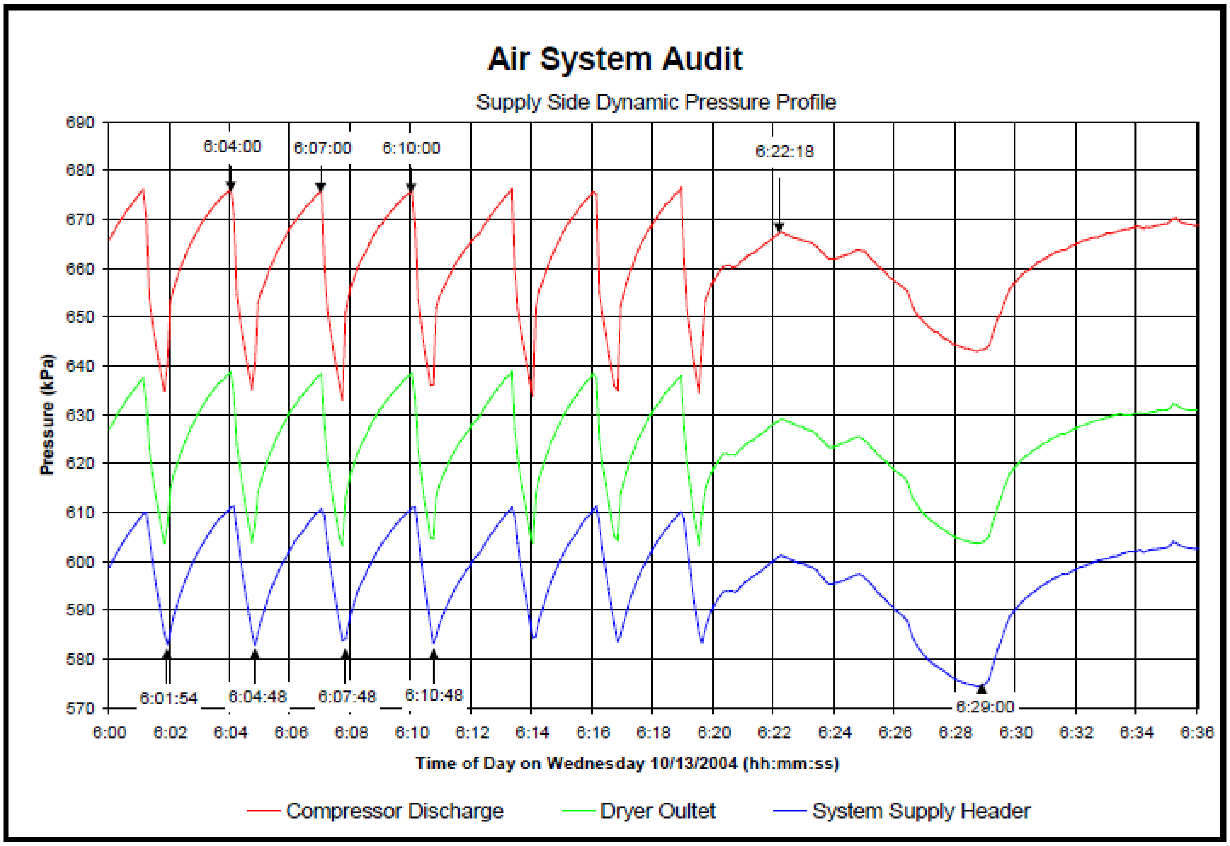
8. A compressed air system includes two compressors and two receiver tanks as shown below. Compressor #1 is fully loaded and compressor #2 is operating loaded for 19 seconds and unloaded for 30 seconds. What is the airflow rate measured by the two flow meters shown; while compressor #2 is loaded, and while compressor #2 is unloaded? What is the air demand at flow meter M3?




9. A compressed air system includes five compressors rated for 23 $Nm^3$ / minute each. All compressors are running with four at full load and one operating as trim capacity with load / unload control. The data tracing below shows a drawdown event. Assuming all compressors are delivering rated capacity and atmospheric pressure is 100 kPa and also given that temperature and relative humidity are at normal conditions answer the following questions.

- What is the total peak air demand rate ($Nm^3$ / minute) during the drawdown event?
- With the existing pressure profile what is the lowest pressure anticipated at the supply header during events of similar characteristic signature?
- If the system supply pressure is to be maintained at or above 570 kPa, how much additional air storage volume is necessary?



Air System Audit
Supply Side Dynamic Pressure Profile
Pressure (kPa)
690
680
670
660
650
640
630
620
610
600
590
580
570
6:04:00
6:07:00
6:10:00
6:22:18
6:01:54
6:04:48
6:07:48
6:10:48
6:29:00
6:00
6:02
6:04
6:06
6:08
6:10
6:12
6:14
6:16
6:18
6:20
6:22
6:24
6:26
6:28
6:30
6:32
6:34
6:36
Time of Day on Wednesday 10/13/2004 (hh:mm:ss)
Compressor Discharge
Dryer Oultet
System Supply Header

# 9. Compressed Air System Controls



Compressed air system controls match the compressed air supply with system demand (although not always in real-time) and are one of the most important determinants of overall system energy efficiency. This section discusses both individual compressor control and overall system control of plants with multiple compressors. Proper control is essential to efficient system operation and high performance. The objective of any control strategy is also to shut off unneeded compressors or delay bringing on additional compressors until needed. All units which are on should be run at full-load, except for one unit for trimming.

Compressor systems are typically comprised of multiple compressors delivering air to a common plant air header. The combined capacity of these machines is sized, at a minimum, to meet the maximum plant air demand. System controls are almost always needed to orchestrate a reduction in the output of the individual compressor(s) during times of lower demand. Compressed air systems are usually designed to operate within a fixed pressure range and to deliver a volume of air which varies with system demand. System pressure is monitored and the control system decreases compressor output when the pressure reaches a predetermined level. Compressor output is then increased again when the pressure drops to a lower predetermined level.

The difference between these two pressure levels is called the control range. Depending on air system demand, the control range can be anywhere from 0.2-1.5 bar. In the past, individual compressor controls and non-supervised multiple machine systems were slow and imprecise. This resulted in wide control ranges and large pressure swings. As a result of these large swings, individual compressor pressure control set points were established to maintain pressures higher than needed. This ensured that swings would not go below the minimum requirements for the system. Today, faster and more accurate microprocessor-based system controls with tighter control ranges allow for a drop in the system pressure set points. This advantage is depicted in the figure below, where the precise control system is able to maintain a much lower average pressure without going below the minimum system requirements. Every 1 bar of pressure difference is equal to about a 6% change in energy consumption. Narrower variations in pressure not only use less energy, but avoid negative effects on production quality control.

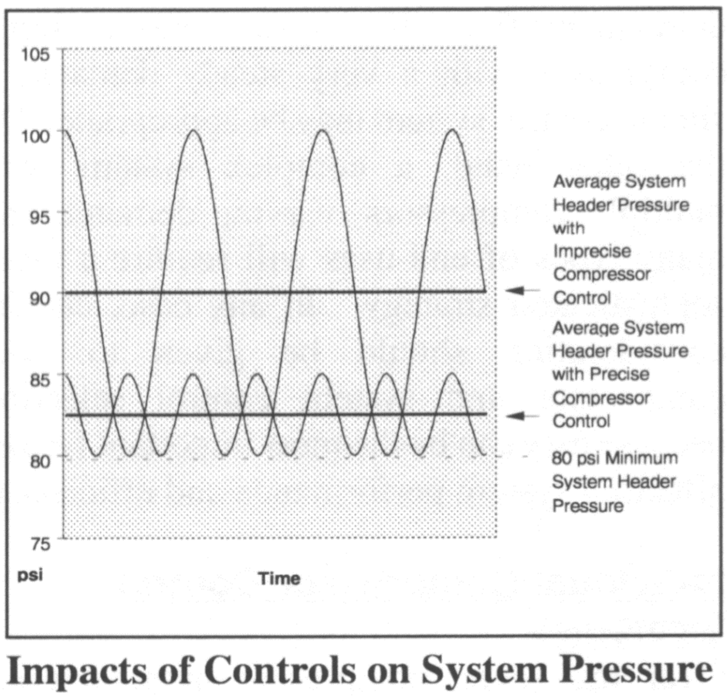



**Figure 9-1: Impacts of Controls on System Pressure**

Caution needs to be taken when lowering average system header pressure because large, sudden changes in demand can cause the pressure to drop below minimum requirements, leading to improper functioning of equipment. With careful matching of system controls and storage capacity, these problems can be avoided.

Few air systems operate at full-load all of the time. Part-load performance is therefore critical, and is primarily influenced by compressor type and control strategy.

## 9.1 Controls and System Performance

The type of control specified for a given system is largely determined by the type of compressor being used and the facility's demand profile. If a system has a single compressor with a very steady demand, a simple control system may be appropriate. On the other hand, a complex system with multiple compressors, varying demand, and many types of end-uses will require a more sophisticated strategy. In any case, careful consideration should be given to both compressor and system control selection because they can be the most important factors affecting system performance and efficiency.

## 9.2 Individual Compressor Control Strategies

Over the years, compressor manufacturers have developed a number of different types of control strategies. Controls such as start/stop and load/unload respond to reductions in air demand, increasing compressor discharge pressure by turning the compressor off or unloading it so that it does not deliver air for periods of time. Modulating inlet and multi-step controls allow the compressor to operate at part-load and deliver a reduced amount of air during periods of reduced demand.

### a. Start/Stop

Start/stop is the simplest control available and can be applied to either reciprocating or rotary screw compressors. The motor driving the compressor is turned on or off in response to the discharge pressure of the machine. Typically, a simple pressure switch provides the motor start/stop signal. This type of control should not be used in an application that has frequent cycling because repeated starts will cause the motor to overheat and other compressor components to require more frequent maintenance. This control scheme is typically only used for applications with very low duty cycles.

### b. Load/Unload

Load/unload control, also known as constant speed control, allows the motor to run continuously, but unloads the compressor when the discharge pressure is adequate. Compressor manufacturers use different strategies for unloading a compressor, but in most cases, an unloaded rotary screw compressor will consume 15-35% of full-load power while delivering no useful work. As a result, some load/unload control schemes can be inefficient.

### c. Modulating Controls

Modulating (throttling) inlet control allows the output of a compressor to be varied to meet flow requirements. Throttling is usually accomplished by closing down the inlet valve, thereby restricting inlet air to the compressor. This control scheme is applied to centrifugal and rotary screw compressors. This control method, when applied to displacement compressors, is an inefficient means of varying compressor output. When used on centrifugal compressors, more efficient results are obtained, particularly with the use of inlet guide vanes which direct the air in the same direction as the impeller inlet. The amount of capacity reduction is limited by the potential for surge and minimum throttling capacity.

### d. Multi-step (Part-load) Controls

Some compressors are designed to operate in two or more partially-loaded conditions. With such a control scheme, output pressure can be closely controlled without requiring the compressor to start/stop or load/unload.

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.

Some rotary screw compressors can vary their compression volumes (ratio) using sliding or turn valves. These are generally applied in conjunction with modulating inlet valves to provide more accurate pressure control with improved part-load efficiency.

### e. Variable Frequency Drives

Historically, the use of Variable Frequency Drives (VFDs) for industrial air compressors has been rare, because the high initial cost of a VFD could not justify the efficiency gain over other control schemes. Cost is no longer a major issue. VFDs may gain acceptance in compressor applications as they become more reliable and efficient at full-load.

## 9.3 System Controls

By definition, system controls orchestrate the actions of the multiple individual compressors that supply air to the system. Prior to the introduction of automatic system controls, compressor systems were set by a method known as cascading set points. Individual compressor operating pressure set points were established to either add or subtract compressor capacity to meet system demand.

The objective of an effective automatic system control strategy is to match system demand with compressors operated at or near their maximum efficiency levels. This can be accomplished in a number of ways, depending on fluctuations in demand, available storage, and the characteristics of the equipment supplying and treating the compressed air.

### a. Single Master (Sequencing) Controls

Sequencers are, as the name implies, devices used to regulate systems by sequencing or staging individual compressor capacity to meet system demand. Sequencers are referred to as single master control units because all compressor operating decisions are made and directed from the master unit. Sequencers control compressor systems by taking individual compressor capacity on- and off-line in response to monitored system pressure (demand). The control system typically offers a higher efficiency because the control range around the system target pressure is tighter. This tighter range allows for a reduction in average system pressure. Again, caution needs to be taken when lowering average system header pressure because large, sudden changes in demand can cause the pressure to drop below minimum requirements, leading to improper functioning of equipment. With careful matching of system controls and storage capacity, these problems can be avoided (see also flow controller).

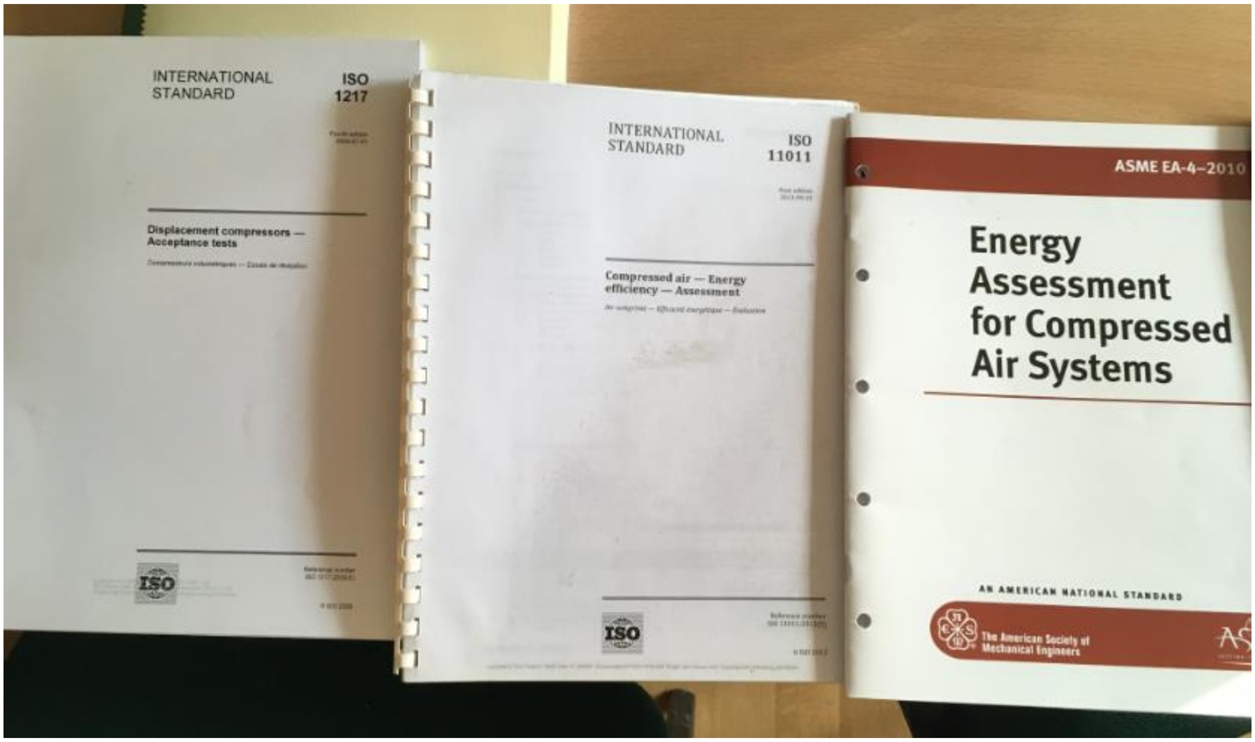
### b. Multi-Master (Network) Controls

Network controls offer the latest in system control. It is important that these controllers be used to shut down any compressors running unnecessarily. They also allow the operating compressors to function in a more efficient mode. Controllers used in networks are combination controllers. They provide individual compressor control as well as system control functions. The term multi-master refers to the system control capability within each individual compressor controller. These individual controllers are linked or networked together, thereby sharing all operating information and status. One of the networked controllers is designated as the leader. Because these controllers share information, compressor operating decisions with respect to changing air demand can be made more quickly and accurately. The effect is a tight pressure control range which allows a further reduction in the air system target pressure. Although initial costs for system controls are often high, these controls are becoming more common because of the resulting reductions in operating costs.

## 9.4 Air Storage and Controls

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

# 10. Compressed Air System Assessment




## 10.1 The Systems Approach

A comprehensive system assessment examines the entire compressed air system; generation, treatment, storage, distribution, use and waste of compressed air. A compressed air system assessment is a detailed engineering study investigating operation of a compressed air system. An assessment begins with a defined list of objectives. The assessment is a discovery process that measures and quantifies system performance in terms of the defined objectives.

The material in this section is summarized from the following documents; U.S. National Standard ASME EA-4 – 2010 Energy Assessment of Compressed Air Systems, ASME EA-4G – 2010 Guidance for Energy Assessment of Compressed Air Systems, International Standards Organization ISO 11011 Compressed Air – Energy Efficiency – Assessment. These documents are highly recommended for individuals involved in Compressed Air System Assessment.

Application of the systems approach to compressed air system assessment evaluates overall system performance, rather than individual component efficiency. The study investigates all sub-systems and components including air compressors, dryers, filters, receiver tanks, piping, controls, pneumatic tools, pneumatically powered machines, and process applications of compressed air. The system boundary includes energy input to the compressed air supply through the production equipment and work performed as a result of the energy input.

The compressed air system assessment must be cost effective. It is unwise to spend a great deal of time and money studying a system that uses a relatively small amount of energy. No two compressed air systems or compressed are system assessments are identical. However, all assessments must evaluate the entire system. It is not necessary to address all parts of the system with equal time and effort. It is necessary is to be sufficiently comprehensive in evaluating all parts of the system to understand it's operation, energy use, performance, and opportunities for improvement.

When the compressed air system assessment is complete the information gathered should allow the assessment team to:

- Understand compressed air point of use applications as they support critical plant production functions,
- correct existing poor performing applications and those that upset system operation,
- eliminate wasteful practices, leaks, artificial demand, and inappropriate use,
- create and maintain an energy balance between supply demand,
- optimize compressed air energy storage and air compressor control.

## 10.2 Plant Background Information

The initial data collection and evaluation process is necessary to define the informational goals and scope of work for the assessment. An estimate of the present energy cost to operate the system is an essential bit information. This will help establish a reasonable budget for the assessment activity. In many cases the present cost of compressed air is unknown. There are two simple calculations that can be used as an estimate.

Nameplate power calculation for annual energy cost:

$$\frac{kW(nameplate) \times load\ \ factor \times hours \times energy\ \mathrm{cost}}{motor\ efficiency} = annual\ energy\ \mathrm{cost}$$

*Where:*

- *kW (nameplate) = full load nameplate kW of the motor*

- *load factor = % driven load operating power in relation to motor nameplate power*
- *hours = annual running hours*
- *energy cost = $ / kWh*
- *motor efficiency = full load motor efficiency*

The calculation above assumes that the air compressor is operated at full load capacity all of the time. Since most air compressors spend some time unloaded or operating a less than full load capacity, this calculation will overestimate the annual energy cost.

Alternatively separate calculations can be made for various part load operating conditions if they are known.

Measured Volts - Amps calculation for annual energy cost:

$$\frac{volts \times amps \times 1.732\ x\ \ pf\ \times hours \times energy\ cost}{1000} = annual\ energy\ cost$$

*Where:*

- *volts = average line to line 3 phase voltage*
- *amps = full load amperage of the motor*
- *1.732 = square root of 3 for phase to neutral voltage from line to line voltage*
- *pf = power factor of the motor (0.80 to 0.85 typical)*
- *hours = annual running hours*
- *energy cost = $ / kWh*

The calculation above assumes that the volts and amps measurement is made with the air compressor operating at full load capacity. Since most air compressors spend some time unloaded or operating a less than full load capacity, this calculation will overestimate the annual energy cost.

Alternatively separate calculations can be made for various part load operating conditions if measurements of volts and amps are made at the different operating condition. NOTE: Power factor also changes with a change in motor amperage.

### a. Example: Determine the Cost of Compressed Air for Your Plan

A typical manufacturing facility has a 150 kW compressor (which requires 170 kW) that operates for 6800 hours annually. It is fully loaded 85% of the time (motor efficiency = 95%) and unloaded the rest of the time (at 25% of full-load power* and motor efficiency = 90%). The aggregate electric rate is $0.15/kWh.

Cost when fully loaded =

$$\frac{170kW \times 6800hours \times \$0.15/kWh \times 0.85 \times 1}{0.95} = \$155{,}147$$

Cost when partially loaded =

$$\frac{170kW \times 6800hours \times \$0.15/kWh \times 0.15 \times 0.25}{0.90} = \$7{,}225$$

Annual energy cost = $155,147 + $7,225 = **$162,372**

** 0.20 to 0.30 is a good estimate for unloaded operation for rotary screw compressors and 0.10 to 0.15 for reciprocating compressors.*

## 10.3 Systems Engineering Process

The systems engineering process is an iterative process of establishing requirements definition >> evaluating the assessment process >> and evaluating outcomes and results.

The chart shown in Figure 10-1 is based on ANSI EIA-632 Processes for Engineering a System and is taken from ASME EA-4 - 2010 Energy Assessment for Compressed Air Systems.

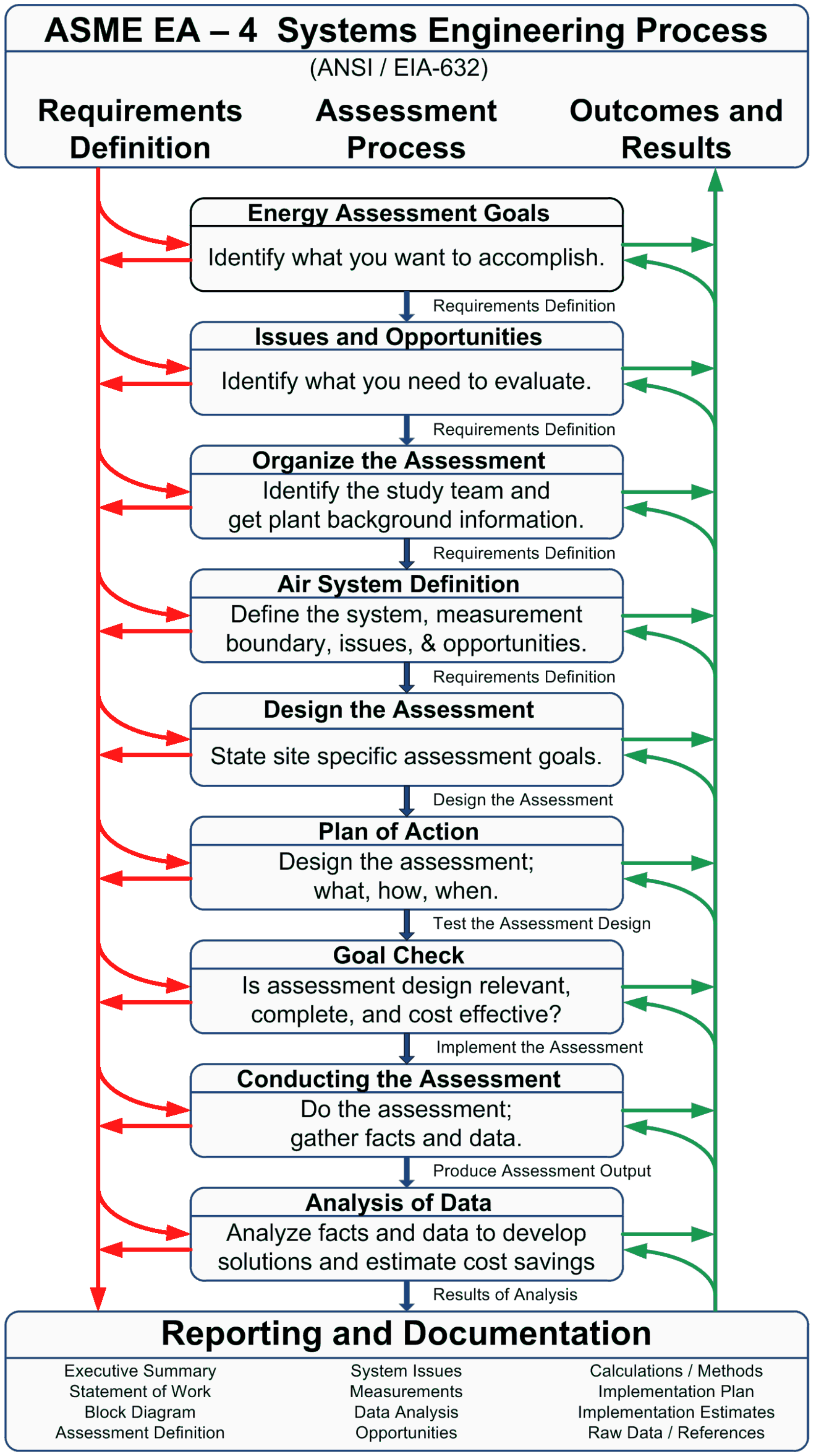



**Figure 10-1: Systems Engineering Process for Compressed System Assessment**

## 10.4 Energy Assessment Goals

There are basic informational objectives that are common goals in all compressed air system assessments.

- Baseline measurement of compressed airflow, and compressor energy use.
- Measurement of system pressure trends during the baseline period.
- Pressure profile information from the compressed air supply through key points in the distribution piping to end use applications.
- Characterize the current performance (flow, pressure, air quality) and operation of poor performing end use applications that cause production issues.
- Identify compressed air waste and inappropriate use and evaluate alternatives to eliminate waste.
- Understand the system's dynamic performance and supply / demand interaction to identify measures to create a consistent balance between supply and demand.
- Gather information to implement an effective control strategy to efficiently maintain the supply / demand balance for all of the plant's typical compressed air demand profiles.

Other informational objectives are more specific to individual plant situations. It is important to engage plant personnel and use their knowledge of the compressed air system performance, issues, and opportunities for improvement.

Understand that highly visible symptoms often mask the root cause of poor compressed air system performance and energy waste. A compressed air system assessment must look beyond the symptoms and identify the true performance issues. Evaluate the total air system to optimize overall compressed air system performance. Attempting to address individual parts of the system such as compressor control without evaluating issues of air storage, distribution and point of use can lead to incomplete analysis. Poor definition of system issues leads to treatment of symptoms. Failure to deal with the root cause of performance issues inevitably leaves the overall system with inconsistent, inefficient operation.

The worst outcome of a compressed air system assessment is to develop an elegant solution to the wrong problem.

***Reality is the supply of compressed air does not drive system performance or cost. If you never take any air out of a system, performance would be stable and cost would be minimal. The determination of both performance and cost is how the compressed air gets out of the system, not how it gets in.***

Controlling the system to a lower, stable operating pressure ultimately achieves the best system reliability. The lower stable operating pressure also results in the lowest energy consumption.

In compressed air management terms this is called balance. When the supply of air (energy) matches the demand, performance is stable and the least energy is consumed.

## 10.5 Compressed Air System Dynamic Performance

The energy consumption of most air systems is constantly changing as air uses start and stop. When a compressed air demand starts, energy demand in the system increases almost instantly. Most air systems are a combination of many air use points. In many systems hundreds of air demands operate in random fashion. The airflow rate consumed by individual air demands may range from a few to several hundred SCFM of compressed air. Individual air demands may last from a few seconds to several minutes or even hours. Compressed air systems are very dynamic with rapidly changing airflow requirements. Therefore, understanding the time based response of airflow to the system is critical for proper performance assessment.

The use and distribution of compressed air determines the dynamics of a system. Definition begins with knowing true pressure requirement of end use applications. In addition the size and frequency of demand swings affect dynamics. Event loads rapidly change system energy use. The distribution network often limits energy delivery.

## 10.6 Issues and Opportunities

Many plants have compressed air systems that have problems with performance and reliability. Issues commonly described are low pressure (either system wide or in localized areas), air quality problems (water and / or oil in the compressed air), interruption or malfunctioning production equipment, among others. Interview facility specialists with-in the plant and determine stakeholder's needs. Identify opportunities in various system areas, components, equipment, and processes that should be evaluated during the system assessment.

It is also important to determine what if any compressed air problems have occurred in the past, and what have been corrective actions have been taken. Very often in the middle of problems with the compressed air system that are interrupting production, immediate solutions are necessary. These immediate solutions may not be the most energy efficient or cost effective solutions. The solutions may be intended as temporary measures, but ultimately become permanent once the production issues are resolved. Look for more energy efficient solutions that may be available.

## 10.7 Organize the Assessment

Identify members of the plant assessment team and define their roles and responsibilities before, during, and after the compressed air system assessment. Gather plant background information including the estimate for annual energy cost of operation described earlier in this section. Other information related to the production process, various functional areas of the plant that use compressed air, operating schedules, production levels and more. What is the plant's present cost of electrical energy, are there demand charges, seasonal rates, or other factors that impact the cost of energy?

Have any compressed air system studies been done in the past? It so what were the findings, recommendations, what if any remedial measures have been implemented? Is maintenance and operating information available?

## 10.8 Air System Definition

Identify the measurement boundary of the assessment. This will normally be from the energy input to the supply side, through the compressed airflow, pressure, and air quality leaving the system supply and the production output of the facility.

Identify the existing compressor controls and intended control strategy with compressor sequencer or centralized automation control. Document the existing air receivers, primary and secondary along with pressure / flow controls, air storage refill control valves, check valves, or other key system components.

Complete a block diagram drawing of each compressor area including, compressors, air dryers, filters, and air receivers. Note the manufacturer, model, and capacity rating for all air compressors, air dryers, and filters.

The interconnecting piping should have the pipe size of each segment identified. Isolation valves, by-pass valves, and check valves if installed should be included in the block diagram.

Demand side pant layout diagram, gather Plant Floor Plan drawings or a plant layout sketch. Include information showing the location of significant air demands. Air demands considered should include; production critical air demands, pressure critical air demands, high volume intermittent demand events, and perceived high pressure air demands. Create an inventory list of air demands including those identified to have operating issues and opportunities for improvement or both.

## 10.9 Design the Assessment

Create a prioritized list of site specific assessment goals. List system performance information required to achieve the assessment goals. Identify the performance measurements and data necessary to gain the required information.

Develop the assessment measurement plan including parameters to be measured, the data sample rate to be recorded, and duration of time for various measurements. Identify the physical location for each measurement point identified in the measurement plan. Identify the type of measurement, the location, and a unique ID for each measured parameter.

The table below is an example of how various measurement points may be identified.

**Table 10-1: Uniquely Identify each Test Measurement Point**

| Measurement | ID | Description |
|---|---|---|
| Test Flow | TF1 | Air flow in 6” header leaving the Compressor Room |
| Test Pressure | TP1 | Air pressure in 6” header leaving the Compressor Room |
| Test Dew Point | TD1 | Air pressure dew point in 6” header leaving the Compressor Room |
| Test Amperage | TA1 | Compressor #1 Package Amperage taken at Disconnect Box |
| Test Power kW | TK1 | Compressor #1 Package Power taken in the Compressor Panel |

Plan for installation of measurement taps, compressed air flow and pressure measurement locations require installation of piping hardware in preparation for the on-site work. Advise team members as to the proper installation and location of the necessary pipe fittings at each measurement point. Installation of other transducers such as electrical power, or amperage measurement may require the assistance of skilled trade personnel such a electricians. Coordinate the availability of personnel and equipment to assist.

## 10.10 Plan of Action

On-site study work will require access to the various measurement points. It is necessary to arrange for the availability of personnel and equipment to assist the study team with access and connection of measurement equipment. For example the assistance of plant electricians may be required to install electrical measurement equipment. Flow and / or pressure measurement locations may require a man lift or ladder to gain access. Plant work rules may require a trades person to accompany the audit team while equipment is installed. The necessary arrangements and plans must be made before on-site study work begins.

Assign preliminary schedule dates for on-site study work considering the time necessary to gather the required information and complete the measurement preparation work. Allocate time necessary to assess each application shown on the inventory list of air demands including those identified to have operating issues and opportunities for improvement or both.

Decide on individual's roles and responsibilities during the system assessment. What action items are planned, who, how, and when will these be accomplished.

## 10.11 Goal Check

Now that the entire assessment plan is complete, review and compare the plan to the original assessment goal. Consider the following check list adapted from Systems Engineering Measurement Primer[6] and may provide guidance in reviewing the assessment plan of action.

**Relevance.** Why perform this action item? Is there more than one reason for this action item? Is it a result of ambiguity in the related assessment objective? Only select action items that are pertinent to an objective to be obtained.

**Completeness.** Are goals, stakeholder's needs, and system requirements being met? Has any key parameter needed to analyze data and achieve results been omitted? Has a balanced set of objectives among supply, distribution, and demand that adheres to the systems approach been identified? Is the assessment sufficiently comprehensive?

**Timeliness.** Can the system assessment meet the required time schedule? Be sure data collection, analysis, and reporting will provide results in the time allowed. Is more time required or should the SOW be modified?

**Simplicity.** Can data be collected and analyzed easily and cost effectively? Will the assessment produce results that can be presented in a manner such that stakeholders will understand what it means?

**Cost Effectiveness.** Is the budget sufficient? Will the system assessment provide more value than it costs? Is the SOW appropriate, or should it be modified?

**Repeatability.** Will the same plant operating conditions provide the virtually the same data and information twice? Are accuracy and precision adequate? This is important for future use of the system assessment's baseline measurement.

**Accuracy.** Are objectives, action items, methodology, and the resultant data relevant to system assessment goals? Are proposed measures reliable, and are measurements being made at the appropriate time? Measures should be accurate, and the resulting analysis should accurately serve the intended purpose of making the measurement.

[6] International Council on Systems Engineering (INCOSE), Systems Engineering Measurement Primer Measurement Working Group Technical Paper, version 1 March 1998

Is the proposed assessment cost effective? Does the potential economic benefit justify the assessment cost and still leave room for investments that must be made to implement various energy efficiency measures?

If the assessment plan falls short or does not meet the requirements listed here, then consider modifying the assessment goal and scope of work to achieve the requirements outlined here.

## 10.12 Conducting the Assessment

Coordinate implementation of the assessment measurement plan. Deploy temporary portable measurement equipment as planned and be sure to verify that the measurements appear to be reasonably correct. If there are obvious errors in the measurement's results attempt to troubleshoot and correct problems that exist. Also coordinate data collection from permanently installed instrumentation from existing plant systems. Again verify that the data is reasonably correct.

Gather facts and data as outlined in the compressed air system assessment plan of action. Recognize that a system assessment is a fluid process that may be changed with modified goals, objectives, and action items as the assessment unfolds. During the on-site assessment work, review the assessment status in regular meetings with key assessment team members.

## 10.13 Analysis of Data

Before data analysis begins validate that all data are reasonable and correct. When making a lot of measurements during an assessment, there are times when some portion of the data is erroneous, incorrect, missing, or otherwise invalid. If this occurs, first consider if the data can be corrected, interpolated, indirectly calculated using reasonable methods. If data cannot be used the assessment team should consider the overall impact on the assessment's results. If necessary the team may have to create an additional measurement plan to get the necessary data.

With all data having been validated, create the various profiles, and information consistent with the established assessment goals. Analyze facts and data to develop energy efficient solutions with estimated savings potential. Compressed air systems have many interactive effects between all components in the system. Therefore, remedial measures should be considered as groups of measures with appropriate consideration given to interactions with-in the system.

It is essential that improvements be a robust combination of multiple measures that will produce the desired result in a reliable, manner that will produce sustainable savings. If the energy efficient solution is perceived to reduce system reliability or in any way negatively impact the production process, the energy efficient solution will quickly be abandoned in favor of the original less efficient but more well know operating method.

## 10.14 Reporting and Documentation

Reporting should include a brief Executive Summary that is a summary of plant operations, assessment goals, findings, recommendations, and projected results. Finally briefly list recommendations for implementation with estimates of energy and cost savings.

More detailed reporting should include a description of the assessment process, the goals and scope of assessment activity. Describe systems and equipment studied and opportunities for improvement. Outline data analysis methods and how the assessment's conclusions were attained. Report baseline performance and projected energy savings opportunities.

Appendices, attachments, and data files should provide a complete record of assessment activities, data, and findings. These data should be organized and structures so as to facilitate third party review by knowledgeable personnel that may in the future review and verify energy savings projections.

## 10.15 Common Compressed Air System Assessment Mistakes

- An air compressor power study is not an air system assessment.
- It can be tempting to study only compressor power usage. After all the power used by the compressor determines the cost of running the air system. Not really, how the air is distributed and used after it leaves the compressor determines the cost of operating the compressed air system.
- Return on investment for air system improvement is a result of many factors. Power cost reduction through energy savings is an important component. However, other factors including improved productivity, increased quality, and avoided capital cost are potentially important.
- Always use the systems approach.
- An air system assessment designed to prove a point usually will.
- Beginning an air system assessment with a predetermined list of system solutions inhibits the discovery process. If the goal is to cost justify a predetermined solution the assessment is a waste of time and money.
- Be sure the assessment is a discovery process, what do I think, what do I know, what can I prove.
- Controlling leaks is not controlling the system.
- Performing a leak survey should not be confused with conducting an air system assessment. Leaks are a significant portion of energy waste in most air systems. A leak survey is often an important component of an air system audit.
- Attempting to control leaks without controlling the system pressure provides minimal improvement.

- An effective compressor control strategy is essential to convert the airflow savings of leak repair to kW reduction that will actually save money.
- Drawing the distribution piping doesn't define performance
- It is desirable to have good documentation of compressed air piping within a facility. The most important concern is how well the distribution system performs. Traditional design recommends piping loop mains as the best distribution method. The fact is most systems have evolved over time and are not designed as a new system. It is also true that while these systems look ugly on paper, they often perform adequately.
- Mapping out the distribution system as part of an air audit may be desirable. Redesigning, and piping a system so it follows traditional design and looks good on paper can be a waste of time and money. The fact is the air distribution system either works or it doesn't.
- Be sure to get a good measurement of the system's pressure profile

# 11. Data Collection & Analysis

## 11.1 Developing a Compressed Air Profile

Developing a compressed air system profile requires performance measurement of the existing air system. The measurement system encompasses everything from the connection point in the compressed air system to the recorded data of physical parameters being measured. For example a pressure measurement system might consist of a pipefitting, pressure gauge, observer, wristwatch, pencil, and note pad.

In this discussion, the compressed air measurement system will consist of:

- Connection to the system
- Sensors, transmitters, and transducers
- Wiring, cables, connections, and switches
- Data acquisition hardware and software
- Measurement techniques

Objectives in developing an air system profile should focus on the information that is desired, not on the raw data. For example, measuring the airflow is not an objective. Defining the plant air consumption profile on a typical weekday and on Saturday is an objective. The data necessary to accomplish this objective is measured flow, power, and pressure delivered to the system over the time period to be evaluated.

The supply side operational baseline should be quantified in detail. Supply side performance should be identified in terms of pressure variation, average airflow, and peak airflow rate. Hourly average profiles should be generated to calculate annual operating cost from measured baseline data. Supply side profiles for typical days of operation including characteristic demand periods should be developed.

Begin developing a plant air demand profile by taking the following measurements:

- Measure the energy (kWh) consumption of an air compressor hourly throughout typical days.
- Measure the airflow delivered by the compressor, sump pressure, discharge pressure, and Power (kW) of the main drive motor.
- Measure the control signal pressure.

A plant air demand profile provides a baseline for analyzing many aspects of system performance. The average hourly air demand swings throughout the day will be documented. Surge demands and transient events will be identified in terms of peak air demand and event duration. Measurements can be used to evaluate plant pressure stability and help define the initial target for controlled system pressure.

System profile data can be used to determine annual operating cost of the current method of operating the existing compressors. Part load response and compressor sequencing control should be defined. In multiple compressor installations, several machines often operate at part-load capacity. Multiple compressors operating at part load can greatly reduce operational efficiency. Frequently, there is a significant potential to improve the efficiency of the system. This can be identified through acquisition of dynamic flow and pressure performance data.

Once the technical objectives make clear what profile information is desired, a test plan should be developed. The test plan should include a list of monitoring points, and the parameters to be measured. Planning the sample rate, data interval and duration of data logging to be performed is also important. Measuring the supply side performance for hourly profile trends, flow, power, and pressure requires the transducer sample rate data to have a frequency that will capture system dynamics. Data averaging can be used to provide a 1-hour or other data interval to trend system performance. At the same time, data recorded at a shorter interval can be used to detect dynamic performance. For example, short interval data logging can capture transient pressure profile responses and identify significant demand events.

## 11.2 Determining Air Flow and Pressure Requirements

Data logging the system dynamics is an effective method to develop supply side pressure profiles for all characteristic demand periods, including transient pressure profiles resulting from routine demand events. Identifying dynamic responses of supply side controls to normal air demands and system demand events is also important.

The pressure profile should include demand side pressure fluctuations and identify excessive system pressure. An evaluation of the impact of existing pressure fluctuations on productivity should be done as well.

Key air demands should be evaluated through observation, short-term data logging, and spot-check one-time measurements. The compressed air demand profile, including correlation of baseline data from measured airflow rates to characteristic signatures of demand events should be studied and documented. Finally, summarize the analysis of demand classifications (peak, average, and continuous air demand) with cycle times.

**Table 11-1: Assessment Informational Goals & Measurement Locations**

| **Typical Compressed Air System Performance Measurements and Locations** | |
|---|---|
| **Informational Goal** | **Measurement and Location** |
| **Demand Profile: (Dynamic Response)**<br>Airflow demand events, continuous, average, and peak airflow with event cycle time and duration<br>Correlation of Characteristic Signatures to Demand Events | Airflow, at each system supply point<br>(Dynamic Response) |

| Typical Compressed Air System Performance Measurements and Locations | |
|---|---|
| **Informational Goal** | **Measurement and Location** |
| **Pressure Profile:**<br>Distribution Pressure Gradient and Variance (Dynamic Response) | Pressure at key points in distribution:<br>• known or suspected problem areas<br>• farthest points of the system<br>• near high volume intermittent demands<br>• near perceived high pressure demands<br>Spot check or short run data logging |
| **High Volume Intermittent Demand Events (Dynamic Response)**<br>Characteristic Signature, impact on system pressure profile, and distribution gradient.<br>Demand Profile, continuous, average, peak airflow, event duration and dwell time between events.<br>Pressure Profile, peak, valley, and rate of decay. | Pressure, at key points related to the demand event and production process:<br>• distribution header in the area<br>• equipment / process supply connection<br>• point of use<br>Airflow:<br>• point of use supply to the demand event,<br>• production equipment or process.<br>NOTE: correlation of the demand event characteristic signature to supply side response may eliminate the need for airflow measurement at the point of use. |
| **Perceived High Pressure Demands (Dynamic Response)**<br>Pressure Profile, to validate pressure requirements, and rule out excessive pressure drop at the point of use connections.<br>Characterize the demand's pressure requirement as flow static, of flow dynamic. | Pressure, at key points related to the demand event and production process:<br>• distribution header in the area<br>• equipment / process supply connection point of use |

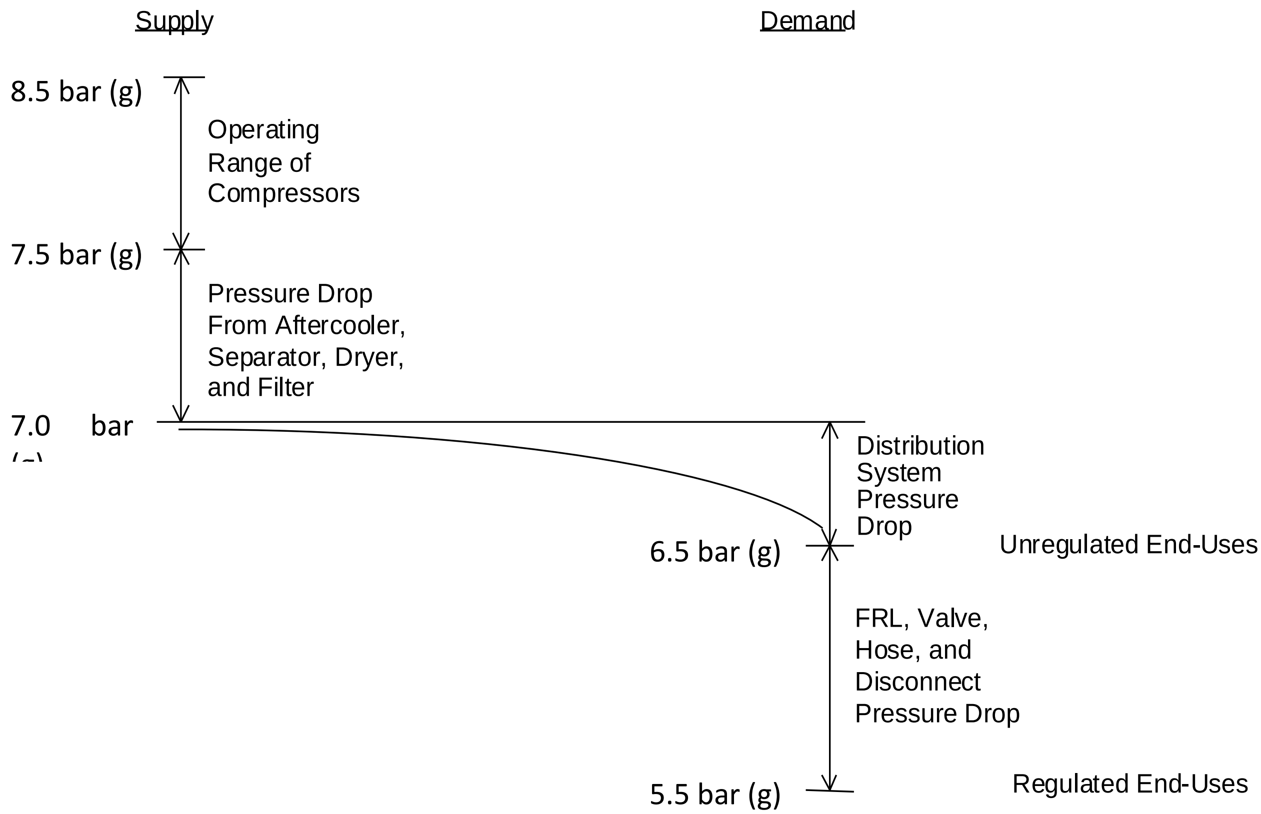



**Figure 11-1: System Pressure Profile**

A system pressure profile will help establish an understanding of exactly how the system is operating. In the profile above, the end use requires only 5.5 bar (g) in order to operate. This establishes a base from which a number of improvements can be proposed. There is a 2 bar differential between the bottom of the compressor control range and the pressure required by the users. If that 2 bar differential could be eliminated, the pressure at the compressor could be reduced. There is about a 6% power savings for each bar of pressure reduction.

If the pressure drop is intermittent, the most likely cause is a large demand event. This would indicate that additional compressed air storage might be required.

## 11.3 Data Analysis Exercise

Review the block diagram in Figure 11-2, which shows a typical air system in a plant manufacturing ceramic products.

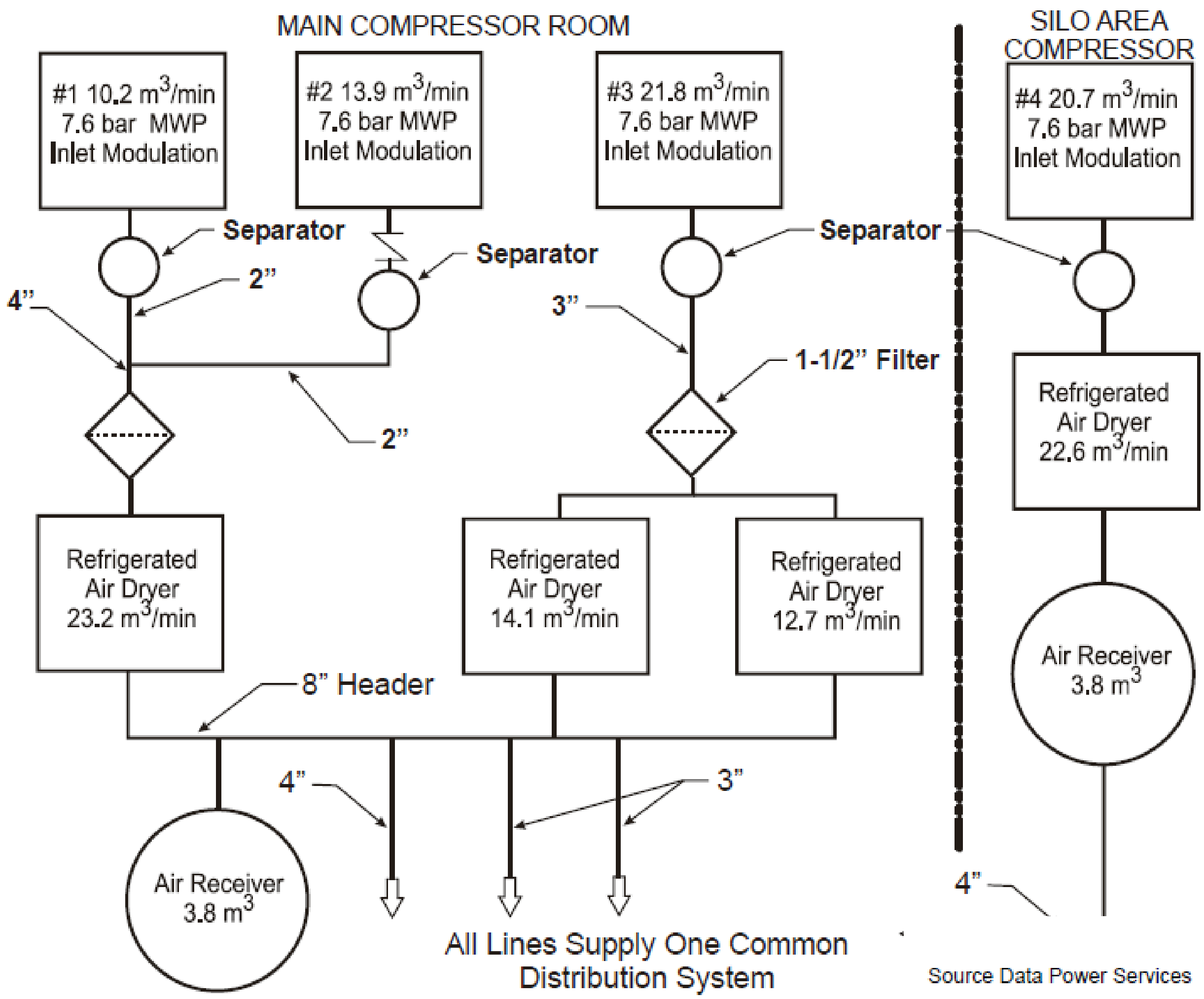



**Figure 11-2: Air System Block Diagram**

- Plant air pressure is currently acceptable. Plant supply pressure rises as high as 6.76 bar during periods of low air demand. However, the pressure routinely drops below 5.52 bar during peak daytime production.
- Poor air quality is affecting painting operations currently causing lost production, and product scrap. This is also believed to be the cause of additional production problems in subsequent coating operations.
- There is currently a plant expansion underway. The majority of the new space will be finished goods inventory and automated warehouse, order processing, and shipping operations. There is very little additional air demand associated with the expansion. However, space is available for a new compressor in the new addition. This would provide additional compressed air supply capacity in a third location within the plant.

- What technical objectives would you use to define an assessment of the plant's compressed air system?
- Develop a system plan to provide reliable consistent compressed air supply to production areas.
- Evaluate current compressor control pressure swings and operational sequencing.
- Establish a target pressure range for supply pressure, and define tolerance of acceptable pressure variations.
- Outline a new supply side system configuration to stabilize air supply pressure within defined pressure range and tolerance. Consider configuration and performance of supply from the existing compressors. Evaluate the application of additional compressors and accessory equipment as necessary.
- Define air distribution performance from the central compressor station out evaluating current pressure gradients that exist. Check control integration of the remotely located compressor in the silo area.
- Maintain prudent capital investment and a phase in remedial plan consistent with technical issues and business parameters.
- What is your measurement plan for this system? Include the type of measurement (airflow, pressure, etc.) and the location that the measurement should be made.
- Measurement points to define plant air demand requires connection of 4 flow transducers, three headers leaving the main compressor room and one at a remotely located compressor in the rail car unloading silo area. System pressure is monitored at the supply points in the compressor room and silo area. Compressor control signal pressures are monitored at each compressor in the main compressor room.

Measurement locations selected for system assessment are shown below.

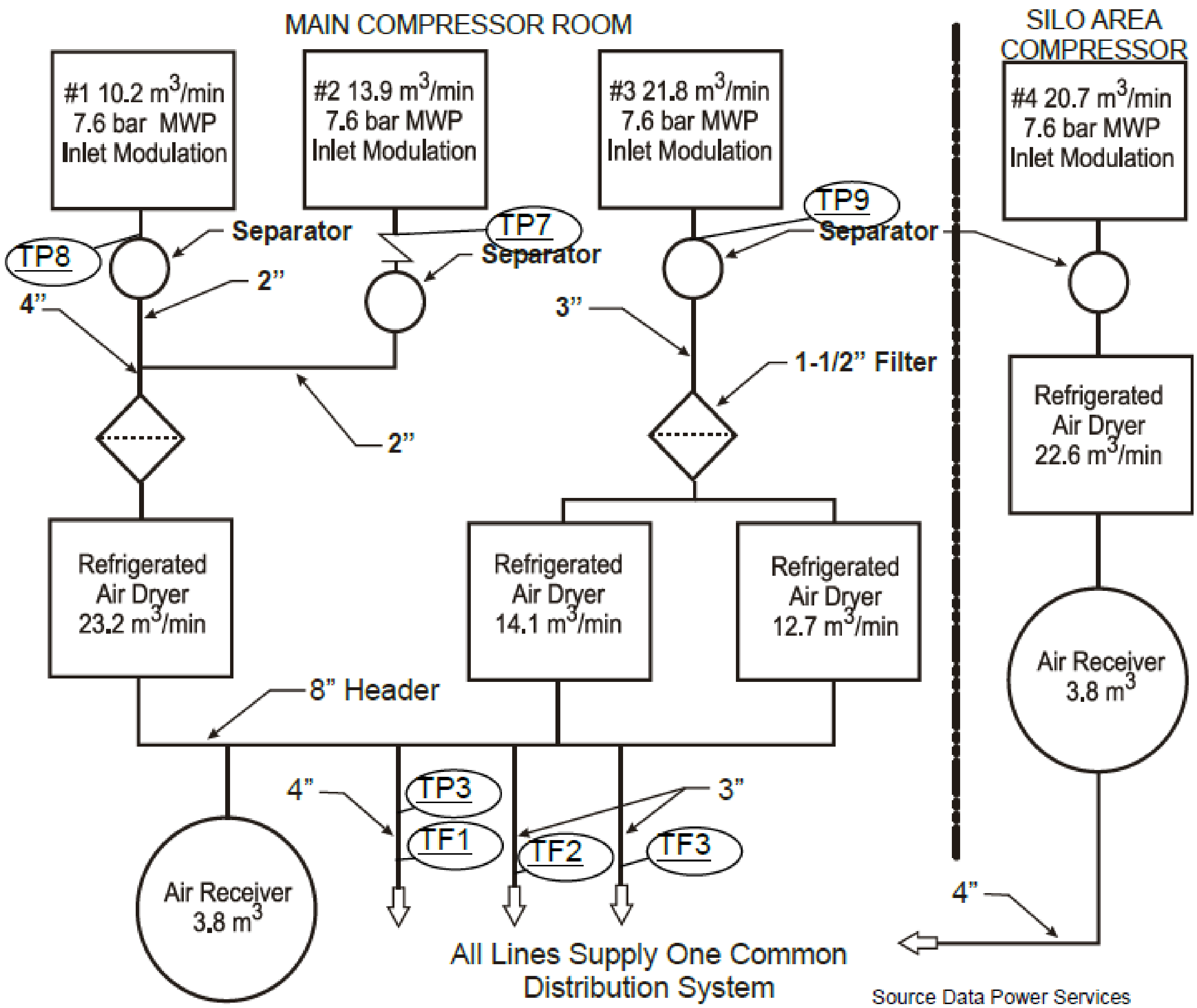



**Figure 11-3: Air System Block Diagram with Test Locations**

The data tracing in Figure 11-3 includes airflow and system pressure measurements that indicate the presence of a cyclic event. Furthermore, the flow data of individual headers shows one event on the 4" header (TF1) and another on the 3" header (TF2). Data also indicates that just after 7:15, an air demand came on-line changing the underlying continuous airflow at TF1 from 8.5 to 14.8 $m^3$/min.

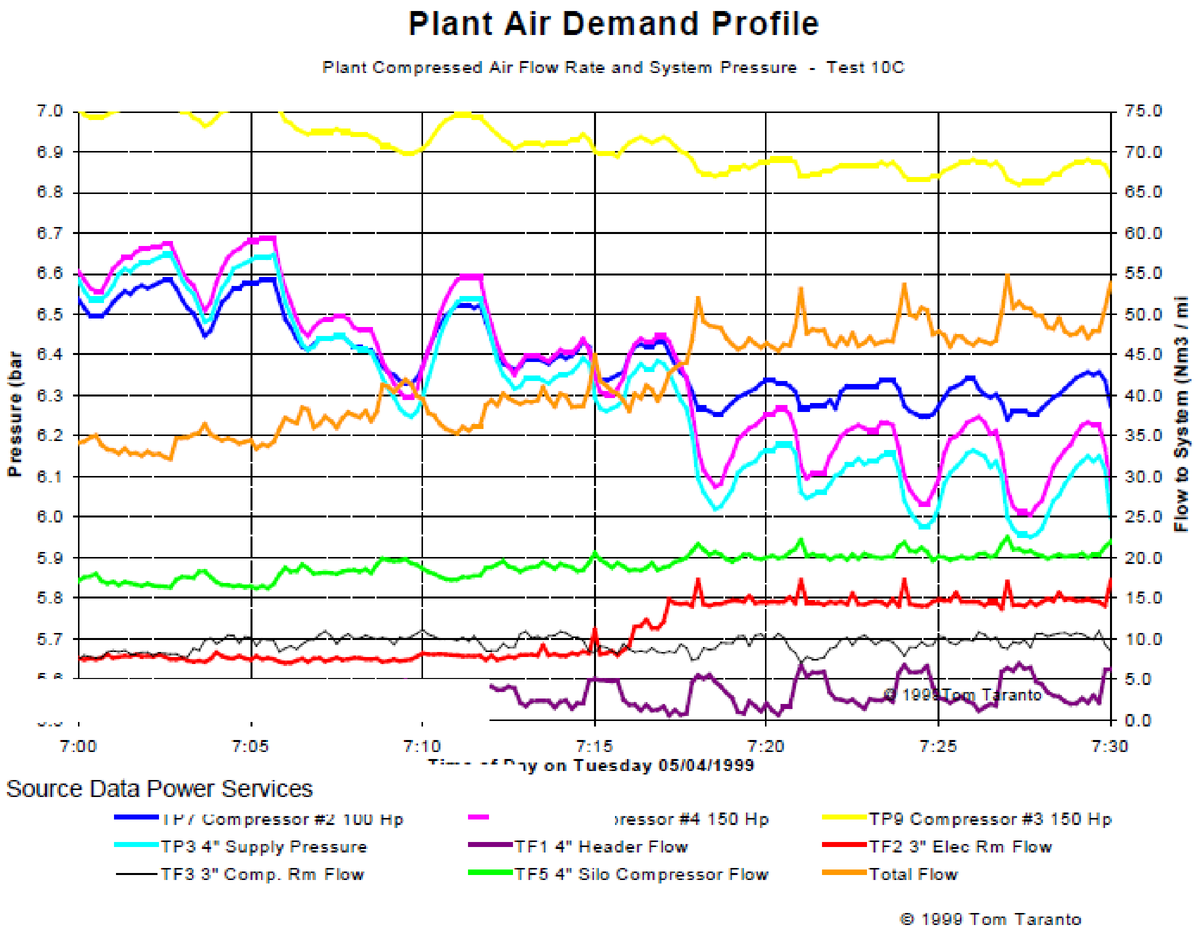



**Figure 11-4: Demand Profile & Characteristic Signature Demand Events**

When analyzing system events, it is important to evaluate the relationship between flow and pressure changes. The data above shows during the event as flow increases, pressure is decreasing. After the event ends and flow is decreasing pressure increases. This clearly defines a demand event. On the other hand, if flow and pressure are both increasing and both decreasing, that suggests a supply side event where compressor capacity is increasing or decreasing respectively.

Repeatable demand events will normally exhibit a Characteristic Signature that can be identified. The drawdown rate, amount of drawdown, and event duration form the characteristic signature for demand events. Different demand events will exhibit individual characteristic signatures.

Accurate identification correlates characteristic signatures with known demand events. Once identified and characterized the systemic effect of demand events can be evaluated. Demand events may be the source of transient localized or system wide pressure upsets that impact other compressed air uses. Also demand events may initiate compressor start-up and / or loading, when additional air storage may stabilize the system and reduce compressor run time.

Investigating Pressure & Flow Dynamic Profiles is important to:

1. Assess the system's supply / demand balance and compressor control Strategy.
2. Identify Characteristic Signatures for normal system events.
3. Characterize system drawdown rates, the magnitude, and duration of drawdown events.
4. Quantify transient supply deficits and evaluate the benefit of increased air storage on system stability and reduced compressor run time.
5. Correlate known demand events with demand side performance upsets and supply side control response.

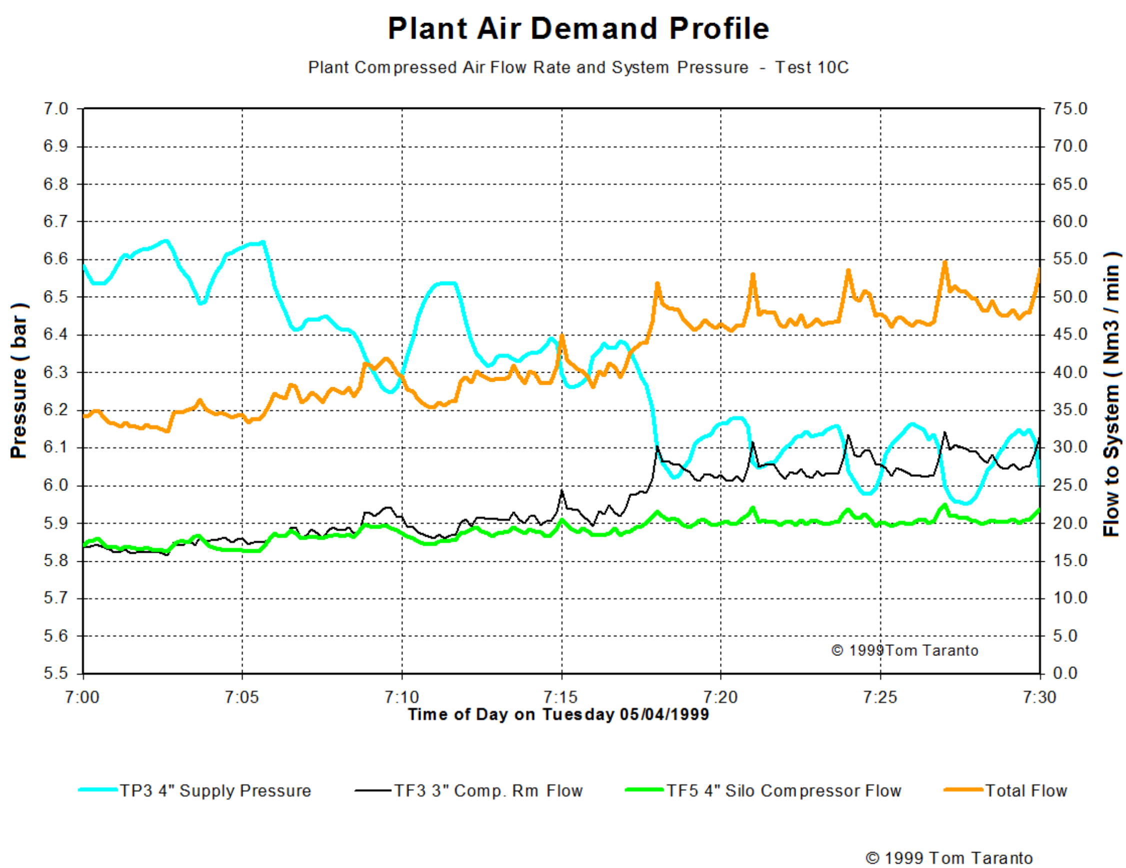



**Figure 11-5: Supply Side Response to the Demand Profile**

### a. Resulting Supply Side Control Response & Supply / Demand Balance

In addition to the presence of event loads, the air system flow data in Figure 11-5 shows total air demand ranging from 35.0 to 55 $m^3$/min. The operating compressor capacity is 66.6 $m^3$/min. Converting to $Nm^3$/min, the compressor capacity is 64.3 $Nm^3$/min. That represents 11.6 $Nm^3$/min more compressor capacity than the total air demand.

The total air delivery from the main compressor room ranges from 17.0 to 31.0 $Nm^3$/min, while the operating compressor capacity is 44.3 $Nm^3$/min. In the main compressor room the machines are sharing the load operating between 42% and 64% of the combined operating compressor capacity.

## 11.4 Key Learning Points

1. Training and education must raise awareness of compressed air cost, opportunities to lower air pressure, and improve system performance.

2. Monitoring compressed air system performance provides necessary management information to keep the air system operating efficiently, and reliably.

3. In today's highly competitive global economy, timely compressed air system management information is essential.

## 11.5 Key Energy Points

4. Compressed air energy is a significant investment including capital, energy, maintenance, and productivity costs.

5. Multiple compressor systems can be very inefficient if not properly controlled.

6. Compressed air demand and pressure profile data can help identify potential areas for improvement.

7. Compressed air system assessment defines performance and current method operating costs.

8. Balancing system operation provides stable performance and reduces energy cost.

9. Inappropriate compressed air demands must be identified and replaced with more energy efficient alternatives.

10. Leak management, correctly sized distribution piping, and good point of use piping practice improves air application performance

11. Reducing system operating pressure to the lowest optimum pressure necessary to supply productive air demands, will reduce energy cost.

## 11.6 Data Acquisition sample rate, data averaging, and data storage interval

Analysis of compressed air system performance is only as good as the data that the analysis is based on. There are many factors that impact the accuracy of measured performance data.

- Sample Rate
- Data Interval
- Accuracy / Repeatability
- Electrical signals
- Scaling of engineering units

Dynamics is the study of the effect of time related events on system performance. The best approach to defining system dynamics is by taking measurements of the system performance in real time. Understanding that the measurement equipment must be capable of sufficiently high sample rates to capture the dynamic event being measured is important. For example, there is little benefit to measure the performance of an event with duration of several seconds by observation of the pointer on an analog pressure gauge. The mechanical damping of a pressure gauge does not respond quickly enough to provide accurate data. Similarly, a data logger that records performance a few times per minute cannot accurately characterize that same event.

**Sample Rate** (scan rate) – The sample rate is the time interval (T) in seconds, at which data acquisition inputs are scanned, reading the signal from the attached sensor or transducer.

**Data Averaging** – Data averaging is a data reduction method that reduces multiple samples to a single data point (averaging is common, although other data reduction methods including filtering can be used). Sensor samples (S) are read at a given sample rate (T) for a selected number of (n) samples. The average value of the sensor reading is calculated:

$$\boxed{S_{AVG} = \left(S_1 + S_2 + \ldots S_n\right)/n}$$

**Data interval** – The data interval is the frequency with which an averaged sensor reading is recorded as a measured data point. This is calculated from the sample rate (T) and Number of samples to average (n); [T x n]. For example, with a sample rate of 3 seconds and an average of 15 samples, the data interval is 45 seconds [3 seconds x 15 samples = 1 data point every 45 seconds].

$$\boxed{Data\ Interval = T \times n}$$

One purpose of using higher sample rates with data reduction is to minimize the effect of transducer Signal Noise on recorded data. For example, pulsations at the discharge of a reciprocating compressor can produce a noisy transducer output (unstable output signal). In this instance, a frequent sampling rate can be averaged to achieve a more stable reading. In the case of stable sensor signals, the sample rate may be relatively slow with averaging of fewer samples used to accomplish the shorter data interval. For example, to achieve a data interval of 10 seconds, a 1 second sample rate averaging 10 samples could be used. Secondly when the physical parameter (power, airflow, pressure, etc.) being measured varies with time, data averaging results in a more accurate trend data. Too frequent data storage on the other hand, can result in huge data files that must be manipulated to reduce the data into useful information.

### a. Hourly Trend Data

Hourly trend data can be used to develop the profile of compressor power, or flow data to calculate operating cost. Trend data will not, however, define dynamic performance. For example, an air system has single-stage, lubricant-injected, load/no-load compressors. The trim compressor is operating with a 90 second load period (the time between the start of one compressor load cycle and the start of the next compressor load cycle). Consider the following two data acquisition methods for a power measurement profile:

**Table 11-2: Acquisition Methods**

| | **Method #1** | **Method #2** |
|---|---|---|
| **Sample Rate** | T = 5 minutes | T = 1 second |
| **Samples to Average** | n = 12 samples | n = 3600 samples |
| **Data Interval** | 60 minutes ( 1 hour ) | 3600 seconds ( 1 hour ) |

Both data acquisition methods, shown above, will result in recording 1 data point per hour, as required to trend operating cost.

Questions:

1. Do you think the average air compressor power readings recorded by the two methods above would have the same value? Why or why not?

2. If not, which of the two methods do you think would result in more accurate power information? Why?

3. What if the trim compressor were a lubricant injected modulating rotary screw compressor?

4. What if the compressor being measured is a base load compressor operating at full load for the entire hour?

## b. Dynamic Response

When the actual time base of the measured physical event is shorter, the data interval must decrease to properly characterize performance. Transducer response time is an additional time-based variable that can affect data collection. Response time is discussed further in the section on Transducer Signals, Response Time, Noise, and Interference on page 226. The data chart below shows a comparison of different sampling rates and data intervals. Inappropriate data collection misrepresents compressor load cycles.

**Table 11-3: Comparison of different sampling rates and data intervals**

| | High Rate | Slow Rate |
|---|---|---|
| **Sample Rate** | 1 sample per 1 second | 1 sample per 3 seconds |
| **Data Averaging** | 10 samples | 15 samples |
| **Data Interval** | 10 seconds | 45 seconds |

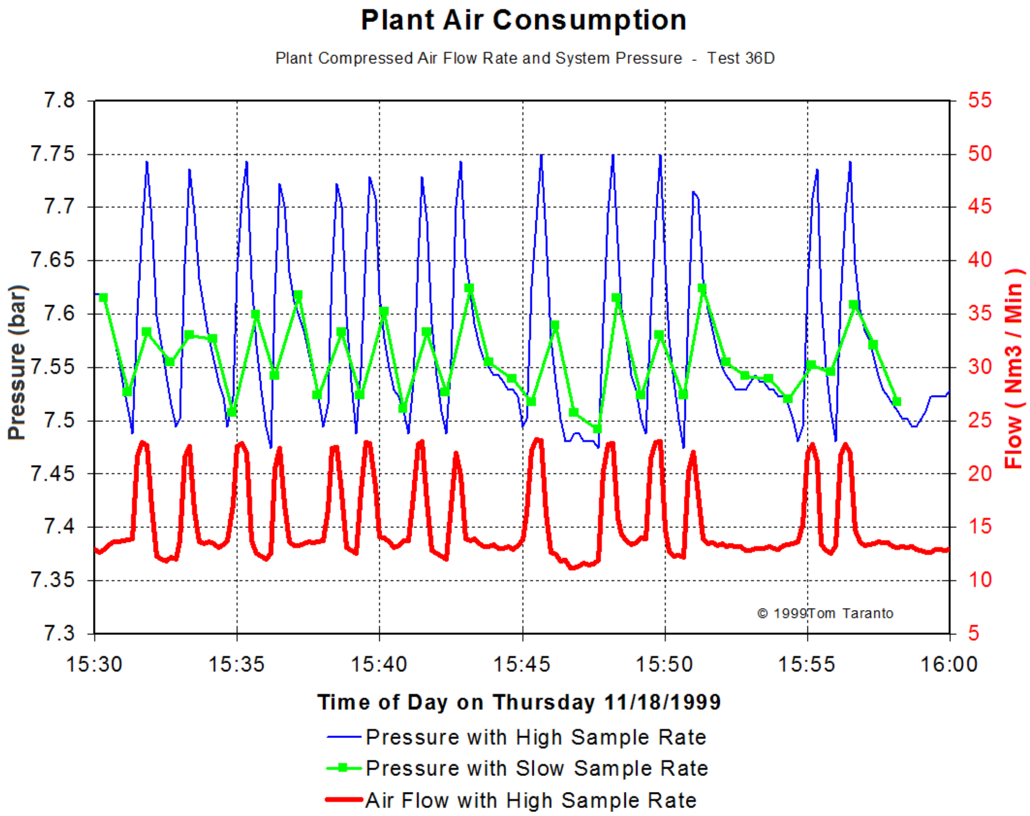



**Figure 11-6: Compressor Load / Unload Cycling**

The data chart in Figure 11-6 shows the dynamic response of the load/unload cycling of an air compressor. Along the X-axis is time showing a total span of 30 minutes. There are two air pressure tracings, scaled on the left hand Y-axis, and one airflow measurement scaled on the right hand Y-axis. Pressure measurements were taken at two different sample rates and data intervals as shown in the chart below.

The period of compressor load cycles in the chart is 90 seconds (1.5 minutes). Data shown for the high sample rate is reading a pressure value once per second and averaging 10 samples for each data value stored. The low sample rate is reading a pressure value every 3 seconds and averaging 15 samples for each data valued stored.

When the sample rate and data interval are too low (or slow) as compared to the time period of the underlying event, signal aliasing can occur.

**Signal Aliasing** – This term describes a condition when data logging records a false waveform that does not accurately represent the true characteristics of the parameter being measured.

Signal aliasing in the data tracing above could easily lead to misinterpretation of the recorded system dynamics. The 10 $Nm^3$/min swing in airflow to the system is NOT a demand event. The true waveforms, collected at the high sample rate, clearly show a direct correlation of increasing flow with increasing pressure. Increasing flow with increasing pressure is due to a compressor load cycle. Increasing flow with decreasing, or no pressure change, is due to a demand event.

The alias pressure tracing between 15:30 hours and 15:35 hours might be interpreted to show increasing flow with decreasing, or no change in pressure. The slow sample rate and alias pressure tracing could lead to the conclusion that the flow increase is a demand event.

There are no set values for sample rate and data interval. Rather they must be determined by the system performance and information that is desired. If dynamic system response is to be evaluated, the nature, and time base (dt) of the underlying event must be considered when determining an appropriate sensor sample rate and data interval.

The first step is to select an appropriate data interval. To capture the characteristics of an event requires recording a minimum of 3 data points during the event. In the example, the compressor's loading period is 90 seconds. However, the shortest event is the loaded time loaded time of 30 seconds. The unloaded time is 60 seconds. Therefore, the time base (*dt*) for load cycle data is 30 seconds.

To properly characterize the compressor load cycle requires at least a 10 second data interval, calculated as the time base divided by number of data points as shown in Equation 11-1.

**Equation 11-1: Solving for Measurement Data Interval**

$$
\begin{aligned}
&\textit{Event to Measure} \;=\; \textit{Compressor Load Cycle}\\
&\textit{Number of Data Points During the Event} \quad n = 3\\
&\textit{Time Base of the Event} \quad dt = 30 \textit{ seconds}\\
&\textit{Data Interval} \;= dt \div n \;=\; 30 \textit{ Seconds} \;\div\; 3 \textit{ Data Points} \;=\; 10\,\frac{\textit{Seconds}}{\textit{Data Point}}
\end{aligned}
$$

Although 3 data points are the minimum necessary to characterize a dynamic event, 5 to 10 data points during the time base of the event are preferred. A 30 second compressor load cycle is best captured with a data interval of 3 to 6 seconds.

Note: Recommendations of the IPMVP (International Performance Measurement and Verification Protocol), ASME EA-4, and ISO 11011; are that the data interval shall be one order of magnitude greater than the time base the event being measured. For example, a 1 second event should be measured at a 0.1 second data interval; which is one order of magnitude faster than the event.

**Question:** Why not just record all compressed air system performance at a frequent data interval of say 1 data point per second?

## 11.7 Selecting Sensors to Support the Informational Goals and Measurement Plan

The informational objectives and measurement plan define the physical parameters, i.e. pressure, power, energy, airflow, etc. to be measured. Data are raw numbers. For example, installing a pressure transducer in an air system, and connecting the signal to a data logger will result in gathering numbers. These data represent physical parameters and time based response of the compressed air system. The performance and application of sensors affect the recorded data.

**Sensor** – A sensor is a hardware component that measures a physical parameter such as pressure or flow and converts it to another form of energy. For example, a strain gauge is a sensor that detects small mechanical displacements, and outputs changing electrical resistance.

**Transmitter** – A transmitter is a hardware component that converts a variable from one energy form to another. For example, a transmitter converts the changing electrical resistance of a strain gauge sensor used to measure pressure to a linear output signal of 4 – 20 mA. In telemetry technology, a radio transmitter is another type of transmitter that changes an electrical signal to a radio broadcast signal.

**Transducer** – A transducer is a hardware component that measures a physical parameter and outputs an electrical signal that is conveniently used in measurement systems. For example, a pressure transducer may be constructed with a strain gauge sensor and built-in transmitter to provide a linear 4 to 20 mA output signal scaled from 0 to 16 bar (0 – 232 psig) range. A transducer includes both a sensor and a transmitter.

In measurement and instrumentation technology the terms "Transmitter" and Transducer" are often used interchangeably. Because transmitter is an ambiguous term, it should generally be avoided. Transducer is the preferred terminology.

### a. Accuracy – vs. – Repeatability

**Accuracy** – Accuracy is the ability of a particular gauge, sensor, or transducer to measure the value of a physical parameter as compared to the true value based on a known standard.

**Repeatability** – Repeatability is the ability of particular gauge, sensor, or transducer to repeat the same measured value.

NOTE: Accuracy requires repeatability since you cannot have accuracy without having repeatability. However, repeatability (sometimes referred to a precision) does not guarantee accuracy.

### b. Accuracy Specifications and Potential Measurement Error

Accuracy specifications for sensors, or transducers are generally expressed as percent error. The percent error is

$$\% error = \frac{true\,value - measured\,value}{true\,value} \times 100\%$$

Accuracy specifications are typically expressed as a percent of the full-scale (FS) reading of the sensor or transducer. The accuracy of a general-purpose pressure transducer might be ± 1.0% of the full-scale range. For a 0 to 15 bar range pressure transducer, a reading ± 0.15 bar is within the specified accuracy. Measuring the pressure differential with two pressure transducers could indicate as much as 0.3 bar variation (± 0.15 bar) when in fact the pressure difference is equal to zero. The example shows that using transducers with greater range increases error. A pressure transducer with a smaller full-scale range could be used to reduce the error.

NOTE: Some sensors or transducers have accuracy specified as a percentage of reading rather than as a percent of full scale.

### c. Transducer Signals, Response Time, Noise, and Interference

Signals generated by sensors or transducers measure a physical parameter and output the measured parameter as an electrical signal. Many types of electrical signals are possible. The most common types are millivolt (mV AC or mV DC), DC voltage, and milliamp (ma) signals. The objective is to ultimately correlate the electrical signal to appropriate engineering units for the physical parameter being measured.

The response time of a transducer is determined by sensor and transmitter design. Response time specifications usually indicate the time required for a designated signal change. For example; "Response Time – 100 ms from 10% to 90% range", indicates the output signal will respond within 0.1 seconds to a change in measured parameter within 10% to 90% of the transducers operating range. If response range is not specified, the response time applies to any measurement within the sensor's operating range. Some transducers have a programmable response time. So called "smart transducers" may allow setting operating parameters to filter, dampen, or average the output signal. This is a feature that can be effective in reducing signal noise. The user should be aware of the impact of signal damping on response time, and confirm that it is appropriate to the measurements being made.

Noise and Interference arises from various operating conditions that affect the electrical output signal from sensors and transducers. Signal interference is generally referred to as noise and affects the stability and accuracy of measurements.

**Table 11-4: Sources of interference or signal noise**

| Type of Interference | Possible Source | Possible Remedy |
|---|---|---|
| **Vibration** | Mechanical vibration of the sensor or transducer | Relocate or change the transducer mounting to isolate from vibration |
| **Electrical** | Capacitive or Inductive Interference | Change the type, size, or location of signal conductors. For example use shielded cable, or twisted pair for signal wiring. |
| **Radio Frequency** | Radio transmitters, electrical arcing, switching power supplies, fluorescent lights | Relocate RF source or move the transmitter and / or signal wiring location. |
| **Ground Loop** | Closed electrical path in the system ground connections | Ground the system at one single location |
| **Floating Source Interference** | Multiple DC power sources and / or batteries | Eliminate common voltage differences or Install signal isolation |

Floating source interference may occur when multiple DC power supplies, multiple batteries, or a combination of DC power supplies and batteries are used. Multiple transducers for flow, pressure, power, and dew point performance of compressed air systems may be connected to a single computer or data logger for the purpose of recording system operation. Very often, the transducers will be some combination of devices, possibly 2-wire or 4-wire, with current or various voltage outputs.

When multiple DC power sources are used, it is possible for voltage differences to exist between the various negative or "common" references. A voltage potential between the signal ground and the Analog/Digital converter reference ground will produce an error. The data chart below shows the error offset that is measured with a 4-wire mass flow transducer commonly used for compressed air measurement. This data logger is powered by a battery, and the flow transducer is powered by a DC power supply.

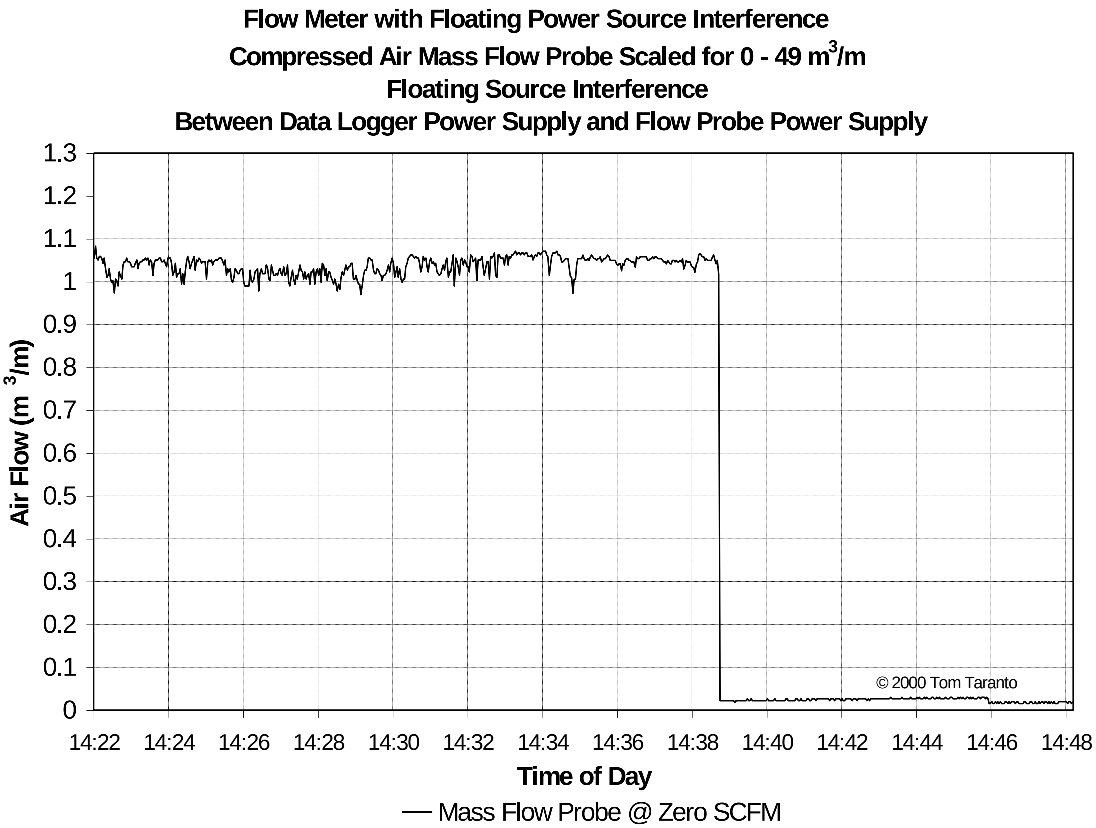



**Figure 11-7: Flow Meter with Floating Power Source Interference**

**! CAUTION –** Improper grounding of power supplies, batteries, transducers, and data acquisition systems can permanently damage equipment.

## d. Pressure Transducers

The data chart below shows the variation in measured value of four general-purpose 0 to 14 bar (g) pressure transducers with 1% FS (Full Scale) accuracy specification. The four pressure transducers are connected to a common pressure signal that measures control signal pressure during load/unload operation of a compressor. As expected, the variation of pressure reading is 0.28 bar (g) (± 0.14 bar).

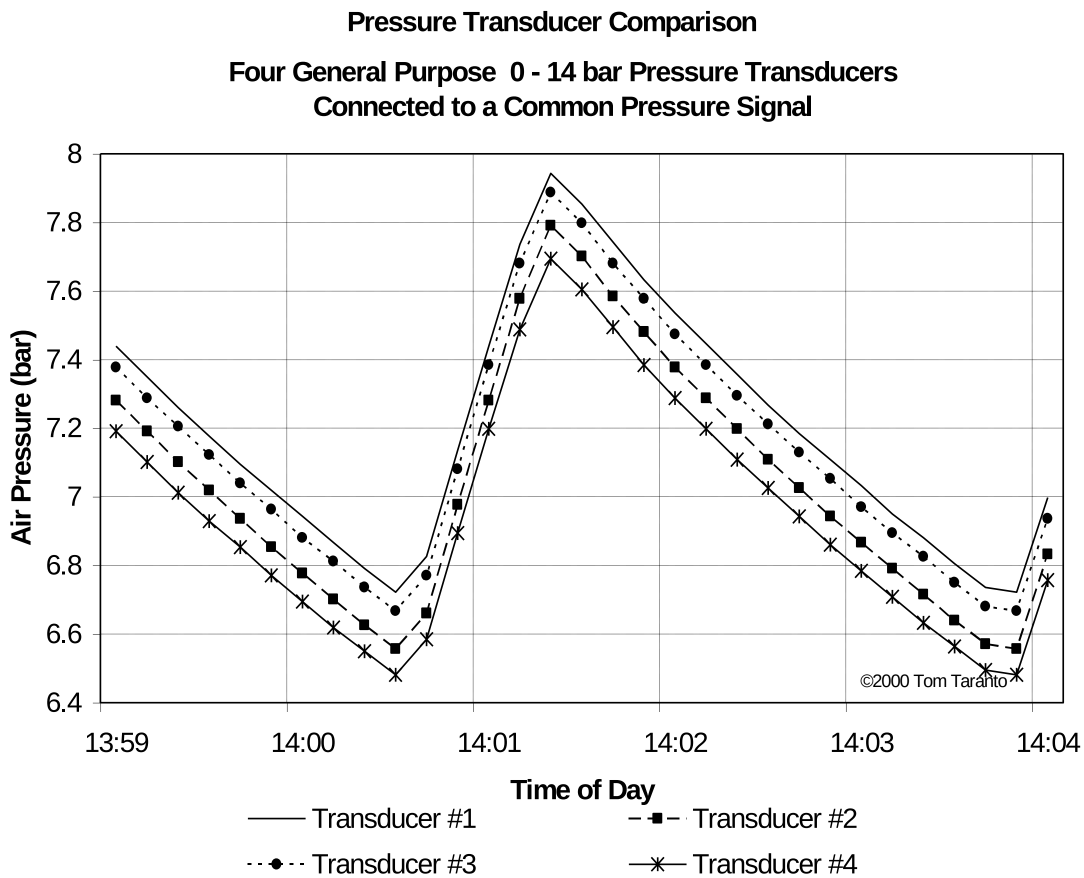



**Figure 11-8: Pressure Transducer Comparison**

By comparison, pressure transducers with 0 to 14 bar(g) range and ± 0.15 % FS accuracy are expected to read a common pressure signal within 0.04 bar variation (± 0.02 bar). Data below shows the result of measurements made with four more accurate (± 0.15 % FS) pressure transducers.

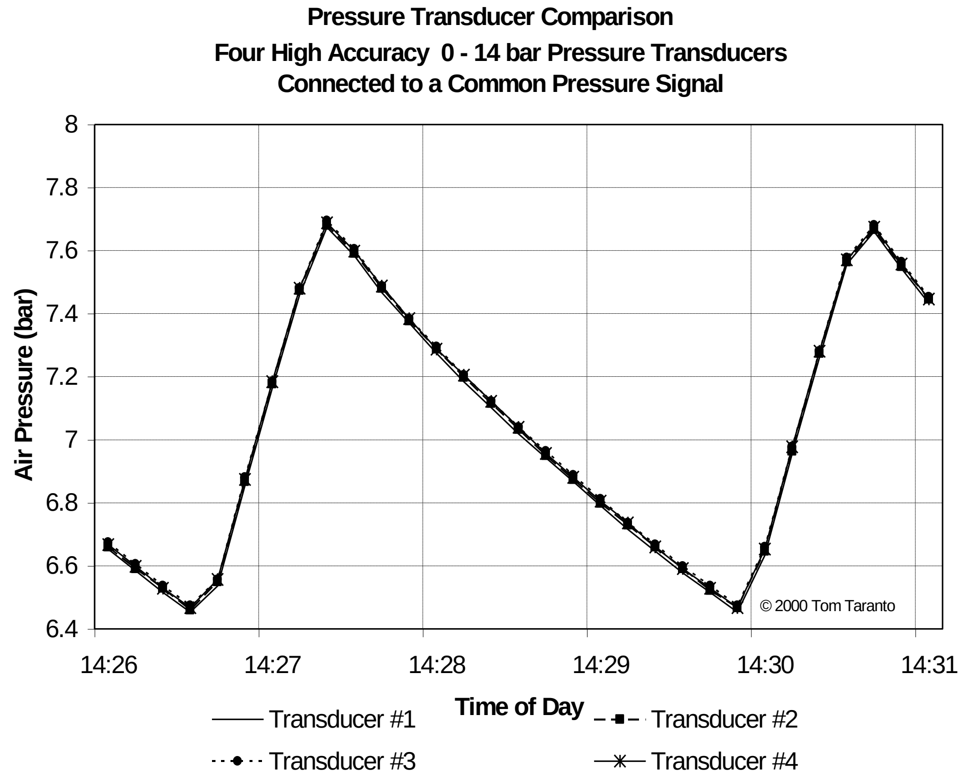



**Figure 11-9: Pressure Transducer Comparison**

If the informational goal of pressure measurement is to compare control signal pressure at two different compressors, measure distribution pressure gradient, or other comparisons of pressure differentials, the more accurate pressure transducers will record more accurate and appropriate data.

## e. Flow Transducers

In compressed air systems, airflow measurement is generally expressed as mass units per period of time, specifically in terms of volumetric flow rate. In Europe, the typical unit of measure is $Nm^3$/min (Normal cubic meters per minute). In the United States, the typical unit of measure is scfm (standard cubic feet per minute). Remember, if the unit of measure begins with "standard" or "normal", the unit is most likely referring to weight. If the unit of measure is just $m^3$/min, liters/sec, acfm or cfm, the unit is most likely a volume measure. For test equipment made in the United States, the calibration will set the air density at standard conditions*, which is 0.07423 lb/$ft^3$, and directly relates standard cubic feet per minute to a mass flow rate in lb/minute. For the DIN standard, the calibration would be set for 1.294 kg/$m^3$.

*** NOTE:** International Standards Organization (ISO) and the Compressed Air and Gas Institute (CAGI), have defined standard air as: 14.5 psia (1 bar); 68°F (20ºC); dry (0% relative humidity). DIN 1343 defines normal (same term as “standard” for CAGI) air as: 1.01325 bar (14.69 psia); 0°C (32°F); dry (0% relative humidity). There are other standards as well. Verifying the standard and unit of measure that a flow meter or other test instrument is reporting is critical to obtaining valid data.

Flow transducers for compressed air generally fall into one or two broad categories: those that measure the gas velocity (e.g. meters per second) in the pipeline, and those that measure the mass velocity in the pipeline.

Thermal dispersion mass flow meters use temperature-sensing elements in the flow stream to measure mass velocity. One temperature sensor measures the temperature of the gas and the second is heated when current passes through the sensor. As gas flows over the heated element, the gas removes heat from the sensor. The rate of heat transfer varies with the fluid density, and also varies as the square root of the velocity at which the fluid flows past the wire. The power necessary to maintain temperature of the heated element is, therefore, related to the mass velocity of gas passing over the sensor.

The sensor signal is calibrated to provide a transducer output signal that changes linearly with mass velocity in the pipeline. Accuracy of the air velocity sensor is typically ± 2% FS. Once mass velocity has been measured, the pipeline’s cross-sectional area is used to determine the volumetric flow rate.

Several factors affect the overall accuracy of the airflow transducer in any specific installation:

- Characteristics of the flow profile in the pipeline.
- Positioning of the sensor inside the pipeline.
- The relative smoothness or roughness of the pipe wall.
- Accuracy of the cross-sectional area used to calculate volumetric flow rate.

Insertion style compressed airflow transducers are designed for easy installation into an existing compressed air pipeline. Every installation in a different pipeline affects the accuracy of the measurement. Typical industrial compressed air pipeline installations will generally result in ±10% FS accuracy of the measured flow. Repeatability approaches the ±2% FS accuracy rating for the sensor. Insertion style meters are better suited to informational objectives that are met with repeatability rather than accuracy.

Informational objectives of compressed air flow measurement are generally satisfied with repeatable measurements. Demand profiles and characteristic signatures of demand events illustrate how airflow changes with time. Identifying supply side control response characteristics by measuring how much the compressor airflow increases or decreases in response to system dynamics is also satisfied with repeatable airflow measurements.

Checking a compressor manufacturer's performance rating, on the other hand, requires accurate flow measurement. Insertion style meters and field measurements do not have the necessary accuracy to check manufacturer's airflow.

### f. Power and Energy Transducers, current, voltage, power and energy

Most industrial compressed air systems utilize air compressors are driven by electric motors supplied with three-phase electrical power. This discussion, therefore, covers measurement of three-phase power and energy to drive an air compressor. Other drives, not considered in this discussion, may include single-phase electric motors, diesel or natural gas engines, natural gas or liquid fuel turbine engines, or steam turbine drives.

Power – Power is typically measured in kilowatts (kW) and is the rate of work at the moment in time when the measurement is made.

Energy – Energy is typically measured in kilowatt-hours (kWH). The amount of energy, when the power being consumed by a load, is calculated in terms of time.

**CAUTION:** Only properly trained personnel should connect the electrical measurement equipment. Electrical power must be de-energized before connections are made. Some measurement equipment designs may allow for connection while the power is energized. Such equipment is typically limited to 600 volts. Measurement of higher voltage systems (greater than 600 volts) requires specialized equipment and training.

Some high voltage motors have permanently installed amperage meters or displays. The leads to these amperage meters may be low voltage connections that will allow installation of secondary current sensors on the leads to the permanently installed amp-meters. If the ratio of both primary and secondary current transformers is known, measurement of the current, with equipment connected to the low voltage leads, may be possible.

Power measurement of air compressors can be done by various methods. Measurement of motor amperage and line voltage allows power to be calculated from the 3-phase power formula as discussed in 10.2 Plant Background Information .

Measuring amperage on one of the three phases supplying the motor is common. However, a more accurate method is to measure current in all three phases and use the average value in the calculation above. Similarly, the average of all line-to-line voltages is preferred to a single voltage measurement. Power factor may be measured, but 0.85 is often assumed for calculation purposes. Measurement of power involves connection of two or three current transformers

(CT's) to the motor leads, along with three leads measuring line-to-line voltages across all three phases. The meter/transducer measures all of the values and computes the power, displays a value, and/or provides an output signal.

### g. Scaling Transducer Signals to Engineering Units

The electrical output from various transducers provide signals that when properly interpreted represent the measurement of physical parameters. These signals may be voltage, amperage, frequency, etc. They are often but not always linear with respect to the physical parameter and the transducer's operating range.

### h. Linear Scaling with Slope and Offset Method

The linear scaling method uses the equation of a straight line, Y = m X + b to scale transducer output signal X to measured parameter Y.

A pressure transducer has an operating range of 0 to 15 bar with an output signal of 4 to 20 ma. The slope is the change in measured parameter (pressure) per unit change in signal (milliamp). The offset is the value of measured parameter (pressure) that results when there is no signal (zero milliamps). What are the slope and offset necessary to properly scale engineering units for this pressure transducer?

**Equation 11-2: Solve for Slope (Scale Factor)**

$$Slope = \frac{Transducer\ Range}{Signal\ Range}$$

$$Slope = \frac{dY}{dX} = \frac{Y_1 - Y_2}{X_1 - X_2} = \frac{0 - 15\ bar}{4 - 20\ ma} = 0.9375\ {}^{bar}\!/_{ma}$$

Solving for the Y intercept given that the transducer's slope is 0.9375 bar/ma and the straight line passes through the point x=4, y=0.

**Equation 11-3: Solve for the Y intercept**

$$
\begin{aligned}
&Y = mX + b \\
&Y - mX = b \\
&b = 0 - 0.9375 * 4 = -3.75
\end{aligned}
$$

Graphically, the offset can be found by multiplying the signal offset times the slope. Signal offset is equal to zero minus the minimum signal range. In this example the minimum signal range is 4 ma. Zero minus 4 means that the signal offset is negative 4. Multiplying the signal offset (-4 ma) times the slope 0.9375 bar / ma, calculates the offset to be –3.75 bar.

**Equation 11-4: Graphically Solve for Offset**

$$
\begin{aligned}
&Offset = (0 - Signal\,Offset) \times Slope \\
&Offset = (0 - 4ma) \times 0.9375bar / ma = -3.75bar
\end{aligned}
$$

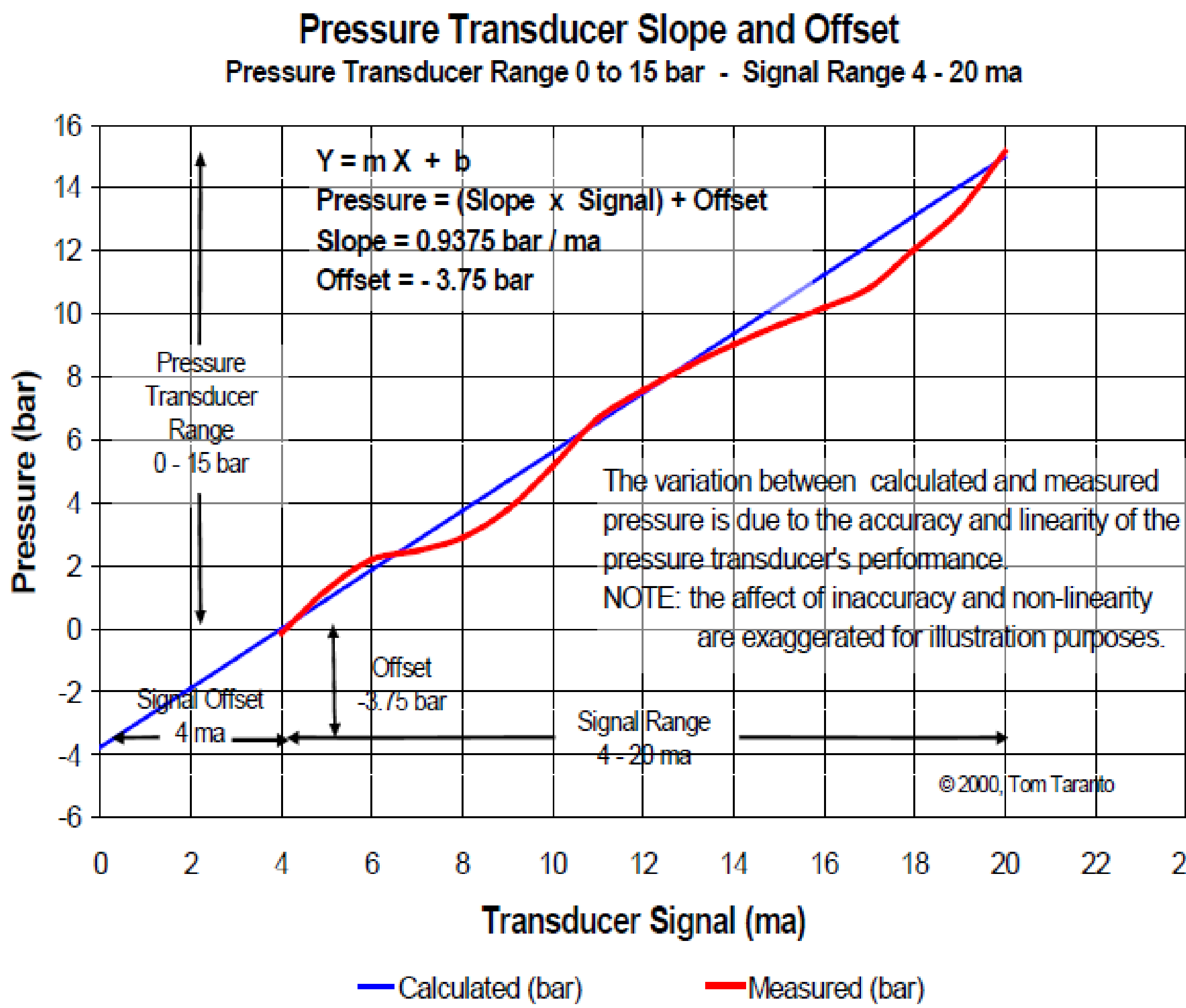



**Figure 11-10: Pressure Transducer Slope and Offset**

The chart in Figure 11-10 shows a plot of slope and offset for the calculated relationship between transducer signal and pressure. The transducer's performance will have some error, non-linearity, and hysteresis that will impact the measurement. The red line in the chart also includes a plot to illustrating what could be the result for a plot of actual measurements made with the pressure transducer.

### i. Linear Scaling with Definition of Two Points

Some data acquisition software may require you to enter the slope and offset value. Other software may provide a method of linear scaling where two reference coordinates (X1,Y1) and (X2,Y2) of transducer performance are defined and the software completes the calculation. In the example above, signal X of 4 ma = pressure Y of 0 bar, therefore, one point is (X1 = 4, Y1 = 0).

Similarly, the second known point is (X2 = 20, Y2 = 15 bar). Entering these two reference points completes the necessary linear scaling.

It is good to know the mathematics of calculating scaling factors so in the event data has been recorded with a scaling error, you can use these calculations to correct the data to the proper engineering values.

### j. Other Scaling Methods

Other scaling methods include square root scaling, interpolation, frequency, and pulse counter inputs. They apply to some particular flow transducers, non-linear transducers, power and energy transducers. Researching and correctly implementing transducer-scaling requirements, as specified by the manufacturer, is very important.

## 11.8 Key Learning Points

1. Measurement system accuracy depends on human factors; connections to the system, transducers; wiring, cables, electrical connections; data acquisition hardware and software; along with measurement techniques.
2. Sample rate, data averaging, and data intervals depend on system characteristics.
3. Use appropriate sensors, transducers, and measurement system accuracy.
4. Transducers output various signals in proportion to the physical parameter being measured.
5. Signals must be properly scaled to correctly record the measurement.

# 12. Maintenance

Like all electro-mechanical equipment, industrial compressed air systems require periodic maintenance to operate at peak efficiency and minimize unscheduled downtime. Inadequate maintenance can have a significant impact on energy consumption via lower compression efficiency, air leakage, or pressure variability. Poor maintenance can also lead to high operating temperatures, poor moisture control, and excessive contamination. Most problems are minor and can be corrected by simple adjustments, cleaning, part replacement, or the elimination of adverse conditions. Compressed air system maintenance is similar to that performed on cars; filters and fluids are replaced, cooling water is inspected, belts are adjusted, and leaks are identified and repaired.

All equipment in the compressed air system should be maintained in accordance with manufacturers' specifications. Manufacturers provide inspection, maintenance, and service schedules that should be followed strictly. In many cases, it makes sense from efficiency and economic standpoints to maintain equipment more frequently than the intervals recommended by manufacturers, which are primarily designed to protect equipment.

One way to tell if a system is being maintained well and is operating properly is to periodically baseline or benchmark the system by tracking power, pressure, flow, and temperature. If power use at a given pressure and flow rate goes up, the system's efficiency is degrading. This baselining will also help determine if the compressor is operating at full capacity, and if the capacity is decreasing over time. On new systems, specifications should be recorded when the system is first set-up and operating properly.

Proper maintenance is essential to compressed air system efficiency and reliability. The key to success requires compressor operators to determine the requirements for each piece of equipment, the necessary resources, and to schedule the maintenance based on the manufacturer's manuals and trend analysis of recorded data. All observations and meter readings should be recorded for compressors, dryers, filters, and any components in the compressor plant. The combination of equipment control panel data, frequent inspections, and log sheets are required to avoid unscheduled system shutdowns, and to utilize the principles of preventive and predictive maintenance. Record the dates of all maintenance and repairs, including a list of all parts that were replaced or services performed.

The maintenance schedules provided in this handout are intended to be used only as a guide. For more exact procedures, always refer to the manufacturer's manuals.

## 12.1 Stopping for Maintenance

The following procedures should be followed when stopping the compressor for maintenance or service:

Step 1
Disconnect and lock out the main power source. Display a sign in clear view at the main power switch stating that the compressor is being serviced.
WARNING! Never assume a compressor is safe to work on just because the compressor is not operating. The controls could signal the compressor to restart at any time.

Step 2
Isolate the compressor from the compressed air supply by closing a manual shutoff valve downstream (and upstream, if applicable in booster service) from the compressor. Display a sign in clear view at the shutoff valve stating that the compressor is being serviced. Be certain that a pressure relief valve is installed upstream of any isolation valve.

Step 3
Open and lock a pressure relief valve within the pressurized system to allow the system to be completely de-pressurized. NEVER remove a plug to relieve the pressure!

Step 4
Shut off the water-cooling supply (water cooled compressors).

Step 5
Open all manual drain valves within the area to be serviced.

Step 6
Wait for the unit to cool before starting to service. (Temperatures of 50°C can burn skin. Some surface temperatures exceed 175°C when the compressor is operating, and just after it is shut down).

Step 7
Refer and give preference to the manufacturer's manuals over these typical maintenance procedures.

## 12.2 Maintenance Schedules

To assure maximum performance and service life of the compressor, a routine maintenance schedule should be developed. Sample schedules have been included here to help in developing a maintenance schedule designed for a particular application. Time frames may need to be shortened in harsher environments.

The documentation shipped with your compressor should contain a Maintenance Schedule Checklist. Make copies of this checklist and retain the master to make more copies as needed. On each copy of the checklist, enter dates and initials in the appropriate spaces. Keep the checklist and this Handout readily available near the compressor.

## 12.3 General Maintenance Discussion

Maintenance issues for specific system components are discussed below.

### a. Compressor Package

The main areas of the compressor package in need of maintenance are the compressor, heat exchanger surfaces, air lubricant separator, lubricant, lubricant filter, and air inlet filter.

The compressor and intercooling surfaces need to be kept clean and foul-free. If they are dirty, compressor efficiency will be adversely affected. Fans and water pumps should also be inspected to ensure that they are operating at peak performance.

The air/lubricant separator in a lubricant-cooled rotary screw compressor generally starts with a 0.14 to 0.2 bar pressure drop at full-load when new. Maintenance manuals usually suggest changing them when there is about a 0.7 bar pressure drop across the separator. In many cases, changing the separator element earlier may make sense, especially if electricity prices are high.

The compressor lubricant and lubricant filter need to be changed per manufacturer's specification. Lubricant can become corrosive and degrade both the equipment and system efficiency.

For lubricant-injected rotary compressors, the lubricant serves to lubricate bearings, gears, and intermeshing rotor surfaces. The lubricant also acts as a seal and removes most of the heat of compression. Only a lubricant meeting the manufacturer’s specifications should be used.

Inlet filters and inlet piping also need to be kept clean. A dirty filter can reduce compressor capacity and efficiency. Filters should be maintained at least per manufacturer's specifications, taking into account the level of contaminants in the facility's air.

### b. Compressor Drives

If the electric motor driving a compressor is not properly maintained, not only will more energy be consumed, but the motor may fail before its expected lifetime. The two most important aspects of motor maintenance are lubrication and cleaning.

Lubrication. Too much lubrication can be just as harmful as too little and is a major cause of premature motor failure. Motors should be lubricated per the manufacturer's specification, which can be anywhere from every 2 months to every 18 months, depending on annual hours of operation and motor speed. On motors with bearing grease fittings, the first step in lubrication is to clean the grease fitting and remove the drain plug. High quality new grease should be added, and the motor should be run for about an hour before the drain plug is replaced. This allows excess grease to be purged from the motor without dripping on the windings and damaging them.

Cleaning. Since motors need to dissipate heat, keeping all of the air passages clean and free of obstruction is important. For enclosed motors, keeping the cooling fins free of debris is vital to the life of the motor. Poor motor cooling can increase motor temperature and winding resistance, which shortens motor life and increases energy consumption.

Belts. Motor v-belt drives also require periodic maintenance. Tight belts can lead to excessive bearing wear, and loose belts can slip and waste energy. Under normal operation, belts stretch and wear and, therefore, require adjustment. A good rule-of-thumb is to examine and adjust belts after every 400 hours of operation.

### c. Air Treatment Equipment

Fouled compressed air treatment equipment can result in excessive energy consumption as well as poor-quality air that can damage other equipment. All filters should be kept clean. Dryers, aftercoolers, and separators should all be cleaned and maintained per manufacturer's specifications.

Automatic Drain Traps. Most compressed air systems have numerous moisture traps located throughout the system. Traps need to be inspected periodically to ensure that they are not stuck in either the open or closed position. An automatic drain trap stuck in the open position will leak compressed air; a drain trap stuck in the closed position will cause condensate to back up and be carried downstream where other system components can be damaged. Traps stuck in the open position can be a major source of wasted energy in some plants.

End-Use Filters, Regulators, and Lubricators. Point-of-use filters, regulators, and lubricators are needed to ensure that a tool is receiving a clean, lubricated supply of air at the proper pressure. Filters should be inspected periodically because a clogged filter will increase pressure drop, which can either reduce pressure at the point of use or increase the pressure required from the compressor, thereby consuming excessive energy. A filter that is not operating properly will also allow contaminants into a tool, causing it to wear out prematurely. The lubricant level should also be checked often enough to ensure that reservoir does not run dry. Tools that are not properly lubricated will wear prematurely and use excess energy.

### d. Leaks

Leak detection and repair is an important part of any maintenance program.

## 12.4 Maintenance Schedules

Establishing a regular, well-organized maintenance program and strictly following the program is critical to maintaining the performance of a compressed air system. One person should be given the responsibility of ensuring that all maintenance is performed properly, on schedule, and adequately documented.

The following are typical recommended minimum maintenance procedures for air-cooled reciprocating compressors, water-cooler double-acting reciprocating compressors, lubricant-injected rotary compressors, and lubricant-free rotary compressors.

## a. Routine Maintenance for Air-Cooled Reciprocating Compressors

Every 8 Hours (or Daily)

- Maintain lubricant level between high and low level marks on bayonet gauge. (Discoloration or a higher lubricant level reading may indicate the presence of condensed liquids). If lubricant is contaminated, drain and replace.
- Drain receiver tank, drop legs, and traps in the distribution system.
- Give compressor an overall visual inspection and be sure safety guards are in place.
- Check for any unusual noise or vibration.
- Check lubricant pressure on pressure lubricated units. Maintain 125 to 140 kPa when compressor is at operating pressure and temperature. High pressure rated compressors should maintain 150 to 175 kPa of lubricant pressure.
- Check for lubricant leaks.

Every 40 Hours (or Weekly)

- Be certain pressure relief valves are working.
- Clean the cooling surfaces of the intercooler and compressor.
- Check the compressor for air leaks.
- Check the compressed air distribution system for leaks.
- Inspect lubricant for contamination & change if necessary.
- Clean or replace the air intake filter. Check more often under humid or dusty conditions.

Every 160 Hours (or Monthly)

- Check belt tension.

Every 500 Hours (or Every 3 Months)

- Change lubricant (more frequently in harsher environments).
- Check lubricant filter on pressure lubricated units (more frequently in harsher environments).
- Torque pulley clamp screws or jam-nut.

Every 1000 Hours (or Every 6 Months)

- When synthetic lubricant is used, lubricant change intervals may be extended to every 1000 hours or every 6 months, whichever occurs first (change more frequently in harsher conditions).

- Inspect compressor valves for leakage and/or carbon build-up. The lubricant sump strainer screen inside the crankcase of pressure-lubricated models should be thoroughly cleaned with a safety solvent during every lubricant change. If excessive sludge build-up exists inside the crankcase, clean the inside of the crankcase as well as the screen. Never use a flammable or toxic solvent for cleaning. Always use a safety solvent and follow the directions provided.

Every 2000 Hours (or Every 12 Months)

- Inspect the pressure switch diaphragm and contacts. Inspect the contact points in the motor starter.

Lubrication

Compressors may be shipped without lubricant in the crankcase. Before starting the compressor, add enough lubricant to the crankcase to register between the high and low marks on the dipstick or on bull's eye sight gauge. Use the specified lubricant or consult the manufacturer for recommendations. Certain synthetic lubricants have proven under extensive testing to minimize friction and wear, limit lubricant carryover, and reduce carbon and varnish deposits. They will support the performance characteristics and life and are highly recommended. Refer to the manufacturer's specifications to determine the correct amount of lubricant and viscosity to use for your model and application. Use the supplier’s lubricant analysis program.

### b. Routine Maintenance for Water-Cooled, Double-Acting Reciprocating Compressors

The following are typical minimum maintenance requirements for this type of compressor.

Daily or every 8 hours*

- Check compressor lubricant level in crankcase and cylinder lubricator and, if necessary, add to level indicated by sight gauge.
- Check cylinder lubrication feed rate and adjust, as necessary.
- Check lubricant pressure and adjust as necessary to meet specified operating pressure.
- Check cylinder jacket cooling water temperatures.
- Check capacity control operation. Observe discharge pressure gauge for proper LOAD and UNLOAD pressures.
- Drain control line strainer.
- Check operation of automatic condensate drain trap (intercooler and aftercooler).
- Drain condensate from discharge piping as applicable (drop-leg and receiver).
- Check intercooler pressure on multi-stage machines, and refer to manufacturer's manual if pressure is not as specified.

Monthly or every 360 hours*

- Check piston rod packing for leaks and for blow-by at gland. Repair or replace as necessary per manufacturer's manual.
- Inspect lubricant scraper rings for leakage. Replace as necessary per manufacturer's manual.
- Inspect air intake filter. Clean or replace as necessary.
- Drain lubricant strainer/filter sediment.
- Lubricate un-loader mechanism per manufacturer's manual.
- Check motor amps at compressor full capacity and pressure.

Semi-annually or every 3000 hours*

- Perform valve inspection per manufacturer's manual.
- Inspect cylinder or cylinder liner, through valve port, for scoring.
- Change crankcase lubricant, if required.
- Clean crankcase breather (if provided).
- Change lubricant filter element.
- Remove and clean control air filter/strainer element.
- Check all safety devices for proper operation.
- Perform piston ring inspection on non-lubricated design. Replace as necessary per manufacturer's manual.

Annually or every 6000 hours*

- Remove and clean crankcase lubricant strainer.
- Check foundation bolts for tightness. Adjust as necessary.
- Perform piston ring inspection. Replace as necessary per manufacturer’s manual.
- Experience gained from a well kept maintenance log may allow the recommended times to be adjusted.

### c. Routine Maintenance for Lubricant Injected Type Rotary Compressor

The following are typical minimum maintenance requirements.

Periodically/Daily-8 hours maximum

- Monitor all gauges and indicators for normal operation.
- Check lubricant level.
- Check for lubricant leaks.
- Check for unusual noise or vibration.
- Drain water from air/lubricant reservoir.
- Drain control line filter.

Weekly

- Check safety valve operation.

Monthly

- Service air filter as needed. (daily or weekly if extremely dusty conditions exist).
- Wipe entire unit down, to maintain appearance.
- Check drive motor amps at compressor full capacity and design pressure.
- Check operation of all controls.
- Check operation of lubricant scavenger/ return system. Clean, as necessary.

6 Months or every 1000 hours

- Take lubricant sample.
- Change lubricant filter*

Periodically/yearly

- Go over unit and check all bolts for tightness.
- Change air/lubricant separator.
- Change air filter.
- Lubricate motors per manufacturer's instructions.  Will probably be more often than annually.
- Check safety shutdown system. Contact authorized serviceman.

***Manufacturers may recommend changing the lubricant filter within the first week of operation, to rid the system of foreign matter, which may have collected during initial assembly and startup.***

### d. Routine Maintenance for Lubricant Free Rotary Screw Compressor

Routine maintenance is relatively minimal. The microprocessor control panel monitors the status of the air and oil filters. When maintenance to either device is required, the control panel may display the appropriate maintenance message, and flash the location on the display as a visual reminder.

DO NOT remove caps, plugs, and/or other components when compressor is running or pressurized. Stop compressor and relieve all internal pressure before doing so.

Daily

Following a routine start, observe the various control panel displays and local gauges to check that normal readings are being displayed - previous records are very helpful in determining the normalcy of the measurements. These observations should be made during all expected modes of operation (i.e. full load, no-load, different line pressures, cooling water temperatures, etc.).

After Initial 50 Hours of Operation

Upon completion of the first 50 hours of operation, a few maintenance requirements are needed to rid the system of any foreign materials, which may have accumulated during assembly:

- Change the lubricant filter element.
- Clean the control line filter element.
- Check/replace the sump breather filter element.

Every 3000 Hours of Operation

The following items should be checked every 3000 hours of operation, although service conditions such as relative cleanliness of process air or quality of cooling water may require shorter inspection intervals.

- Check/change oil charge and filter element.
- Check/change air filter element.
- Check/change sump breather filter element.
- Check/clean control line filter element.
- Check/clean condensate drain valve.
- Check condition of shaft coupling element and tightness of fasteners.
- Measure and record vibration signatures on compressor, gearbox and motor (optional).

*NOTE: Please refer to the motor manufacturer's documentation for recommended maintenance. Keep in mind that the specified type and quantity of lubricating grease for motor bearings is crucial.*

Every 15,000 Hours of Operation

In addition to those items covered by the 3000-hour maintenance interval, the following items must also be checked every 15,000 hours of operation, depending upon conditions of service:

- Operate/test all safety devices.
- Check/clean heat exchangers.
- Check/clean blowdown valve.
- Check operation of balancing switch/valve assembly.
- Check/clean water regulating valve.
- Check/clean check valve.
- Check/clean galvanized interstage pipe work.
- Check condition of isolation mounts under compressor unit and motor.
- Check/clean strainer and check valve included in oil pump suction line, inside oil sump.
- Check compressor unit internal clearances.

Please be aware that work on the compressor stages and gearbox must be conducted by manufacturer's personnel only. Any work done by unauthorized personnel can render the manufacturer's equipment warranty null and void.

Parts Replacement and Adjustment Procedures: Familiarize yourself with the safety guidelines offered in the Safety Section of the manufacturer's manual before attempting any maintenance on the package.

### e. Routine Maintenance for Centrifugal Air Compressors

The following are typical maintenance requirements for this type of compressor.

Daily

- Record operating air inlet, interstage and discharge pressures and temperatures.
- Record cooling water inlet and outlet pressures and temperatures.
- Record lubricant pressure and temperatures.
- Record all vibration levels.
- Check air inlet filter differential pressure.
- Check proper operation of drain traps.
- Drain control air filter.
- Check for leaks, air, water and lubricant. Repair and clean as necessary.
- Check lubricant sump level and adjust as necessary.
- Check drive motor for smooth operation and record amperes.

Every 3 months

- Check lubricant filter differential pressure. Replace element as necessary.
- Check lubricant sump venting system. Replace filter elements as necessary.
- Check operation of capacity control system.
- Check operation of surge control system.
- Check main drive motor amperes at full load operation.
- Check automatic drain traps and strainers. Clean and/or replace as necessary.

Every 6 months

- Check air inlet filter and replace element as necessary.
- Take oil sample for analysis. Replace lubricant as necessary.

Annually

- Inspect intercooler, aftercooler, and lubricant cooler. Clean and/or replace as necessary.

- Inspect main drive motor for loose mounting bolts, frayed or worn electrical cables, accumulated dirt. Follow manufacturer's recommendations, including lubrication.
- Inspect main drive coupling for alignment and required lubrication.
- Inspect gearbox for loose mounting bolts, vibration, unusual noise or wear and axial clearances per manufacturer's manual.
- Check impeller inlets and diffusers for signs of wear, rubbing or cracking.
- Check control panel for complete and proper operation.
- Check all control valves for proper operation.
- Check all safety devices for proper settings and operation.
- Inspect check valve; replace worn parts.
- Keep all components/accessories clean and follow all recommended safety procedures.

# 13. Heat Recovery


As much as 80-93% of the electrical energy used by an industrial air compressor is converted into heat. In many cases, a properly designed heat recovery unit can recover anywhere from 50-90% of this available thermal energy and put it to useful work heating air or water.

Typical uses for recovered heat include supplemental space heating, industrial process heating, water heating, makeup air heating, and boiler makeup water preheating. Recoverable heat from a compressed air system is not, however, normally hot enough to be used to produce steam directly. Heat recovery systems are available for both air- and water-cooled compressors.

## 13.1 Heat Recovery with Air-Cooled Rotary Screw Compressors

### a. Heating Air

Air-cooled packaged rotary screw compressors are very amenable to heat recovery for space heating or other hot air uses. Ambient atmospheric air is heated by passing it across the system's aftercooler and lubricant cooler, where it extracts heat from both the compressed air and the lubricant that is used to lubricate and cool the compressor.

Since packaged compressors are typically enclosed in cabinets and already include heat exchangers and fans, the only system modifications needed are the addition of ducting and another fan to handle the duct loading and to eliminate any back pressure on the compressor cooling fan. These heat recovery systems can be modulated with a simple thermostatically-controlled hinged vent. When heating is not required -- such as in the summer months -- the hot air can be ducted outside the building. The vent can also be thermostatically regulated to provide a constant temperature for a heated area.

Hot air can be used for space heating, industrial drying, preheating aspirated air for oil burners, or any other application requiring warm air. As a rule of thumb, approximately 5.3 kW of energy is available for each $m^3$/min of capacity (at full-load). Air temperatures of 17-22°C above the cooling air inlet temperature can be obtained. Recovery efficiencies of 80-90% are common.

Caution should be applied because if the supply air for the compressor is not from outside, and the recovered heat is used in another space, you can decrease the static pressure in the cabinet and reduce the efficiency of the compressor. If outside air is used, some return air may be required to avoid damaging the compressor with below freezing air.

### b. Heating Water

Using a heat exchanger, it is also possible to extract waste heat from the lubricant coolers found in packaged water-cooled reciprocating or rotary screw compressors and produce hot water. Depending on design, heat exchangers can produce non-potable (gray) or potable water. When hot water is not required, the lubricant is routed to the standard lubricant cooler.

Hot water can be used 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 hot air and hot water, and allow the operator some ability to vary the hot air/hot water ratio.

## 13.2 Heat Recovery with Water-Cooled Compressors

Heat recovery for space heating is not as common with water-cooled compressors because an extra stage of heat exchange is required and the temperature of the available heat is lower. Since many water-cooled compressors are quite large, however, heat recovery for space heating can be an attractive opportunity. Recovery efficiencies of 50-60% are typical.

## 13.3 Calculating Energy Savings

When calculating energy savings and payback periods for heat recovery units, it is important to compare heat recovery with the current source of energy for generating thermal energy, which may be a low-price fossil fuel such as natural gas. The equations in the text box below illustrate the annual energy and costs savings available by recovering heat for space heating from an air-cooled rotary screw compressor. Applications where the existing heater is less than 85% efficient will see proportionally higher savings.

**Energy Savings Calculations**

**Energy Savings (kWh/year)** = 0.8 x Compressor kW x hours of operation

**Example:** A 75kW compressors running two shifts, 5 days per week

= (0.80) x 75kW x 4160 hours per year

***= 249,600 kWh per year***

*Where 0.80 is the recoverable heat as a percentage of the unit's output*

$$\text{Cost savings}(\$/\text{y}) = \frac{\text{Energy savings in kWh/y} \times \text{kWh/unit of fuel} \times \$/\text{unit of fuel}}{\text{Primary heater efficiency}}$$

**Example:** Waste heat will be displacing heat produced by a natural gas forced air system with an efficiency of 85%. Assume the cost for natural gas is \$0.14/m$^3$, and the energy content of natural gas is 37MJ per m$^3$.

$$\text{Cost savings} = \frac{249{,}600\ \text{kWh/y} \times \frac{\text{m}^3}{37\text{MJ}} \times \frac{3.6\text{MJ}}{1\text{kWh}} \times \frac{\$0.14}{\text{m}^3}}{85\%}$$

$$\text{Cost savings} = \$4000\ \text{per year}$$

** Cost of operating an additional fan for duct loading has not been included*

# 14. Egypt Case Studies

The Egyptian industrial sector is responsible for approximately 43% of national final energy consumption and 33% of national electricity consumption (IEA, 2013). Overall industry-related emissions accounted for 29% of the total emissions in 2005 and are expected to increase their relative share to 36% by 2030 (McKinsey 2010). The final energy consumption per unit of output in the most important industries in Egypt is typically 10 to 50% higher than the international average. Therefore, increased energy efficiency in the Egyptian industry has the potential to make a significant contribution to meet the growing energy supply challenges facing the country.

The Industrial Energy Efficiency Project (IEEP) in Egypt started in January 2013 with a fund from the Global Environmental Facility (GEF). This project was implemented by the United Nations Industrial Development Organization (UNIDO), with the Egyptian Environmental Affairs Agency (EEAA) as the lead executing partner and in full cooperation with the Industrial Development Authority (IDA), the Egyptian Organization for Standardization (EOS), the Industrial Modernization Center (IMC) and the Federation of Egyptian Industries (FEI).

The IEEP seeks to address some of the key barriers to industrial energy efficiency (IEE), to deliver measurable results and to make an impact on how Egyptian industries manage energy through an integrated approach that combines capacity building and technical assistance interventions at the policy, institutional and enterprise level. Primary target groups of the project are the Egyptian industries, with more of a focus on industrial decision-makers (managers), engineers, vendors and other professionals as well as IEE policy-making and/or implementing institutions.

One component of the IEEP is compressed air system optimization, which addresses technical training and capacity building to candidates from the industrial sector, governmental sector, academia and consultants. The objective of that component is to allow the candidates to study, investigate, and optimize the compressed air system for their assigned industrial enterprises, so as to propose feasible options to the enterprises' top management for reducing the system's energy consumption, and improving the compressed air system performance.

## 14.1 El Araby Group: Quessna Industrial Complex

### a. Company Background

In 1964, a company was established as a small trading store in El Moski, Cairo. In 1982 this trading agent transformed to an industrial group of companies with a total investment volume of more than 1'600 MEGP (2016). The group produces home appliances and air conditioning units. More than 330 different products (in more than 4000 models) are included in the product range; e.g. refrigerators, air conditioners, washing machines, electric water heaters, gas water heaters, televisions, LED screens, florescent light lamps, fans of various types, ventilators, vacuum cleaners, food blinders, irons and other home small appliances. The El Araby Group employs 10'000 workers.

All products and more than 90% of the components are produced in 17 manufacturing plants spread over two main industrial complexes in Banha city and Quessna City. The Group has six trading and manufacturing companies that include the manufacturing plants and the shared utilities; as well as two non-profit organizations.

The El Araby group is considered to be one of the industry leaders in developing Energy Management Systems (EMS) with the assistance from the IEEP; by mid-2016, six companies of the group were ISO 50001 certified. As the CASO serves well in developing saving opportunities for the group, the UNIDO consultants, within agreement of the company and the IEEP, developed this case study on one of the compressed air systems of the Quessna complex.

## b. Compressed Air System Overview

The focus of this compressed air systems optimisation assessment was on the foam-factory sub-system. The foam factory has five compressors; all Ingersoll Rand R160i's rated at 160kW and 29m$^3$/min each.

**Table 14-1: Existing compressor nameplate data in the foam plant**

| Station | Compressor type | P bar | Nominal Q m$^3$/min | Nom. Power kW | Total Consumption kWh/year (2017) | Total Cost EGP (2017) |
|---|---|---|---|---|---|---|
| Foam Plant | IR R160i | 8.5 | 29 | 160 | 532,243 | 408,230 |
| | IR R160i | 8.5 | 29 | 160 | 1,234,732 | 947,039 |
| | IR R160i | 8.5 | 29 | 160 | 1,035,361 | 794,122 |
| | IR R160i | 8.5 | 29 | 160 | 873,992 | 670,352 |
| | IR R160i | 8.5 | 29 | 160 | 922,868 | 707,840 |

The generating capacity of the foam factory was 145m$^3$/min, which represents 50 % of the total compressed air consumption of all factories in the Quessna Complex.

The selected compressor station feeds the foam plant that includes 16 foam machines and other users as well. Figure 14-1 shows a single line diagram of the compressed air system inside the foam plant.

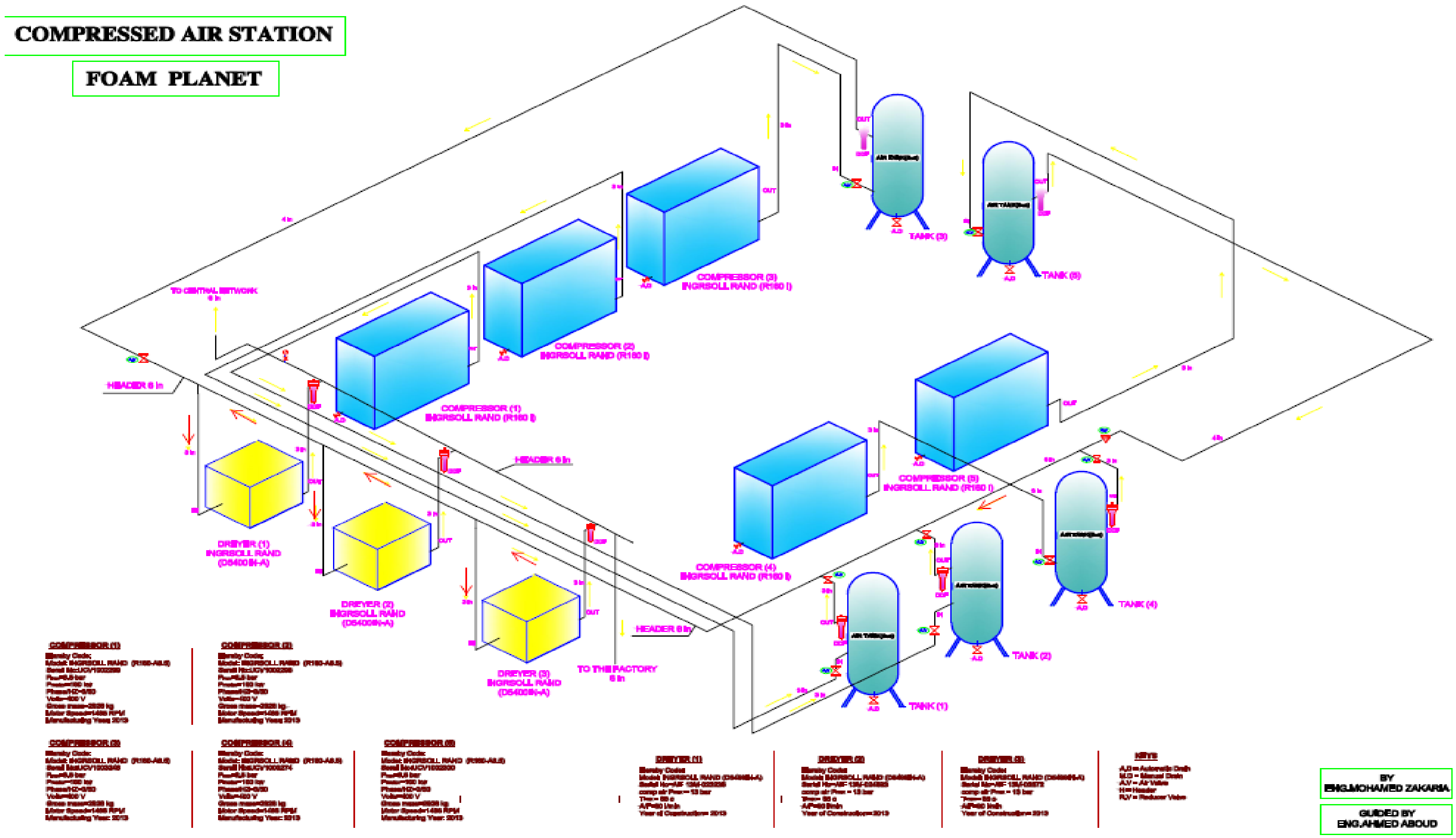



**Figure 14-1: Compressed air system - Foam plant**

Compressed air exits each compressor via a 3” pipe leading to a wet air receiver of 6 $m^3$ capacity and air exits each receiver via a 3” pipe leading to a 6” header, where 3” lines are branched off to three water cooled refrigerated air driers. The dried air is distributed through a 6” header leaving the compressor room to users. Automatic drains are installed at the bottom of each air receiver tank.

The main header of the compressed air system is connected to the central compressed air network via a shut off valve; which was always open. During the study it was decided to permanently close it; except during emergencies or shutting down one of the stations. Figure 14-2 illustrates the general arrangement in the foam factory’s compressor station.

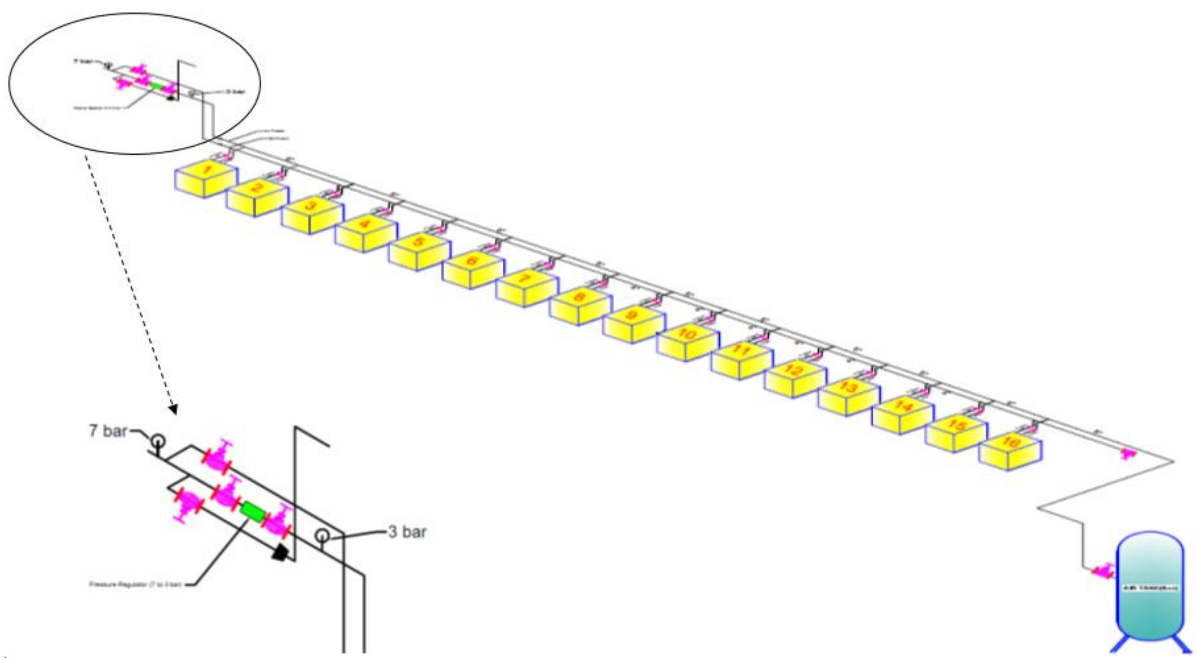



**Figure 14-2: End user single line diagram at the foam plant**

The foam compressed air station is served with three refrigerated air dryers with the following rated capacities:

**Table 14-2: Foam plant dryer nameplate information**

| | Dryer 1 | Dryer 2 | Dryer 3 |
|---|---|---|---|
| Serial No | AIF 13M-024593 | AIF 13M-025721 | AIF 13M-025721 |
| Year | 2013 | | |
| Model | D5400IN-A | | |
| Motor Power | 7 kW | | |
| $P_{max}$ | 13 bar | | |

| | |
|---|---|
| $T_{max}$ | 65 °C |
| Q | 90 $m^3$/min |

When the second measurement period took place in November 2017, the company had started to separate the three compressors from the central compressed air network, and dedicated them only for supplying the low and high pressure demands within the foam factory.

## c. Baseline

Although the El Araby group pays around EGP 7 million per annum for compressed air usage, the annual electricity consumption of the foam factory was 4,599,187 kWh. The Group currently pays for compressed air with an average electricity tariff of 0.767 EGP/kWh.

The most significant consumer of compressed air among all facilities of the Quessna is the foam plant; consuming 59 $m^3$/min out of the industrial complex's 272 $m^3$/min; i.e. 21.6%. The energy consumption required for each cubic meter of air is 0.147 kWh/$m^3$. The cost of each $m^3$ of air is 0.133 EGP/$m^3$.

**Table 14-3: Foam plant's compressors with control method and pressure setpoints indicated**

| Compressor Manufacturer | Model | Capacity Control Method | Power (shaft/ package) kW | Rated Capacity $m^3$/min | Operating Pressure(Load-Unload) |
|---|---|---|---|---|---|
| **Compressor 1 Ingersoll Rand** | R160i-8.5 2013 | Load/Unload | 160 kW nom. shaft 180 kW total pkg. | 29 | (6.3-7.0) |
| **Compressor 2 Ingersoll Rand** | R160i-8.5 2013 | Load/Unload | 160 kW nom. shaft 180 kW total pkg. | 29 | (6.7-7.4) |
| **Compressor 3 Ingersoll Rand** | R160i-8.5 2014 | Load/Unload | 160 kW nom. shaft 180 kW total pkg. | 29 | (6.9-7.6) |
| **Compressor 4 Ingersoll Rand** | R160i-8.5 2013 | Load/Unload | 160 kW nom. shaft 180 kW total pkg. | 29 | (6.4-7.1) |
| **Compressor 5 Ingersoll Rand** | R160i-8.5 2014 | Load/Unload (Standby) | 160 kW nom. shaft 180 kW total pkg. | 29 | (6.2-7.0) |

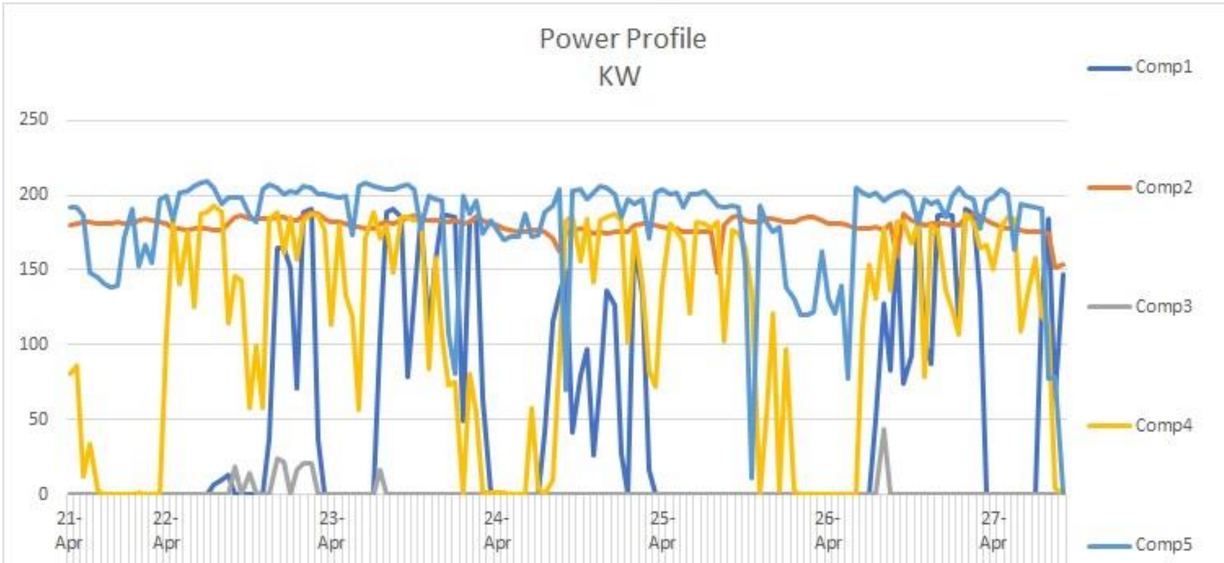



**Figure 14-3: Power profile of operating compressors during 10 day assessment period**

The average power use during the assessment was 520.8 kW for the sum of the five compressors. Based on this number, when using 8760 operating hours per year, the annual energy consumption is calculated as:

$$\text{Energy}\left(\frac{\text{kWh}}{\text{year}}\right) = \text{Power (kW)}.\text{hours of operation} = 520.8 * 8760 = 4{,}562{,}225\,\frac{\text{kWh}}{\text{year}}$$

This calculated value is very close to that measured at the end of the year 2017 (4,599,187 kWh). In terms of monetary values, the energy cost of the compressed air is equivalent to 3.5 million EGP/ year (0.767 EGP/kWh).

## d. Key Findings during the Assessment

**Leakages**

El-Araby Company started addressing the air leakage in the plants by doing a low load test during a non-production time. The Company started this test after ensuring the accuracy of the existing flow meter that is already installed in the system. According to the test result, the total amount of leakage calculated for all the companies within the El-Araby complex at Quessna was around 47.5 $m^3$/hr. The share of leakage within the foam factory according to test, was 5.25 $m^3$/min.

According to the measurement that has been done in the foam factory during the compressed air systems optimisation assessment, it was found that the average leakage was around 6.5 $m^3$/ min; 23% higher than that measured by the company. This leakage leads to an annual leakage value of around 3,416,400 $m^3$ which is equivalent to 478,296 kWh per year at an annual cost of EGP 366,853.

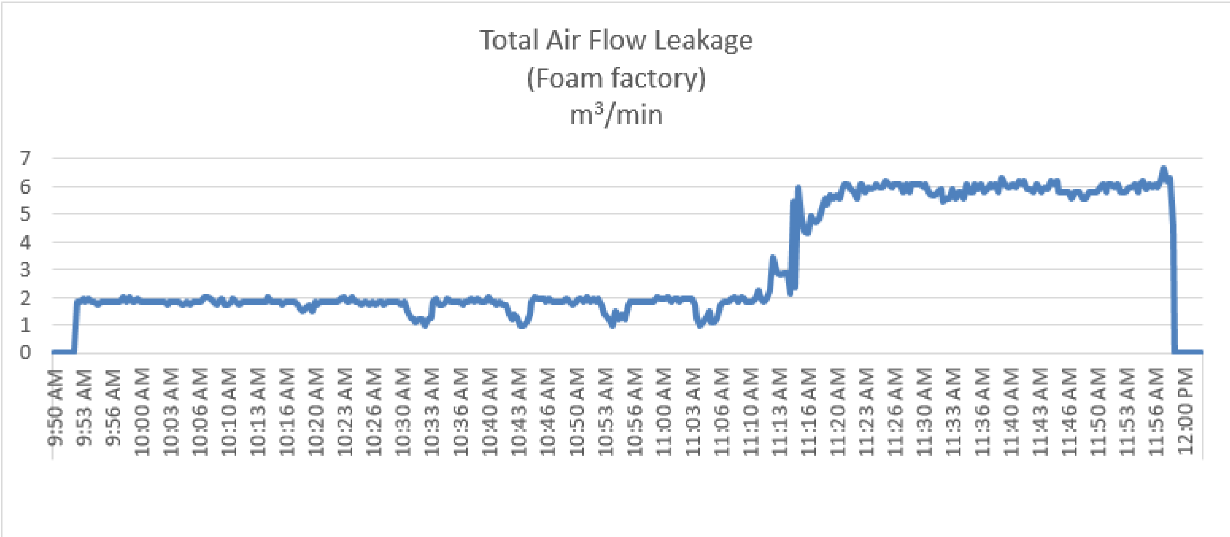



**Figure 14-4: Measured total foam factory air leakage profile**

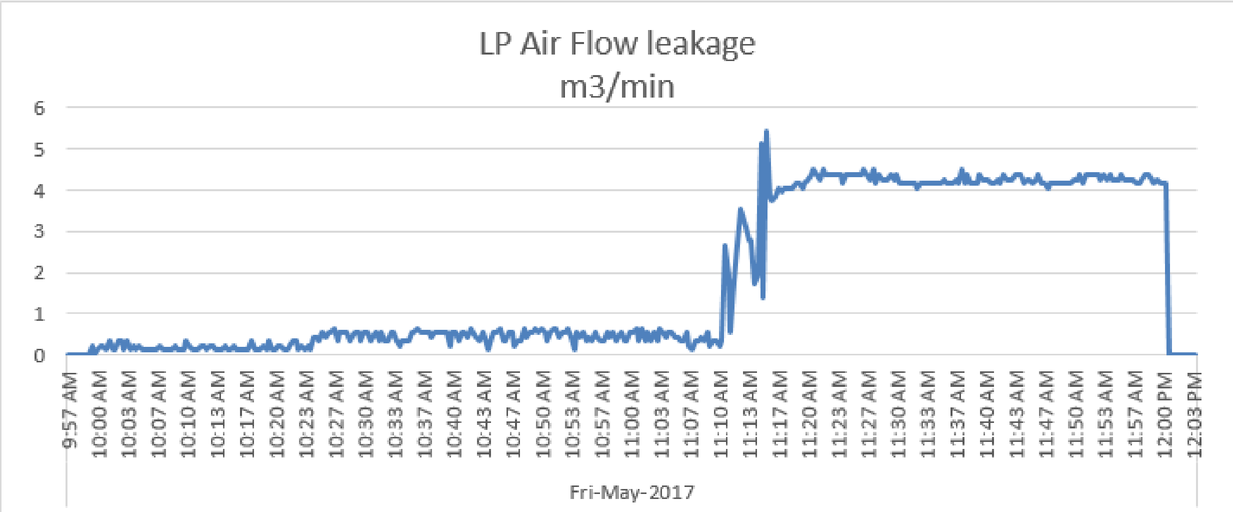



**Figure 14-5: Measured Low Pressure (LP) foam factory air leakage profile**

During the assessment, the team used the ultrasonic leak detector tool to identify and quantify the leakage losses on the individual foam machine hoses:

**Table 14-4: Leakage losses measured per foam machine**

| Machine | M1 | M2 | M3 | M4 | M5 | M6 | M7 |
|---|---|---|---|---|---|---|---|
| $m^3$/min | 0.597642 | 0.82962 | 0.510987 | 0.390308 | 0.39158 | 0.66342 | 0.629107 |

| Machine | M8 | M9 | M10 | M11 | M12 | M13 | M14 |
|---|---|---|---|---|---|---|---|
| $m^3$/min | 0.671116 | 0.72236 | 0.220543 | 0.74801 | 0.578532 | 0.462339 | 0.434713 |

**Table 14-5: Total leakage losses measured on all foam machines**

| Total machine leaks $m^3$/min | Ave | Max | Min |
|---|---|---|---|
| | 0.56 | 0.83 | 0.22 |
| Total machine leaks $m^3$/min | 7.85 | | |

**Figure 14-6: Ultrasonic leak detector measurements**

**Frequent Load/Unload Cycling**

During the assessment, measurements showed that the compressors unloads to a power level varying from 84 to 110 kW, which is more than the total package input power that is documented on the technical data sheet of 60 kW. The compressor does therefor not reach its fully unloaded level, and the separator is not fully relieved before the next load cycle occurs.

**Compressor Controls**

The foam plant operates three out of the five compressors to supply the compressed air demand to the foam factory. Two of them are specified to supply the low pressure demand (Comp1, Comp2) and the third one is responsible for supplying the high pressure demand (Comp4).

**Table 14-6: Measured average, max and min flow during high and low demand periods**

| | ***Low Pressure Demand Air Flow at Foam Factory*** | ***High Pressure Demand Air Flow at Foam Factory*** |
|---|---|---|
| Average Air Flow | 28.74 | 13.39 |
| Maximum Air Flow | 44.44 | 15.07 |
| Minimum Air Flow | 10.77 | 9.5 |

For three days during the assessment's second measurement period in November 2017, the foam factory was working around 65% of its full capacity. As shown in Figure 14-7, Comp1 (red) was operating loaded for around 90% of the time, while Comp2 was turned off. Comp1 was able to cover the 65% demand capacity which is less than the average demand for the foam factory of 28 $m^3$/ min that has been measured in June. Thus Comp1 was usually always operating at close to full load capacity while the demand varies according to production.

During the November 2017 measurement for the high pressure demand of the foam factory, which was at 65% of its full capacity, the dedicated compressor (Comp4), is also running at part load with around 62% unloading and only 38% loading as shown in Figure 14-8. The reason for this low load factor is due to the average high pressure demand being 14 $m^3$/ min.

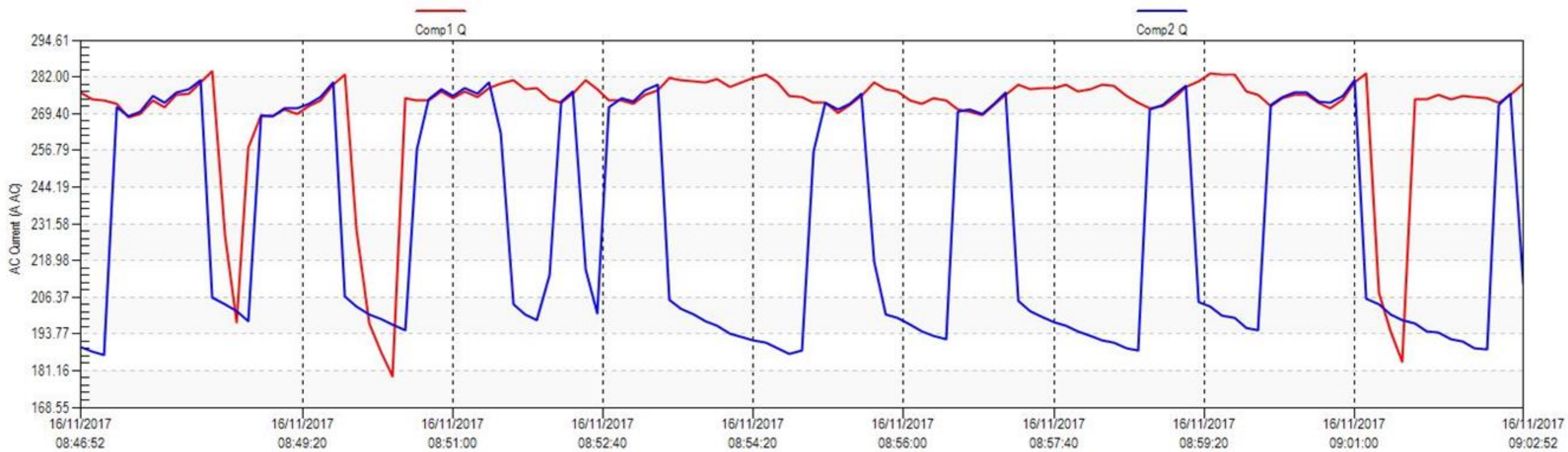



**Figure 14-7: Comp 1 & 2 profile after separation from central network**

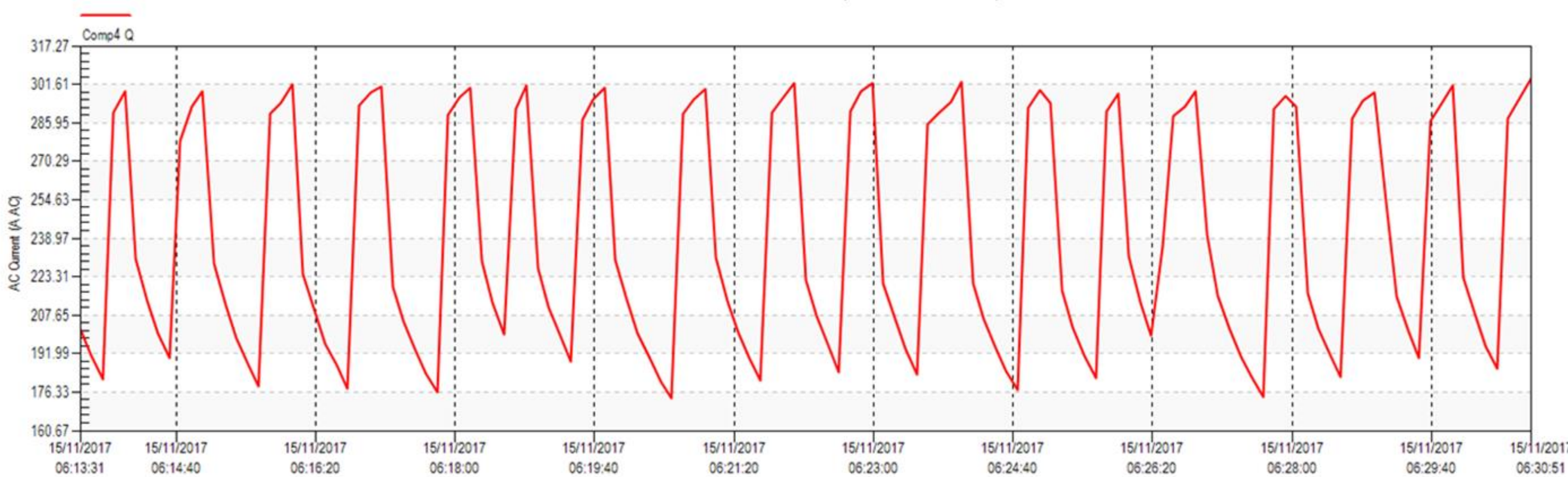



**Figure 14-8: Comp 4 profile after separation from central network**

**Airflow Measurements**

Airflow measurements were taken at the main feeder out of compressors station and at the low-pressure pipe that feed the foam factory. During measurements the main pipe was dedicated to the foam factory, thus the flow in main feeder is representing the total air flow that supply the foam factory.

Figure 14-9: Air flow transducers at main feeder (left) and at LP pipe (right)

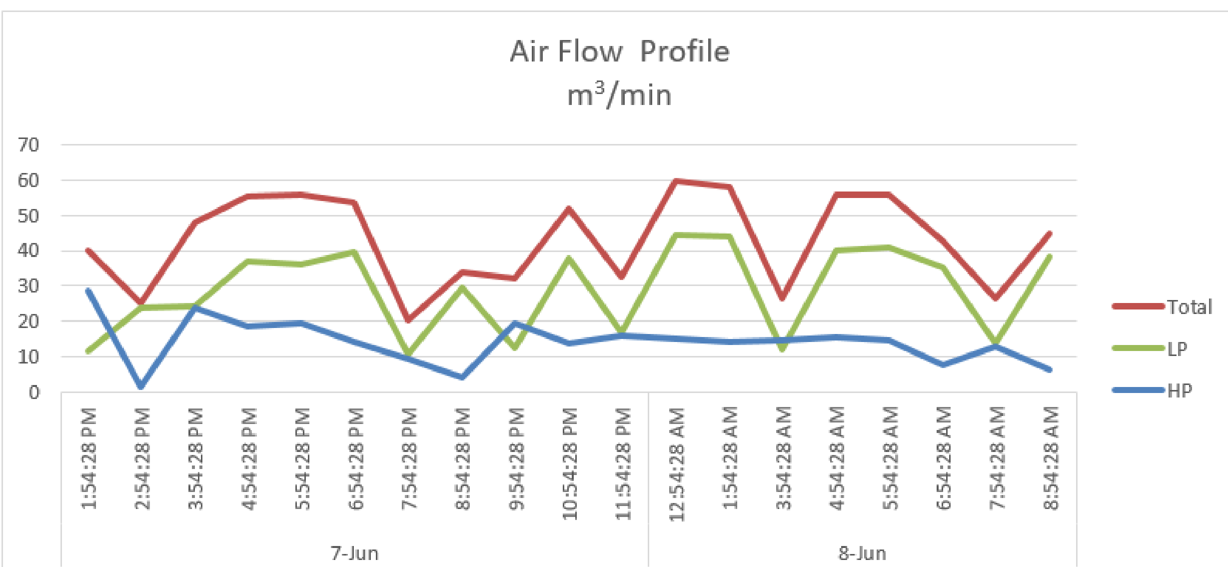



**Figure 14-10: Measured air flow profile during the assessment**

## e. Recommended Energy Efficiency Measures (EEM)

**EEM#1: Optimise the System Pressure**

Since the first CASO assessment after considering the demand profile, the company started to reduce the system pressure gradually without affecting the production. For compressor 1 the pressure was reduced from (6.3 – 7.0) to (4.5 – 5.2) bar, which lead to 145,288 kWh/year and 111,435 EGP/year cost savings, which is 11% of this compressors consumption.

For compressor 2 the pressure was reduced from (6.7 – 7.4) to (4.5 – 5.2) which lead to savings of 13%, quantified to a reduction of 190,704 kWh/year and 146,270 EGP/year cost savings. For compressor 4 the pressure was reduced from (6.4 – 7.1) to (5.8 – 6.8) which lead to savings of 3%, quantified to a reduction of 35,910 kWh/year and 27,543 EGP/year cost savings.

The annual energy savings for the three compressors due to the pressure reduction which was realised after the separation, was 371,902 kWh/year, and 285,248 EGP/year cost savings.

**EEM#2: Reduce Leakages**

Leakages is consuming 478,296 kWh worth of compressed air per year, at an annual cost of EGP 366,853. The company started to install fittings with appropriate sealants, isolated non-operating equipment with a valve in the distribution system, reduced the air pressure of the system where possible, and installed high quality fittings from reputable suppliers including air hoses, tubing, and disconnects.

**EEM#3: Compressor Control**

Multiple compressors operating as part load trimming machines wastes approximately 394,200 kWh/year. It is recommended that only one compressor is used as a trim machine through the implementation of a centralized master controller. In the longer term when considering purchasing new compressors, it is recommended to have compressors of different capacities in order to have more options in staggering compressor control. Alternatively, it is beneficial to have one larger VSD compressor used as the trim machine, or to have a small fixed speed machine as the trim machine if given that storage volume is sufficient.

## 14.2 EZZ Flat Steel Company: Screw Compressor System

### a. Company Background

EZZ Industries is the largest producer of steel in the Middle East and is the market leader in Egypt holding more than 10,000 employees. The company produces long and flat steel products at its manufacturing facilities strategically located in the port cities of Alexandria (EZZ Dikheila-EZDK) and Suez (EZZ Flat Steel -EFS) and in the Egyptian interior at Sadat City and 10th of Ramadan City (ERM). In addition, EZZ Industries owns Al-Jawhara for Ceramics & Porcelain Company that is recognized as the leading brand of quality ceramic, porcelain and decorative accessories in Egypt and the Middle East

The energy consumption in the iron and steel sector depends on the process routes. There are two main routes for producing steel from iron ores. Blast Furnace with basic oxygen furnace (BF-BOF, known as integrated plants) and direct reduced iron with electric furnace (DRI-EF, known as mini-mills). The most common production route in Egypt is the second route. Only 8% of the total iron production in 2012 in Egypt was produced by the integrated BOF plant.

### b. Compressed Air System Overview

EZZ flat steel has three compressed air stations, including two identical screw compressors stations. The initial design of operation was that one of each station is responsible for the operation of one bar mill plant. For both these two stations there is two receiver tanks with a capacity of 10 $m^3$ each. There is a connection between the two tanks to integrate the operation of the two stations.

The third compressed air station is the centrifugal compressor station which consists of 3 compressors and is responsible for the operation of the flat steel plant.

The assessed compressed air system is the two screw compressor stations that serve the bar mill 1 and bar mill 2 plants (BM#1, BM#2). The two stations are identical and each one consists of three GA160W Atlas Copco compressors, with the following specifications:

**Table 14-7: Assessed GA160W compressor technical data**

| | | | |
|---|---|---|---|
| *DATA SHEET FOR THE COMPRESSED AIR SYSTEM (one form for each compressor)* | | | |
| 1 | Manufacturer: ATLAS COPCO | | |
| 2 | Model Number: GA 160W-7.5-50 | Production Date: | |
| | □ Air-cooled ☒ Water-cooled | Type: ☒ Screw □ Centrifugal | |
| | ☒ Oil-injected □ Oil-free | Number of Stages: 1<br>no. of compressors per line is 3 | |
| | | *Number* | *[Unit]* |
| 3 | Rated Capacity at Full Load Operating Pressure (acfm, Nm3/min etc.) | 1070 (1800) | Cfm ($m^3$/hr) |
| 4 | Full Load Operating Pressure (psig, bar etc.) | 109 | psig |
| 5 | Maximum Full Flow Operating Pressure | 116 | psig |
| 6 | Drive Motor Nominal Rating (hp, kW) | Star-Delta | |
| 7 | Drive Motor Nominal Efficiency | --- | [%] |
| 8 | Motor Nameplate Power | 160 | kW |
| 9 | Motor Power Factor | 0.83 | [-] |
| 10 | Motor Voltage | 400 | [V] |
| 11 | Fan Motor Nominal Rating (if applicable) | N/A | |
| 12 | Fan Motor Nominal Efficiency | N/A | [%] |

The compressed air network has one air receiver tank per production line, but the two tanks are linked mechanically. The capacity of the tank is 10 $m^3$. These tanks are downstream of the compressor station, whereafter there are three oil filters. The rated capacity of these filters are 1872 $m^3$/h FAD and its initial pressure drop is 0.12 barg. After the oil filters there are three FD750 refrigeration dryers (each one dedicated to a single compressor) of 1710 $m^3$/h FAD rated capacity.

After the refrigeration dryer there is another oil filter with a capacity of 430 $m^3$/h FAD, and its initial pressure drop is 0.2 barg. After this oil filter there is a “adsorption-absorption” dryer, with a capacity of 223 $m^3$/h FAD and an initial pressure drop of 0.12 barg. Further downstream there is a dust filter with a capacity of 430 $m^3$/h FAD and its initial pressure drop is 0.05 barg.

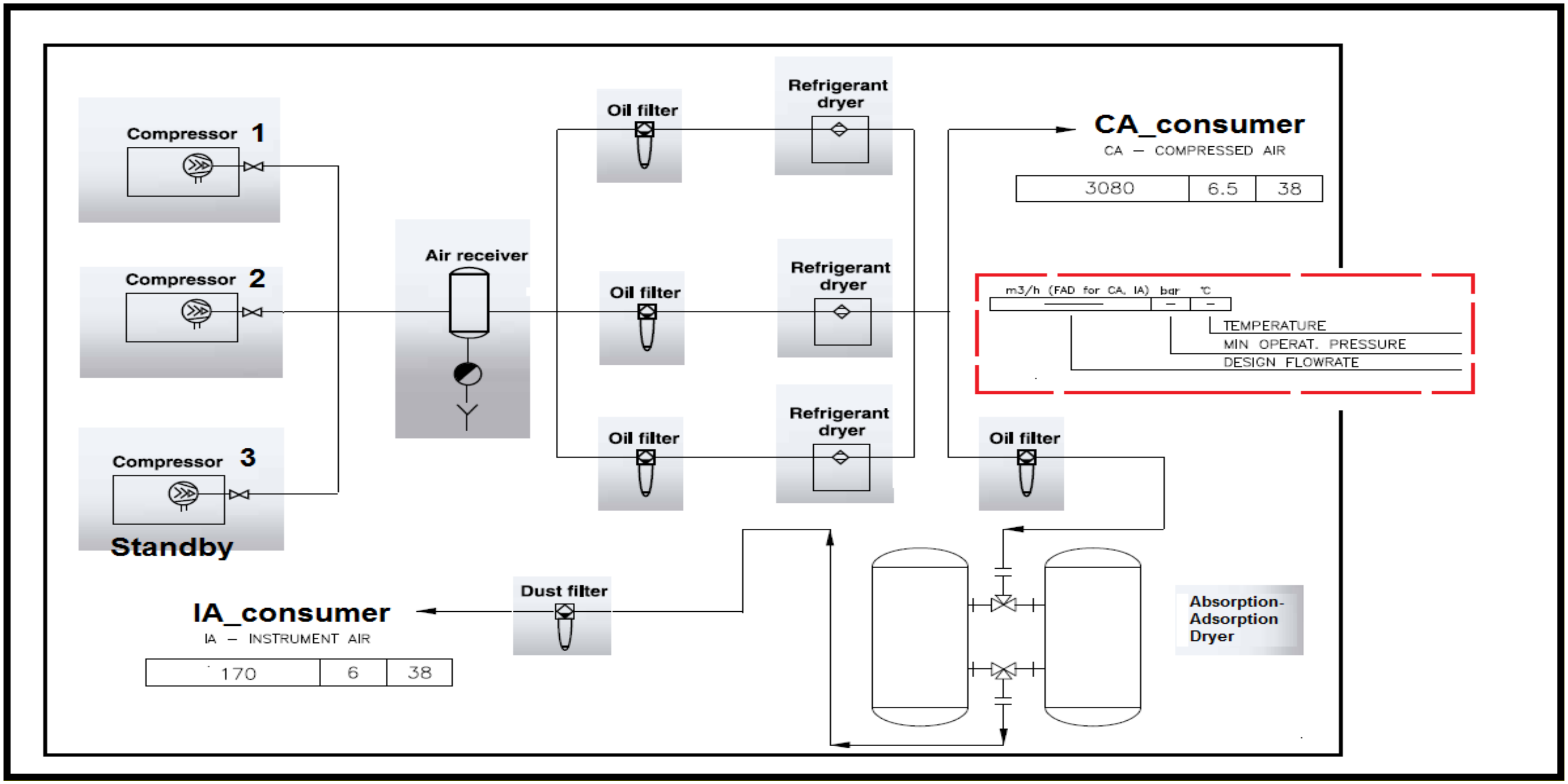



**Figure 14-11: Supply side single line diagram**

The designed flow demand for all consumers is 3300 $m^3$/hr, divided into two areas; Compressed Air (CA) & and Instrument Air (IA). CA consumes 3080 $m^3$/hr and IA consumes 170 $m^3$/hr, while the remaining 40 $m^3$/hr is used for the regeneration of the absorption/adsorption dryer.

The compressors are controlled by a sequential controller with a simple algorithm model to control the system; two compressors are in operation to deliver 3300 $m^3$/hr and the third compressor is the stand by compressor to be operated when there is a sudden demand requirement.

However, currently, due to the malfunctions of compressor#3 in BM#1 and compressor#1 in BM#2, there are two control strategies:

1. In the case where only one production line is in operation, there are four compressors in service. Three of them are running to deliver to the required CA and IA (and also to recover the losses of leakage and other inappropriate uses).
2. In the case where two production lines are operating, there are two compressors in-service and the rest of compressed air requirements is recovered through the connection with the centrifugal station from the flat steel plant.

## c. Baseline

The current calculated baseline for the annual kWh consumption of the screw compressor station is 1,849,719 kWh/year. The compressed air supply is measured at 18.75-million $m^3$ per year. The specific energy of the system is 0.099 kWh/$m^3$. The cost of compressed air is equivalent to 0.06 EGP/$m^3$.

## d. Key Findings during the Assessment

**Compressor Operation**

The as-found counter meter on the compressors gives the following data regarding the operation of the compressors:

**Table 14-8: Compressors' counter data**

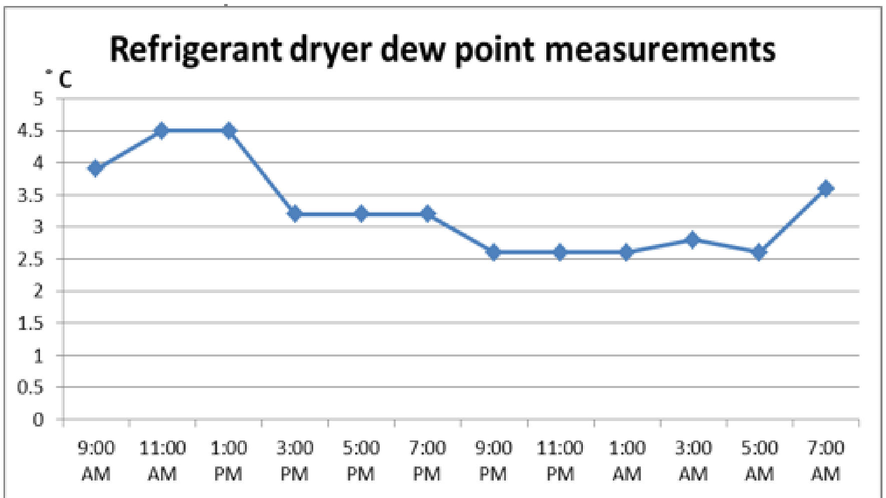
| Comp#1 | |
|---|---|
| Running hours | 29156 |
| loading hours | 26556 |
| motor starts | 3881 |
| load % | 91% |

| Comp#2 | |
|---|---|
| Running hours | 23509 |
| loading hours | 16767 |
| motor starts | 9892 |
| load % | 71% |

| Comp#3 | |
|---|---|
| Running hours | 25091 |
| loading hours | 18531 |
| motor starts | 9574 |
| load % | 74% |

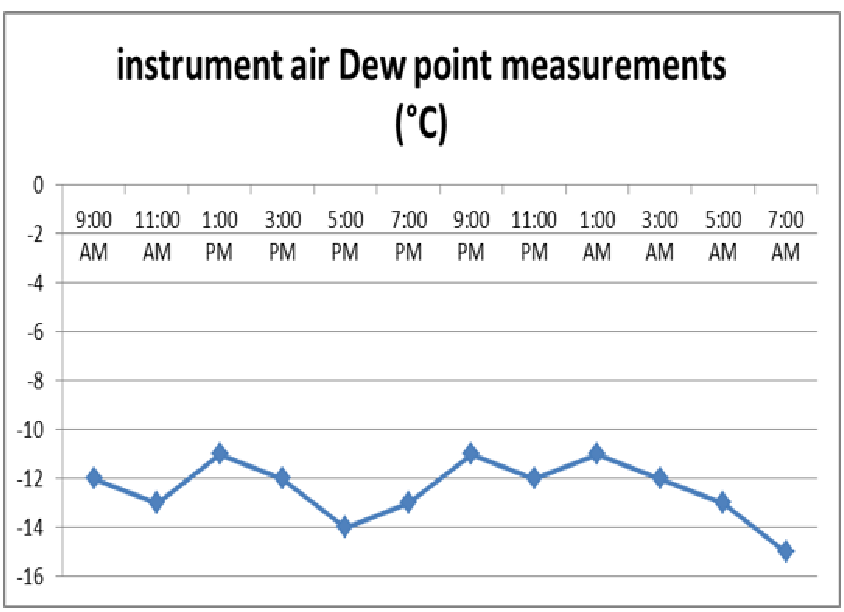
**Pressure Dewpoint Measurements**

For process air the dryer used is the refrigeration dryer with the dewpoint setting of 4˚C. For instrument air the dryer used is the adsorption/absorption dryer, with a dewpoint setting of -12˚C. Dewpoint measurements are recorded manually each hour.




| time(hrs) | Dryer#3(°C) |
|---|---|
| 9:00 AM | 3.9 |
| 11:00 AM | 4.5 |
| 1:00 PM | 4.5 |
| 3:00 PM | 3.2 |
| 5:00 PM | 3.2 |
| 7:00 PM | 3.2 |
| 9:00 PM | 2.6 |
| 11:00 PM | 2.6 |
| 1:00 AM | 2.6 |
| 3:00 AM | 2.8 |
| 5:00 AM | 2.6 |
| 7:00 AM | 3.6 |

**Figure 14-12: Manually recorded pressure dewpoint data for one day on the refrigeration dryer**




| time(hrs) | I.A_Dewpoint (°C) |
|---|---|
| 9:00 AM | -12 |
| 11:00 AM | -13 |
| 1:00 PM | -11 |
| 3:00 PM | -12 |
| 5:00 PM | -14 |
| 7:00 PM | -13 |
| 9:00 PM | -11 |
| 11:00 PM | -12 |
| 1:00 AM | -11 |
| 3:00 AM | -12 |
| 5:00 AM | -13 |
| 7:00 AM | -15 |

**Figure 14-13: Manually recorded pressure dewpoint data for one day on the adsorption/absorption dryer**

**Distribution**

The compressed air velocity for the CA end use demand was calculated at 14.7 m/s, which is acceptable for end use demand and indicates that the piping is of sufficient diameter.

## e. Recommended Energy Efficiency Measures

**EEM#1: Reduce Leakages**

A load/unload leakage test was performed when bar mill 1 plant's operation was stopped. The quantified leakage rate over five tests was averaged to 16.6%. This represents leakage waste of 5,933,859 $m^3$/year, which represents energy losses of 593,386 kWh/year. According to the electricity prices of 0.63 EGP/kWh, this leakage is costing 373,833 EGP/year.

Reducing the current leakage level by 50% could be targeted. This will exceed the 10% generic recommended leakage target. A leak management system will be adopted in the company to identify and tag leakages, and to fix it periodically.

The investment required is 50,000 EGP, which could yield energy savings of 593,386 kWh/ year at a cost saving of 373,833 EGP/year. This project has a 2 month simple payback, having a $CO_{2e}$ reduction of 338 ton/ year.

**EEM#2: Inappropriate Uses**

During the assessment, inappropriate uses at each station was identified. These were in the form of 28 air cooling points that uses compressed air at 6.5 bar, from 12 mm diameter pipe lines to hoses with 6 mm diameter.

One of the proposed solutions to optimize the consumption of these 28 points of cooling air is to reduce the pressure from 6.5 bar to 4 bar via pressure regulators. The quantified savings available for this implementation was calculated at 1,233,806 kWh/year and 777,298 EGP/year, at an implementation cost of 28,000 EGP/year – a payback of less than one month.

**EEM#3: Install Zero Air Loss Condensate Drain Valves**

During the assessment it was found that the condensate drains have malfunctioned and were all replaced with manual valves which is permanently open at approximately 50 % of their diameters. The system with its current operation has the following drains:

- 4 drains for the compressor after coolers
- 4 drains for refrigerant dryers
- 2 drains for air receivers

- 8 drains for oil filters
- 2 drains for dust filters

These 20 condensate drain valves are quantified to be wasting an equivalent of 30.6 $m^3$/min of compressed air in total. This represents an electricity loss of 1,321,920 kWh/year and a cost of 832,810 EGP/year. It was recommended that these drain traps are replaced with electronic level sensing drain traps. At a total investment of EGP 400,000 EGP, the payback for the project is 6 months.

**EEM#4: Increase the System's Storage Volume**

Increasing the system's storage volume will enable the operating trim compressor to reduce its cycling durations and run at an increased load factor. Consequently, 64,800 kWh/year could be saved at a cost reduction of 40,824 EGP/year. The total cost of installing two new air receivers with a capacity of 5 $m^3$ each is 30,000 EGP – a payback of 6 months.

**EEM#5: Connections with the Centrifugal Compressed Air Network**

The centrifugal compressed air station, consisting of three identical centrifugal compressors, are dedicated to the flat steel plant. Each compressor can deliver 10,800 $m^3$/hr. The current operation of this station is two compressors running with the third compressor operating at no load. Ampere measurements during the assessment indicated that one of the two compressor is running at full load and the second one is running at minimum load with the by-pass valve open 100%. Two options for implementation can be considered:

1. Stop one of the two screw stations and compensate the air demand of one Bar Mill from the centrifugal station. Available savings are quantified to a reduction in compressed air of 237,600,00 $m^3$/year, equivalent to 2,376,000 kWh/year and 1,496,880 EGP/yr.
2. Totally stop the screw compressor stations and supply the air requirements to BM#1 and BM#2 from the centrifugal stations with its current control strategy. The energy saving represents 4,752,000 kWh/year and the cost saving is approximately 3,000,000 EGP/year. There is no capital investment cost required for this opportunity, thus the payback is immediate.

**Recommendation Summary Table of Energy Efficiency Measures**

**Table 14-9: Energy Efficiency Measures Recommendation Summary Table**

| **Annual Energy Savings Impact** | |
|---|---|
| Total Electricity Savings [kWh] | 5,589,912 |
| Carbon Dioxide Equivalent Reduction [Tons/year] | 3,186 |

| | |
|---|---|
| Total Cost savings [EGP] | 3,521,645 |
| Total Investment Cost [EGP] | 508,000 |

## 14.3 EZZ Flat Steel Company: Centrifugal Compressor System

### a. Compressed Air System Overview

Three Ingersoll Rand three stage dynamic centrifugal compressors are located in a single compressor plant. Each compressor is controlled by means of inlet throttling control. The air compressor data is shown in the table below.

**Table 14-10: Existing plant air compressors**

| Compressor Manufacturer | Model | Capacity Control Method | Motor Nominal Power kW | Rated Capacity | Operating Pressure |
|---|---|---|---|---|---|
| **Ingersoll Rand** | C155MX3-2RH<br>Mfg. year. 2000 | Inlet butterfly throttling control | 1050 kW | 170m$^3$/min | 7 bar(g) rated<br>7.1 bar(g) set point |

There are three different air demands are defined as:

Compressed air (CA) (Max. Simultaneous flow rate = 7816 m$^3$/h, operating pressure (barg)=6 bar$_g$

1. Instrument Air (IA) (Max. Simultaneous flow rate = 280 m$^3$/h, operating pressure=5 bar$_g$
2. Process Air (PA) (Max. Simultaneous flow rate = 10,400 m$^3$/h, operating pressure =6 bar$_g$

The compressed air generated from the compressor station is split in two ways due to air quality requirements; part of the compressed air supplies the process air demand through a 10 m$^3$ air storage tank directly downstream of the after cooler, and the remaining compressed air goes to another 10 m$^3$ air storage tank where there are three parallel connected Ingersoll Rand refrigerated air dryers.

Each dryer is rated for 4800 m$^3$/h FAD and 12 kW with 7 (barg) working pressure, delivering a 3 $^o$C pressure dew point. Part of the refrigeration dryers output supplies the compressed air demand. The remaining output is treated again by a 470 Nm$^3$/h desiccant adsorption dryer with a maximum working pressure of 6.5 barg and -10$^o$C design pressure dew point, which supplies instrument air demand. The changeover between the towers is controlled by a timed cycle.

The pre-filter of the desiccant dryer is filtering oil to 0.01 micron. The after-filter is filtering particles to 0.01 micron to capture any entrained desiccant dust from the drier compressed air.

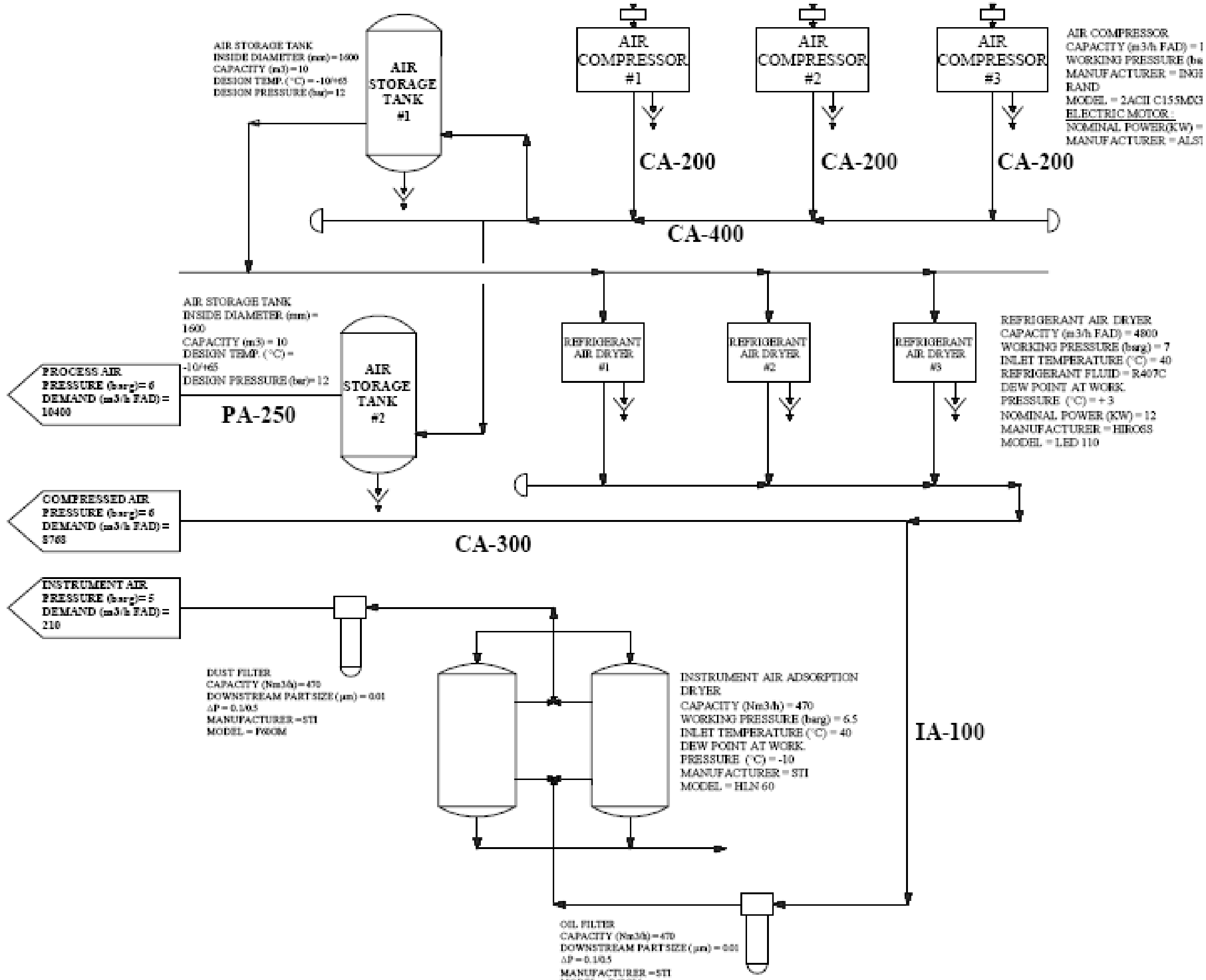



**Figure 14-14: Existing Compressor Installations & Mainline Distribution Piping**

In the compressed air plant the delivery piping for each of the three compressors is 8 inches, which is feeding the main distribution header of 16 inches. The main distribution header is subdivided to feed a pipe line of 12 inches for compressed air demand, and a pipe line of 10 inches for process air demand. 4 Inch distribution drops from the 12 inches compressed air demand, feed instrument air demand.

A Compressed air flow meter is installed on the 12-inch compressed air demand pipe line after the instrument air sub pipe. The operating velocity in delivery pipe of each compressor at full load condition are calculated at 11.3 m/s, which is nearly double the recommended design velocity of 5m/s. A 12-inch piping should be considered. The peak velocity in the 12-inch delivery pipe of compressed air demand is calculated at 4.2 m/s, which is acceptable and within recommended limits.

## b. Baseline

The three centrifugal compressors consume approximately 19,576,000 kWh per annum, at an electricity cost of 14,682,000 EGP per annum. The average total compressor demand is 2,447 kW. The average actual air demand from the flow meter display is 9,200 $m^3$/hr (153 $m^3$/min).

***Actual compressed air specific power consumption is 15.9 kW/$m^3$/min, which is extremely high when compared to the rated specific power consumption for the compressor of 6.1 kW/$m^3$/min!*** The reason for this difference is due to the huge amount of blow-off compressed air that goes to the atmosphere.

## c. Key Findings during the Assessment

**Measured Data**

The current operation situation for the compressed air plant is to have three centrifugal compressors running continuously. The measured compressed air demand for plant production on the main delivery pipe is 9,500 $m^3$/h.

The annual electricity consumption of compressor # 1, based on the installed amperage measurements during the assessment, is calculated at 6,733,353 kWh/year. The annual electricity consumption of compressor # 2 is calculated at 6,279,446 kWh/year. The annual electricity consumption of compressor # 3 is calculated at 6,582,070 kWh/year.

Thus the total electrical energy consumption for the three centrifugal compressors is 19,576,000 kWh per annum, at an electricity cost of 14,682,000 EGP per annum. The recorded amperage load profiles during the assessment is shown in the figures below.

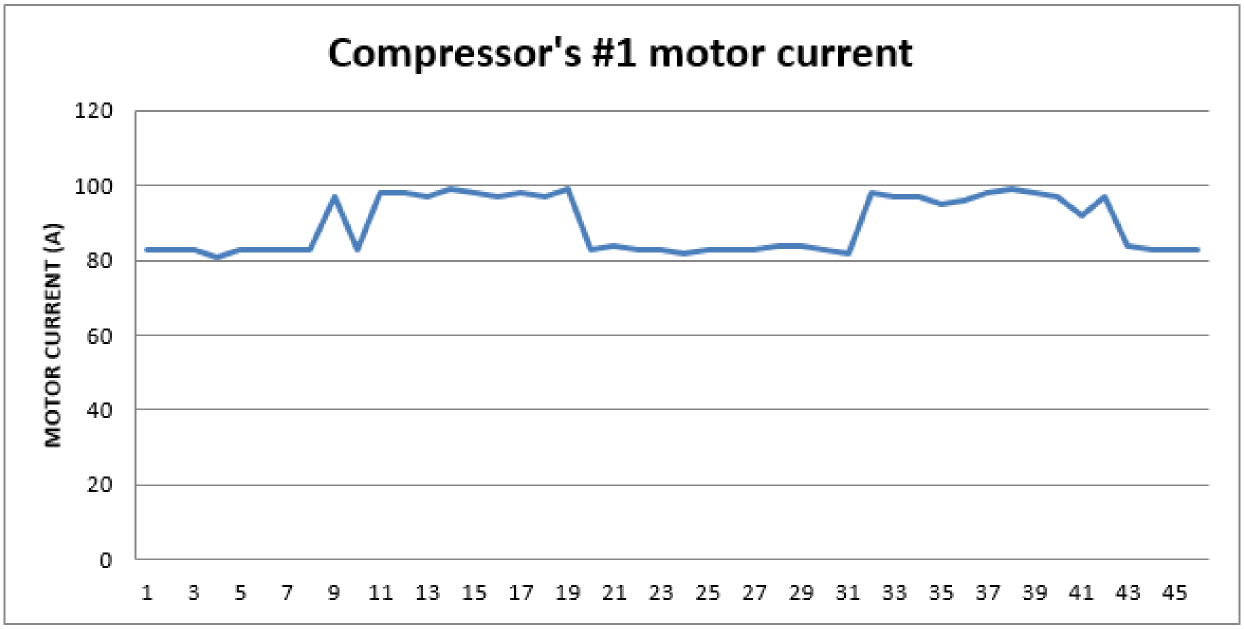



**Figure 14-15: Compressor #1's motor drawn current trend**

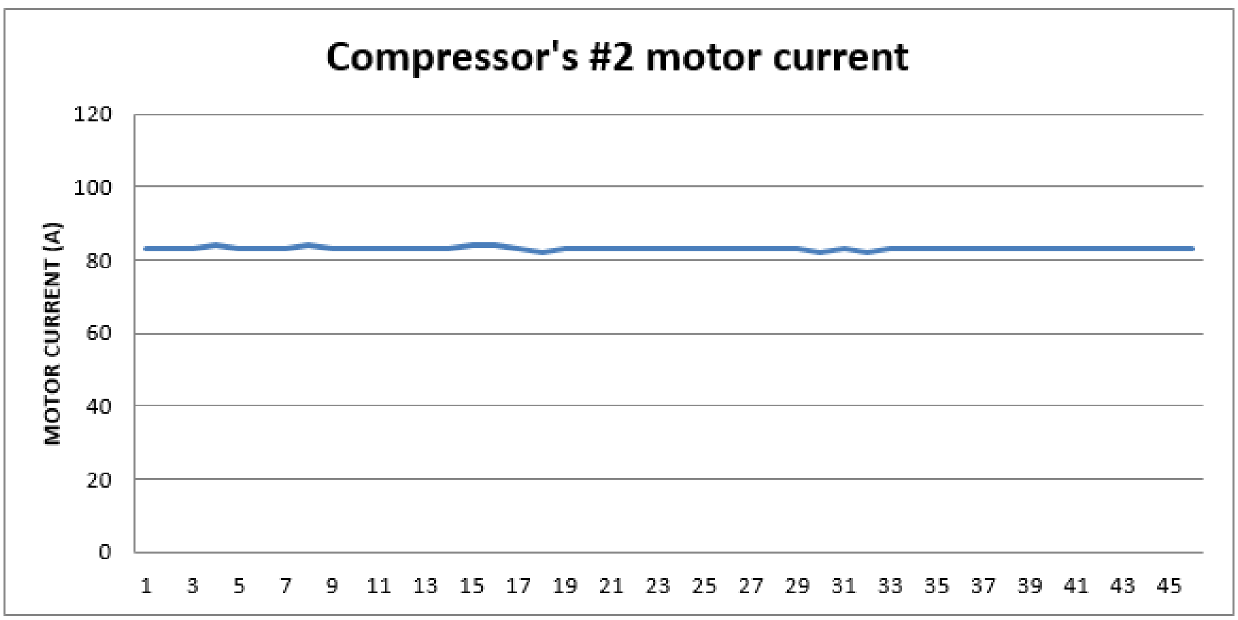



**Figure 14-16: Compressor #2's motor drawn current trend**

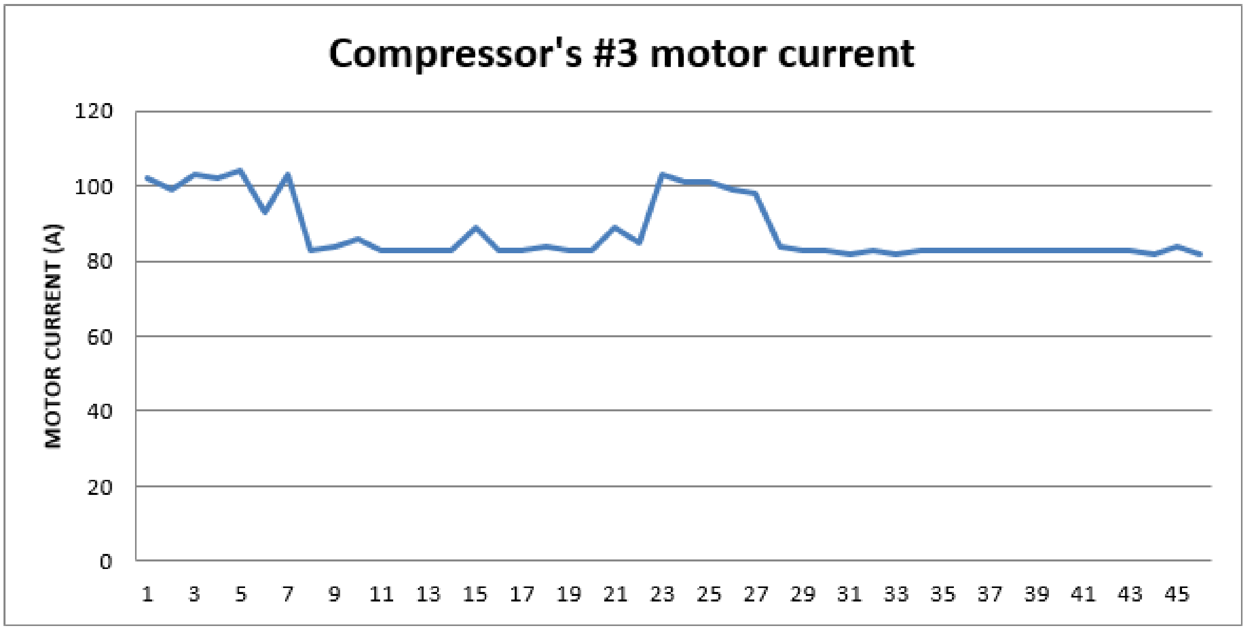



**Figure 14-17: Compressor #1's motor drawn current trend**

Pressure measurements were taken hourly at three different locations for 24 hours. The average pressure at the main pipe connected directly to the compressors was 7.0 barg, the average pressure at the delivery pipe of the compressed air plant and after the refrigeration dryers was 6.7 barg, and the average pressure at the farthest compressed air user was 6 barg.

Hourly pressure dew point measurements were also taken manually from the refrigeration dryers' display throughout 24 hours, giving +7°C.

**Inappropriate Uses**

Evidence of inappropriate uses was identified during the assessment. Compressed air was used for cooling a group of withdrawal motors due to overheating problems, as shown below. The overheating led to motor thermal trips and a risk of production interruption.


**Figure 14-18: Withdrawal motors with compressed air cooling supply (before remedy action)**

The cooling fans for most of the motors was found to be damaged. The motor temperatures were measured and was found to be below the limit of its rated H-insulation class.

Subsequently, new cooling fans were installed for each motor and steel cover plates were installed around the motors that are exposed to potential overheat. Eventually the compressed air supply for the motor was permanently closed as shown below.

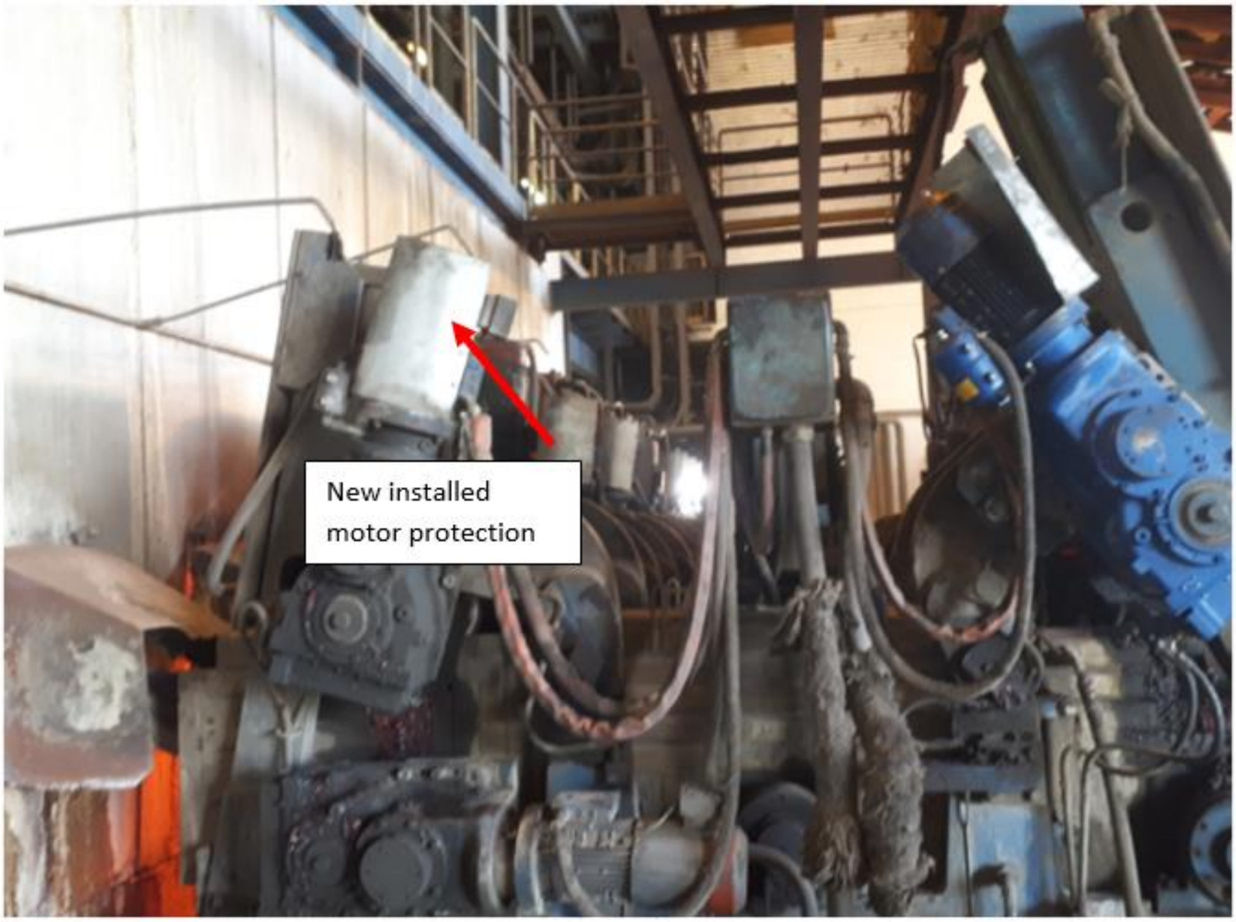



**Figure 14-19: Withdrawal motors after new covers were installed (after remedy action)**

Another case was found where a 20 mm compressed air pipe is used for cooling a hot area in the lime plant shaft furnace body. An axial fan with an 11kW motor was installed in front of the overheated area as an alternative solution and the compressed air supply was closed as shown in photo below.


**Figure 14-20: Lime plant shaft furnace with new air fan installed (after remedy action)**

## d. Recommended Energy Efficiency Measures

**EEM#1: Shut down one 1050 kW centrifugal compressor**

The original plant was designed to operate with only two prime compressors and one on standby to supply compressed air demand for flat production. Due to the increase of air demand as a result of inappropriate uses e.g. cooling of motor bearings and sensors, and due to installing new pneumatic equipment, the standby compressor was also introduced. The compressed air plant is thus working now with three compressors in service, without any on standby.

When production was changed over from flat to billet production, the compressed air demand decreases significantly according to the designed air demand. However, the compressed air plant was left as is without any reduction in operating compressors.

A reduction in the amount of compressed air can thus be achieved due to the changing over from flat to billet production, the reduction in specific power consumption, and the decrease in the amount of blow-off air.

The designed amount of compressed air required for flat production is 18,496 $m^3$/h. Each compressor is rated for 10,200 $m^3$/hr, however all three compressors are continuously in operation – able to deliver 30,600 $m^3$/hr. For the billet production, the new compressed air demand is only 5,935 $m^3$/hr, which is well within the capabilities of only one compressor. Because the compressed air supervisory control is currently out of service, another compressor will have to be left continuously running to supply air demand if the base compressor failed.

The actual average compressed air consumption for billet production measured by the process flow meter was 9,200 $m^3$/h, which is larger than the designed demand given above of 5,935 $m^3$/hr.

A test was performed during the assessment to verify whether two compressors are capable to supply the air demand for billet production. A load profile was developed for each compressor when the three compressors are working together and when two compressors are running and the third compressor is unloaded. Test results were taken from each compressor's user interface by the operator for 103 records with one-hour time intervals.

**Test Results:**

(Refer Figure 14-21, Figure 14-22, and Figure 14-23 below) When the three compressors are working during billet production:

- Compressor #2 is minimum loaded with inlet valve throttled to maximum limit (30%), with the by-pass (blow-off) valve fully open at 100 %. The operating sheet of this compressor stated that during this operating condition no flow is supplied to the system from this compressor.
- Compressors #1& 3 are working part loaded with continuous changes in inlet valve and by-pass valve positions.

When compressors 1&3 are loaded and compressor #2 is unloaded:

- At hour record 47 in the profiles below, Compressor #2 is unloaded as indicated.
- When Compressor #2 is unloaded, its motor current drops from 83 A to 46 A with the inlet valve position at 11 % and the by-pass valve position fully open at 100 %, with no contribution to the system demand.
- Compressor #1 increased its amperage, with the inlet valve opening more and the by-pass valve completely closed.
- Compressor #3 is at minimum load with maximum throttling capacity (inlet valve opens 30 %) and variable by-pass valve position.

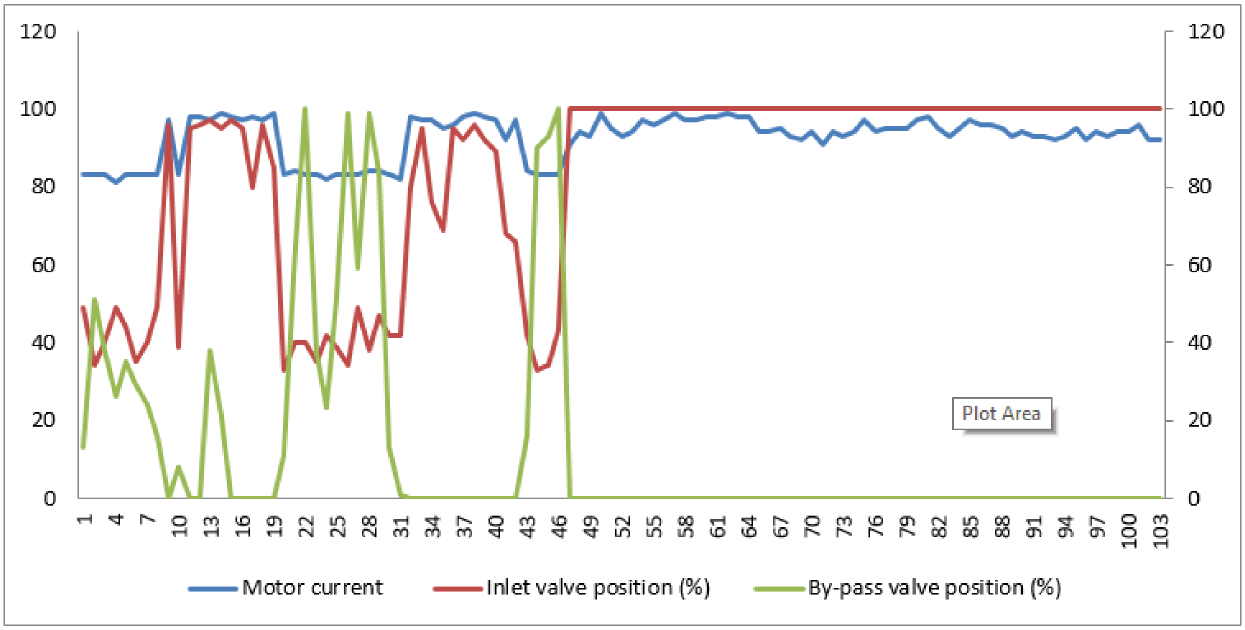



**Figure 14-21**: **Compressor #3 valve position versus motor current trend**

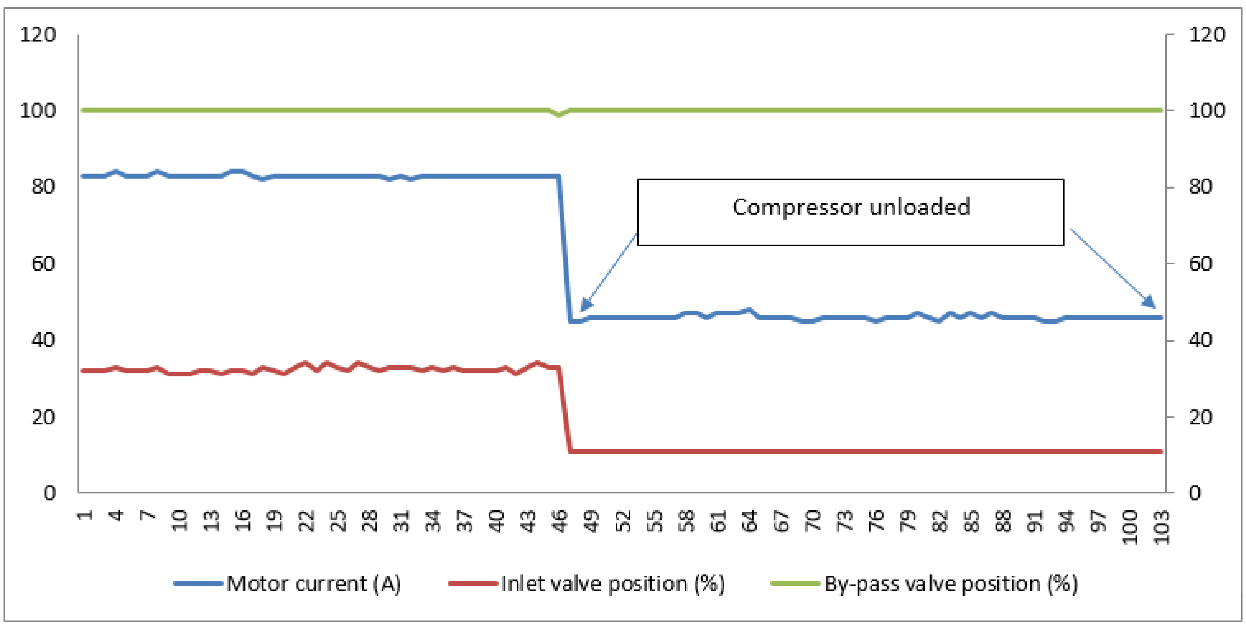



**Figure 14-22: Compressor #2 valve position versus motor current trend**

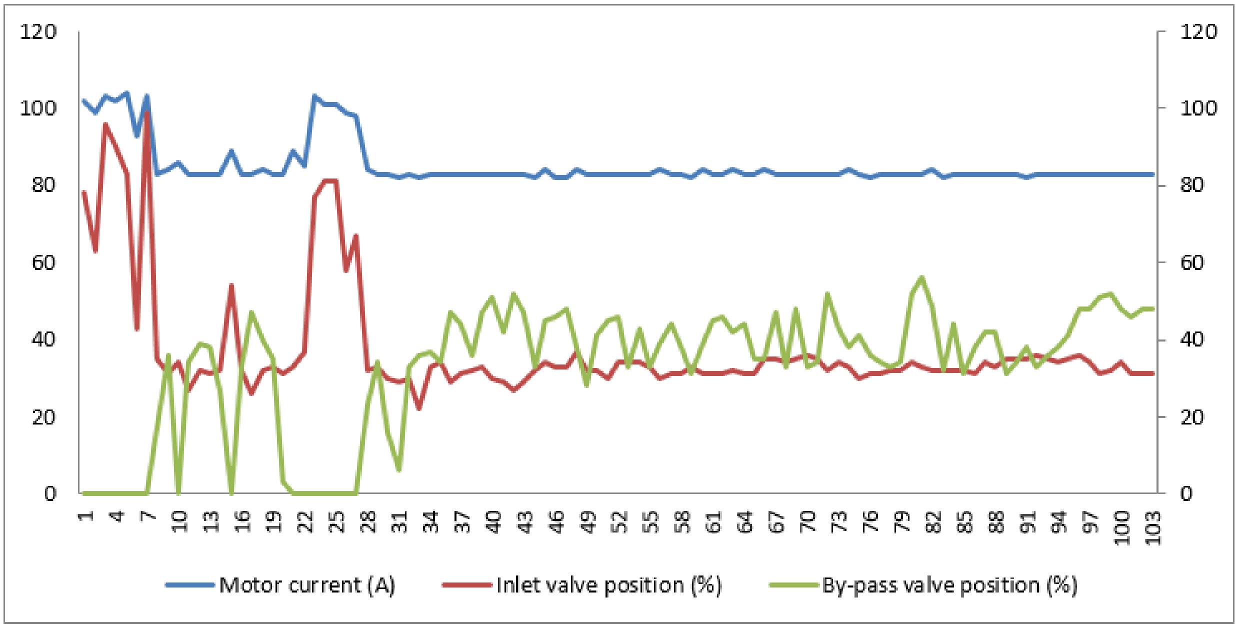



**Figure 14-23: Compressor #3 valve position versus motor current trend**

Consequently, due to the successful test, Compressor #2 was shut down as agreed with all concerned departments during billet production, and this action achieved savings with no cost investment. The annual energy savings achieved is 6,272,000 kWh at a cost saving of 4,263,391 EGP/year. The specific power consumption for compressed air has reduced from 15.9 kW/m$^3$/min to 11 kW/m$^3$/min.

**EEM#2: Eliminate some inappropriate uses of compressed air**

Some inappropriate uses identified during the assessment included cooling applications for equipment exposed to overheating, and permanently open condensate traps. To ensure commitment, a presentation with subject ‘’energy efficiency improvement in the compressed air plant’’ was held for top and senior management. This presentation targeted the following topics:

1. Compressed air plant annual consumption cost
2. True cost of compressed air as an expensive source of energy with low efficiency, emphasising that its use must be scrutinised
3. Actual inappropriate uses of compressed air in the plant and potential alternatives
4. Further potential energy saving opportunities

As a result of this awareness creation to management, management commitment was established. This enabled the original design of the compressed air network to be restored and any unapproved modifications were eliminated. The compressed air cooling applications were also replaced with applicable alternatives, as a part of the continual improvement programme.

The annual kWh savings from the elimination of this inappropriate use of compressed air is 917,928 kWh, at a cost saving of 688,446 EGP per year.

The savings from replacing the 20 mm compressed air pipe with the 11kW axial fan in the lime plant, is 524,370 kWh/year, at energy cost reduction of 393,000 EGP per annum, and a payback of 13 days.

**EEM#3: Shutdown another centrifugal compressor with 1050 KW power rating**

As discussed, when production changes over from flat production to billet production, the theoretical compressed air demand dropped to 5,935 $m^3$/hr, although the measured demand from the flow meter dropped to 9,500 $m^3$/hr. This theoretically means that only one compressor should be sufficient to supply this demand.

A test was done, and for any unanticipated failure of the operating compressor, another compressor has been left running at its reserved minimum capacity. To minimise the waste to blow-off, a pre-erected tie line between this network and the bar mill production was opened, and all compressors in this network were shut down. Despite this, the compressor was blowing off, as shown in the figure below:

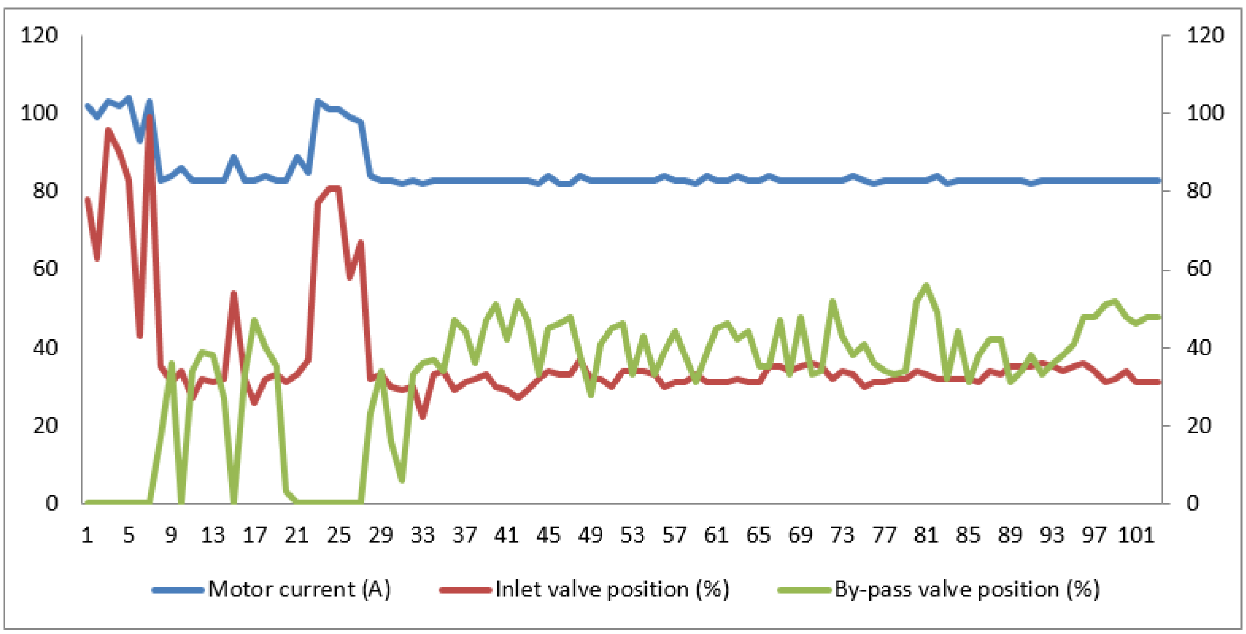



**Figure 14-24: Compressor #3 load profile trend versus valves' positions, when the tie line between the two compressed air networks was opened**

After the test, the decision was taken by management to:

1. Close the tie line between the two networks

2. Investigate and remediate the reason for the deviation in the current compressed air measurement (9,200 $m^3$/hr) and the theoretical demand (5,935 m3/hr)
3. Run the plant with only one compressor

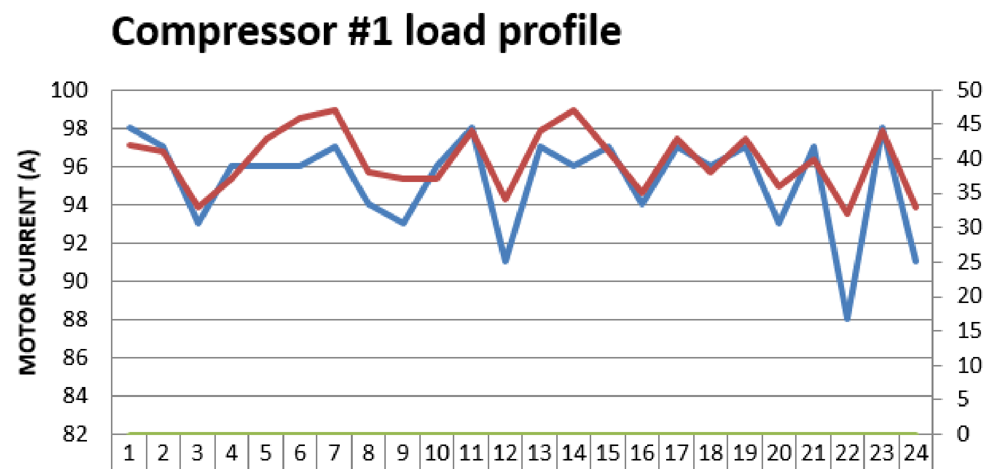
The risk of an unanticipated shutdown of that compressor is incurred by the management in return of the earned savings, until permanently securing this condition by putting the faulty sequencer controller back into service. Accordingly, a plan of action was prepared to implement these decisions in three steps:

1. Investigate and reduce the compressed air demand through:
   - A daily investigation tour in the stopped flat production line
   - Stopping many unused compressed air equipment
   - Reducing the inappropriate uses of compressed air as discussed in EEM#1

   These resulted reducing the compressed air demand measured during billet production from 9,200 $m^3$/hr to avg. 6,000 $m^3$/hr

2. Stop compressor #3 and calculate the savings
3. Trend load profiles for the only running compressor based on the new situation, and calculate consumption and specific power improvements

Figure 14-25 proves the new operating philosophy in which the by-bass valve is permanently closed, with Figure 14-26 showing the new and improved measured flow demand recorded hourly over 24-hours. The average compressed air demand is now 5,887 $m^3$/hr (98 $m^3$/min), and the specific power consumption have improved to 9 kWh/$m^3$/min.




| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Motor current | 98 | 97 | 93 | 96 | 96 | 96 | 97 | 94 | 93 | 96 | 98 | 91 | 97 | 96 | 97 | 94 | 97 | 96 | 97 | 93 | 97 | 88 | 98 | 91 |
| Inlet valve position (%) | 42 | 41 | 33 | 37 | 43 | 46 | 47 | 38 | 37 | 37 | 44 | 34 | 44 | 47 | 41 | 35 | 43 | 38 | 43 | 36 | 40 | 32 | 44 | 33 |
| By-pass valve position (%) | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |

**Figure 14-25: After implementation compressor #1 motor current trend versus valves' position**

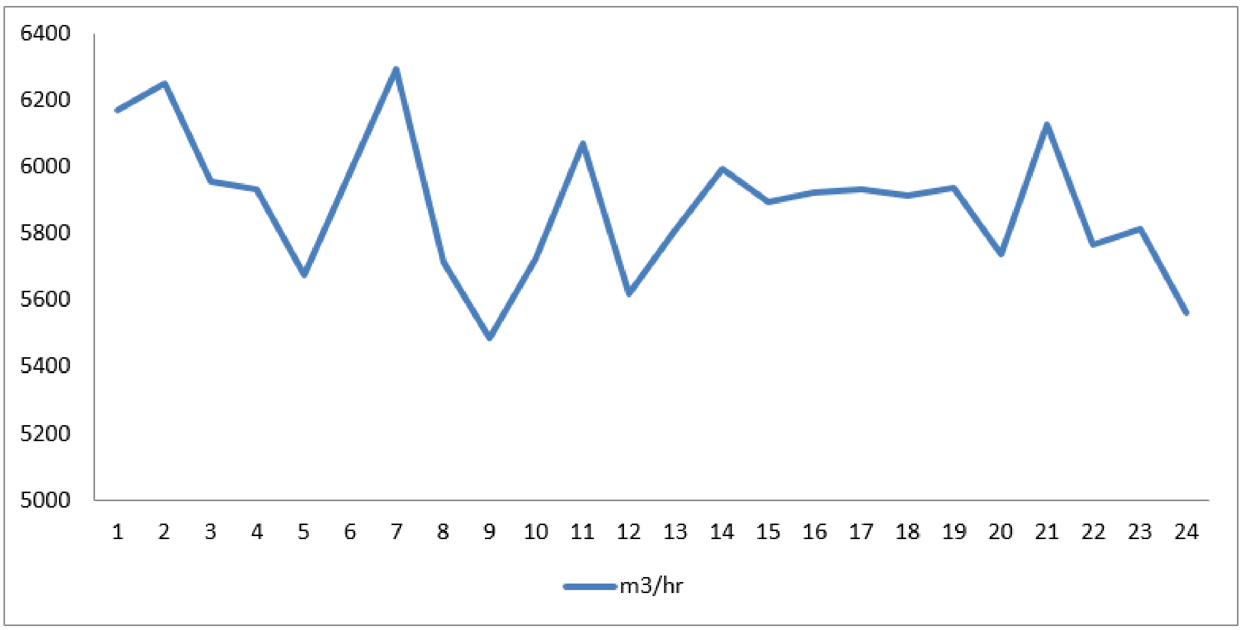



**Figure 14-26: After implementation total measured compressed air demand**

**Summary of Savings Achieved**

Before the UNIDO compressed air systems optimisation training and assessment at EZZ Flat Steel, three centrifugal compressors were operational, consuming approximately 19,576,000 kWh/y at a electricity cost of 14,682,000 EGP/y. After the assessment and successful implementation, only one centrifugal compressor was operational, consuming approximately 7,187,318 kWh/y at a cost of 5,390,488 EGP/y for the same amount of production.

This translates to a 63% reduction in consumed electrical energy. The savings achieved are 12,388,682 kWh/y and 9,291,512 EGP/y. As mentioned, the specific power improved from 15.9 kW/m$^3$/min to 9 kWh/m$^3$/min.

This great improvement and successful case study was ultimately realised through an improved control philosophy, and with strong commitment from top management.

## 14.4 Alnahda Cement Plant: Qena

### a. Company Background

ElNahda cement is one of the recently established Egyptian cement producers. The production plant is located at Qena Governorate, in Upper Egypt. The company was established in 2008, with the production line commissioned in 2012. The company owns one grey cement production line, supplied by a Chinese Contractor. It has a design capacity of 5500 ton per day of clinker, while the actual capacity reaches 6150 ton per day clinker. The operation and maintenance of the production line is contracted to an O&M company, which is one of the leading companies in the Middle East (Arab Swiss Engineering Company, ASEC). Representatives from ASEC were directly in close cooperation with the Nahda company team throughout the whole process of the EnMS and CASO assessments. The plant is composed of one production line using a dry process. The company serves domestic market and exports products to Arab, Africa and European markets.

### b. Compressed Air System Overview

The company has two main compressed air systems, each running with five screw compressors. One compressed air system serves the clinker production section, while the other system serves the cement grinding and packing. Based on discussion and agreement with the plant staff, this report focuses on the compressed air system of the clinker production.

**Supply Side**

In order to satisfy the compressed air demand of the different users, five screw compressors are installed in the compressor room to generate the required compressed air. Initially, the idea was to have four compressors running and the fifth as a stand-by. However, the current situation is to have the five compressors running simultaneously with no stand-by.

The screw compressors are Ingersoll Rand R132IU-A8 compressors. The compressors are air cooled, oil injected, screw type. The rated Free Air Delivery (FAD) of each compressor is 22.3 $m^3$/min. Each compressor is followed by an Ingersoll Rand IR265RCW water cooled refrigerant air drier, and a receiver tank of 2500 L capacity. All pipes from the compressor to the tank are of size DN80. A single collecting header DN200 is installed after the receiving tanks to deliver the compressed air to users. The air driers are out of service, and the humid compressed air is delivered to the tanks directly. The receiver tanks are equipped with manual drain valves, which are permanently open to drain the condensate from the tanks. The condensate is drained to the ground under the tanks. Figure 14-27 and Figure 14-28 illustrates a single line representation of the clinker compressed air system of Nahda Company.

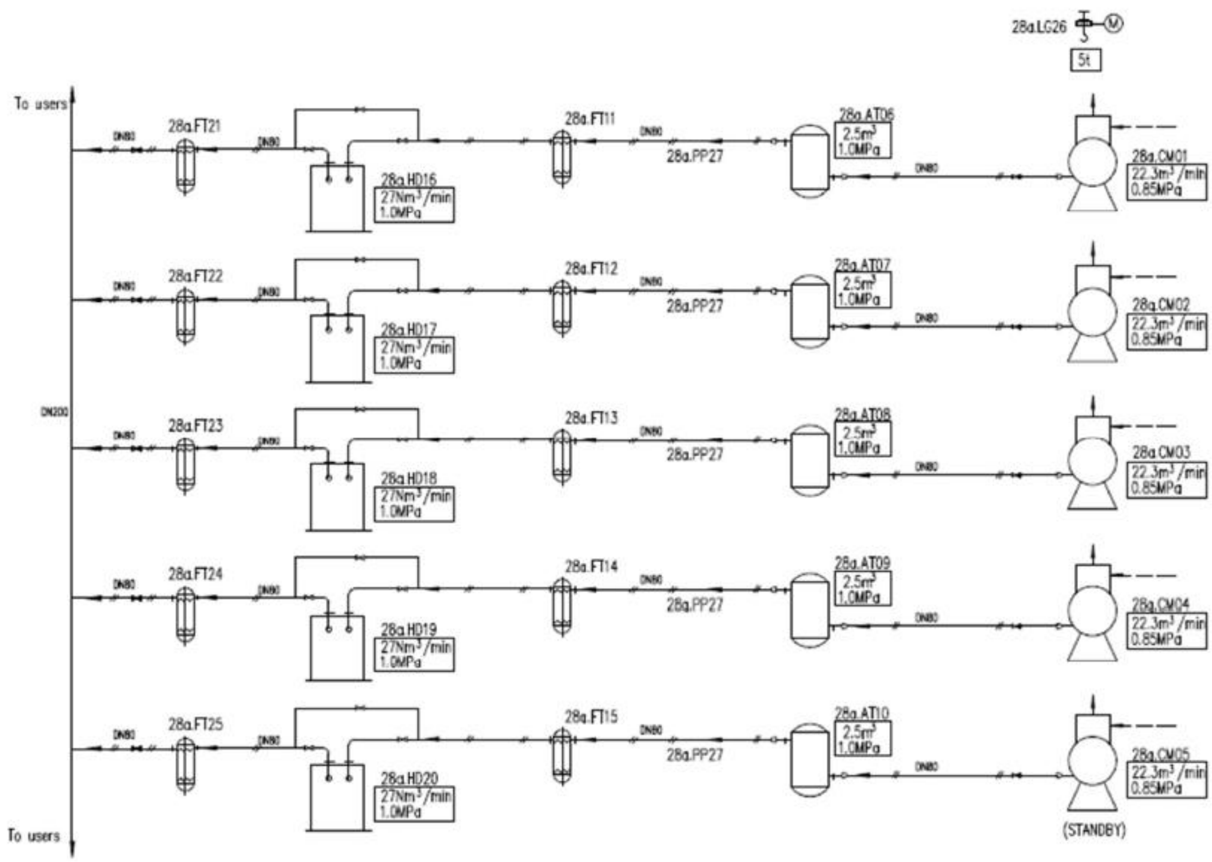



**Figure 14-27: Compressed Air System Single Line**

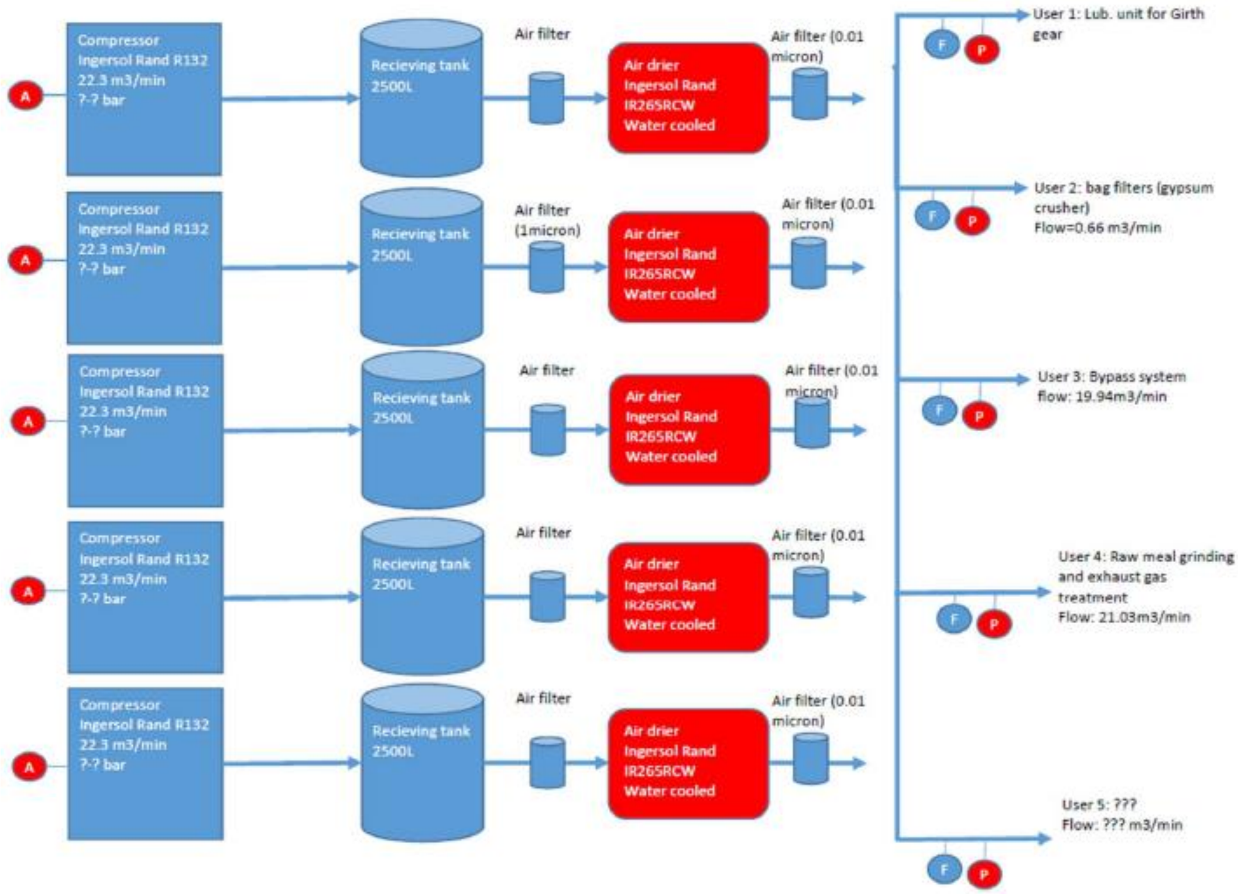



**Figure 14-28: Compressed air system single line diagram with measurement points indicated**

**Figure 14-29: Receiving tank and drain system**

**Characterization of the main compressed air users in the plant**

Based on the data collected from the company equipment list, and focusing only on the users within the clinker production that are served by the selected compressed air system, the main users of compressed air identified are indicated as follows:

- Lubrication of girth gear: 0.2 $m^3$/min
- Bypass system: 19.94 $m^3$/min
- Raw meal grinding and exhaust gas treatment: 21.03 $m^3$/min
- App. 15 bag filter (avg. 0.3 $m^3$/min each): 4.5 $m^3$/min

Thus, the total identified air demand is around 45.67 $Nm^3$/min. The working pressure at the users could be lower than 5.5 bar, as this was the pressure at the compressors discharge and no complaints were noted from the production.

### c. Baseline

The electricity consumption of the compressor at Nahda Cement Company was monitored for 10 days' measurement session using ACR3 data loggers with 500 Amps clamp meters. The five compressors were operational for the whole measurement period.

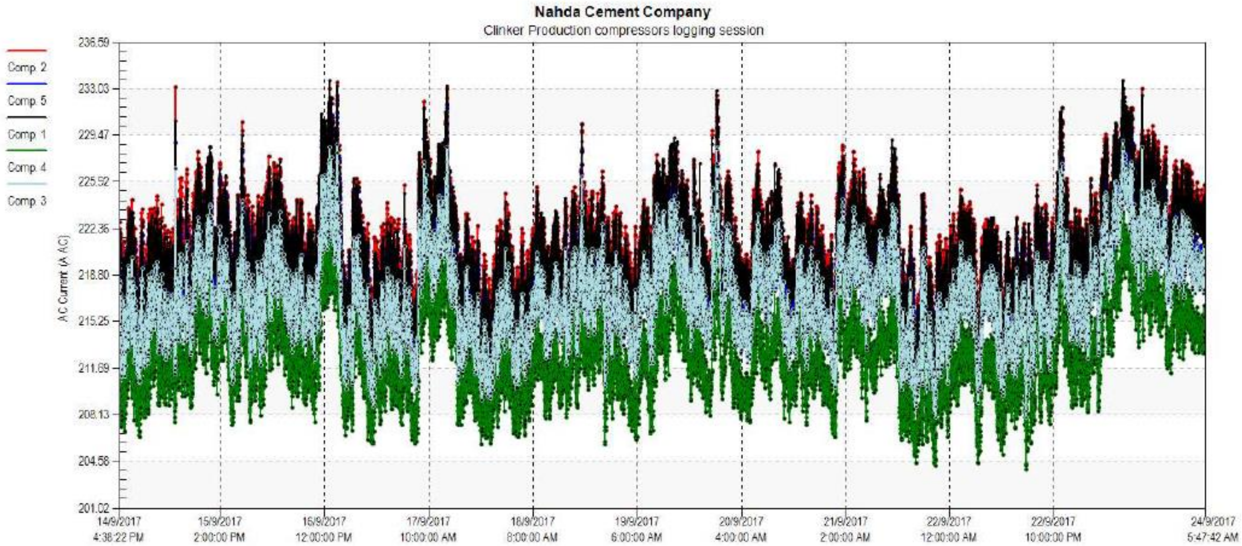



**Figure 14-30: Amperage profile of the five operating compressors during the assessment period**

The baseline electricity consumption was calculated to be 5,990,015 kWh/y at an electricity cost of 4,531,428 EGP/y.

## d. Key Findings during the Assessment

**Compressor Control**

The hour meter on the panels of each of the compressors indicated the following:

**Table 14-11: Compressors Running hours and pressure setting**

| S | Compressor | Run Hours | Load Hours | Pressure (bar) | Percentage of Loading hours | Setting (bar) Load / Unload |
|---|---|---|---|---|---|---|
| 1 | Compressor 1 | 41'956 | 40'464 | 5.8 | 96.4 | 5.9 / 6.9 |
| 2 | Compressor 2 | 41'439 | 39'223 | 5.6 | 94.6 | 6.0 / 7.0 |
| 3 | Compressor 3 | 44'278 | 43'255 | 5.6 | 97.7 | 6.1 / 7.1 |
| 4 | Compressor 4 | 40'263 | 37'292 | 5.5 | 92.6 | 6.5 / 7.5 |
| 5 | Compressor 5 | 46'298 | 43'036 | 5.5 | 99.4 | 6.5 / 7.5 |

The compressors are set to a cascade control. The first two compressors are continuously operational as long as the system pressure is below 7.5 bar (which should only be reached if production is stopped). The next compressor comes online if pressure falls to 6.1 bar, and will unload at 7.1 bar. The next compressor comes online at 6.0 bar and unloads at 7.0 bar. The last compressor will load and unload at 5.9 and 6.9 bar respectively. This control philosophy aims to have only one compressor cycling as the trim compressors. However, system pressure has to reach an artificially high excess pressure in order to unload the trim machine, thus the system pressure will fluctuate.

The compressors are all loaded from 92.6% to 99.4% of the time. The low pressure readings also indicate that none of the unload pressures are reached, and that more compressed air is demanded than can be supplied. However, since there are no complaints regarding low pressure, the compressor setpoints may have to be re-evaluated.

The 45.67 $m^3$/min theoretic compressed air demand requirement are currently supplied by the 5 compressors with a rotating capacity of 111.5 $m^3$/min.

**Distribution**

Initial observations indicate that the system storage volume is inadequate. The ratio of 0.22 $m^3$ system volume per $m^3$/min of trim compressor capacity, are comparatively low. The ratio should be closer to 1 $m^3/m^3$/min, although the exact volume requirements have to be properly engineered.

The piping in the compressor room, connecting the compressor with the storage tank, is a DN80 pipe. Calculating the air velocity shows an air velocity of 10.6 m/s, which is more than the recommended air velocity 5 m/s.

### e. Recommended Energy Efficiency Measures

The following opportunities to improve the performance of the compressed air system and save on energy consumption, are recommended:

- Measure the actual flow demand, and compare this to the rotating capacity of the operating compressors. Investigate the cause of the difference in these values in order to target to potentially switch off a compressor.
- Quantify the leakages and implement a leakage management program.
- Increase the diameter of the pipes in the compressor room in order to reduce air velocity and pressure drop.
- Replace current condensate traps with electronic sensing types.
- Capture and treat condensate discharge before disposal.
- Develop the system's pressure profile and optimise system pressure.
- Re-commission dryers, and investigate separating instrumentation quality air with general plant air quality.
- Investigate treating air at the point of use rather at the point of compressed air generation.
- Investigate, calculate and implement actual storage volume requirements.
- Repair or replace the faulty pressure gauges on the receiver tanks.
- Improve the control philosophy of the compressors, and consider an automated centralised master controller in order to stabilize system pressure and optimise compressor energy consumption.

## 14.5 El Marwa Food Industries: Juhayna Group

### a. Company Background

El Marwa Food Industries is a member of the leading industrial group Juhayna s.a.l, the largest dairy and juice product company in Egypt. It was found in 1997 with invested capital of 20 million Egyptian pounds.

The target clients of El Marwa food industries are both local and international clients. They produce the concentrates and puree for their sister companies, as well as for multinational brands (such as McDonalds).

The company is composed of two separate production buildings; El Marwa for the production of tropical fruit pulp, and Modern which produces citrus fruit concentrates. The two buildings are separated from each other in terms of the production seasons, the production process, the compressed air system and the electricity meters, while they both share the same steam system, with the boiler house located within El Marwa building.

The focus of the current report covers the compressed air system of El Marwa company.

## b. Compressed Air System Overview

Compressor accounts for around 19% of the total electricity consumption of the plant.

**Supply**

In order to satisfy the compressed air demand of the different users, a screw compressor is installed in the compressor room to generate the required compressed air. The screw compressor is an Atlas Copco GA45 Plus, followed by a wet receiver tank of 2000 L capacity, and a refrigerant drier of Atlas Copco (FX13).

The compressor is an air cooled, oil injected, screw type compressor. The rated Free Air Delivery (FAD) is 7.28m$^3$/min. The compressor motor is a 45kW motor. The rated input power is 56kW, at capacity and at full load operating pressure, and 13.7kW at no load conditions.

The main air pipe connecting the compressor and the receiver tank is a 1.5” diameter pipe. The compressor after cooler, and the receiver tank are equipped with an automatic drain system with a timed solenoid valve type (open for 3 seconds every 30 minutes), which directs the condensate water towards the industrial drain network (refer to Figure 14-31).

Figure 14-31: Receiver tank and drain system

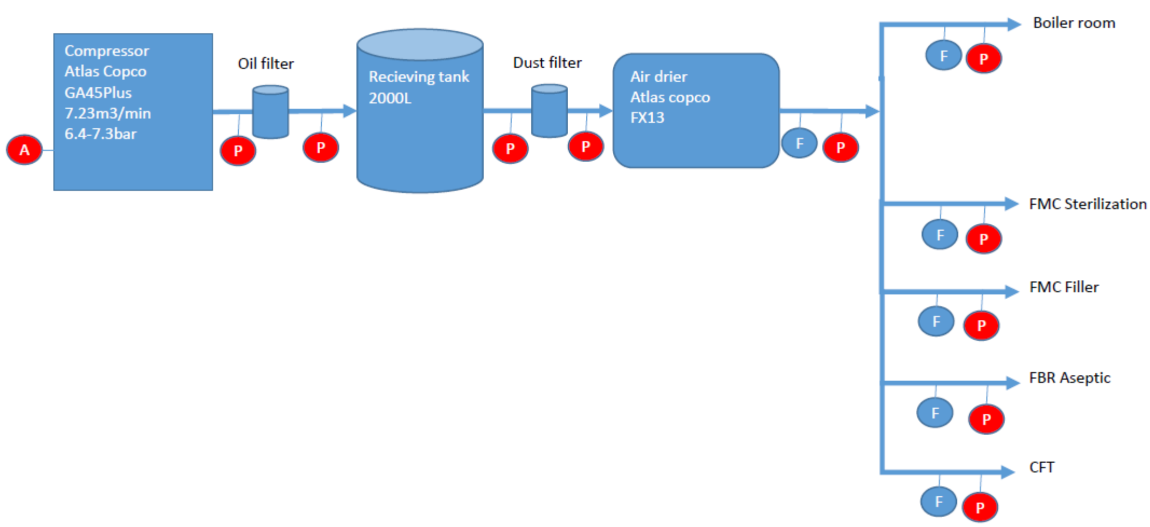



Figure 14-32: Single line diagram with measurement points indicated

**Demand**

Based on the data collected from the utility engineers of El Marwa company, the main users of compressed air identified by the company team are indicated as follows:

- **CFT line**: 54 $Nm^3/h$ (8 bar).

- **FBR line:** 4 $Nm^3/h$ (6 bar).
- **Aseptic tank:** 40 $Nm^3/h$ (6 bar).
- **FMC filler:** 6 $Nm^3/h$ (6-8 bar).
- **FMC sterilization:** No data available, not expected to exceed 2 $Nm^3/h$.
- **Boilers:** No data available, not expected to exceed 2 $Nm^3/h$.
- **Diaphragm pumps:** No data available, used occasionally only to pump Zabado (High density) syrup.

Thus, the total air demand is around 108 $Nm^3/h$ (1.8 $Nm^3/min$). The working pressure at the users should not be below 6 bar, according to the utility engineers.

**Distribution**

The pipe network, indicating the location of the compressor room and the main users is illustrated in Figure 14-33.

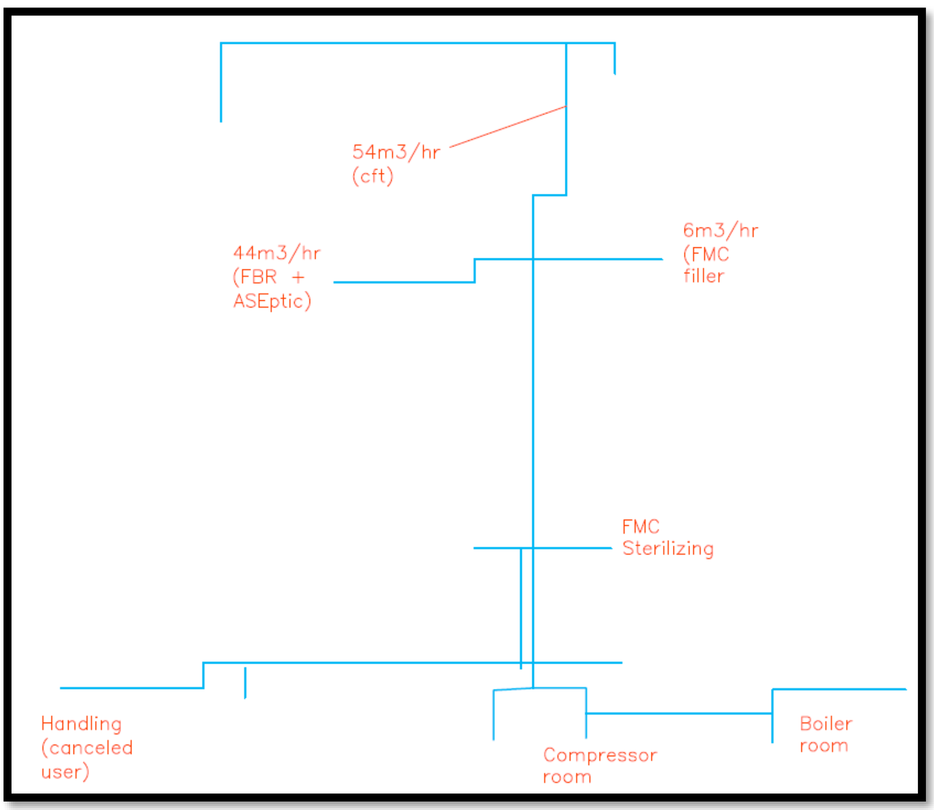



**Figure 14-33: Compressed air network**

## c. Baseline

The baseline of the compressor consumption according to the installed electricity submeter, is 1024 kWh/day or 373,760 kWh/y (equivalent to 659.4 EGP/day or 240,681 EGP/y), which was based on the base period of Nov. 30th – Dec. 31st 2016.

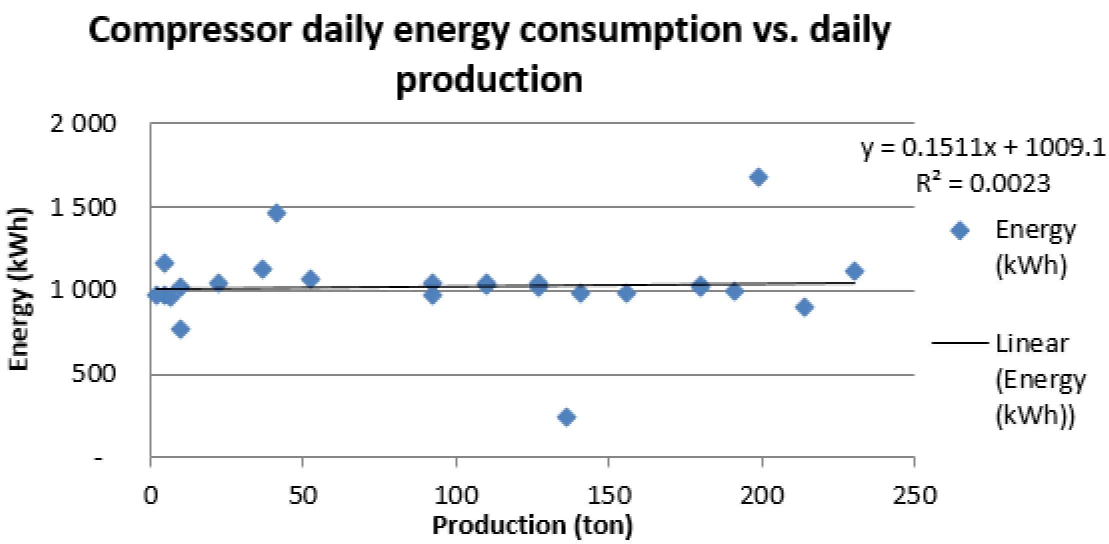



**Figure 14-34: Compressor daily energy consumption vs. daily production**

Based on the hour counter readings on the compressor in terms of loading and running times, it is calculated that the expected specific power is 14.65 kW/$m^3$/min. The compressor's rated specific power is 6.96 kW/$m^3$/min. The cost for each $m^3$ of compressed air is 0.157 EGP/$m^3$.

## d. Key Findings during the Assessment

**Compressor Control**

The compressor is loaded for 40.5% of the time according to the hour counter readings on the compressor. Measurements also indicated rapid cycling of the compressor, with a cycle of 50 seconds each - about 20 seconds for loading, and 30 seconds for unloading. Thus the load/running is 20/50 = 40%, which is identical to the same conclusion based on the hour meter. The receiving air tank has a capacity of 2000 Liters, serving a peak 7.28 $m^3$/min compressor capacity. This is equivalent to 0.27 $m^3$ per $m^3$/min, which is below the rule of thumb recommendation of 1$m^3$ per $m^3$/min. Based on the estimation of the average kW of the compressor package versus the average capacity with the storage capacity, indicated in Figure 14-35, the compressor is expected to be running at 75% of its power input for the given configuration and conditions. This 75% is equivalent to $75\% * 56kW * \frac{24hrs}{day} = 1008 \frac{kWh}{day}$. This figure is close to the baseline consumption of 1024 kWh/day discussed earlier.

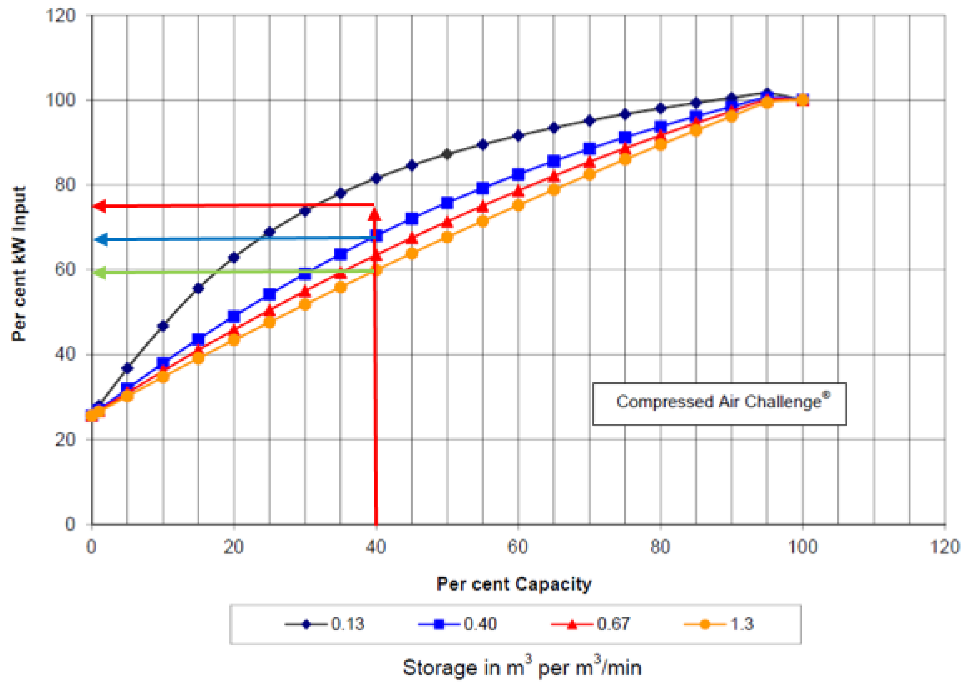



**Figure 14-35: Expected kW input for the given load/running conditions**

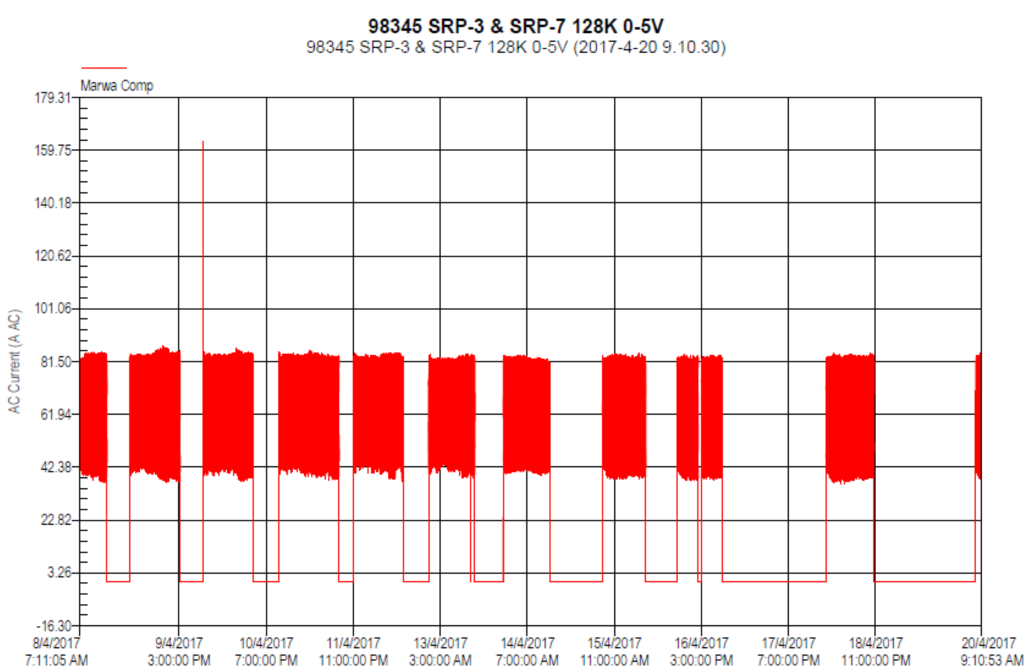



**Figure 14-36: Compressor's measured amperage profile during the assessment**

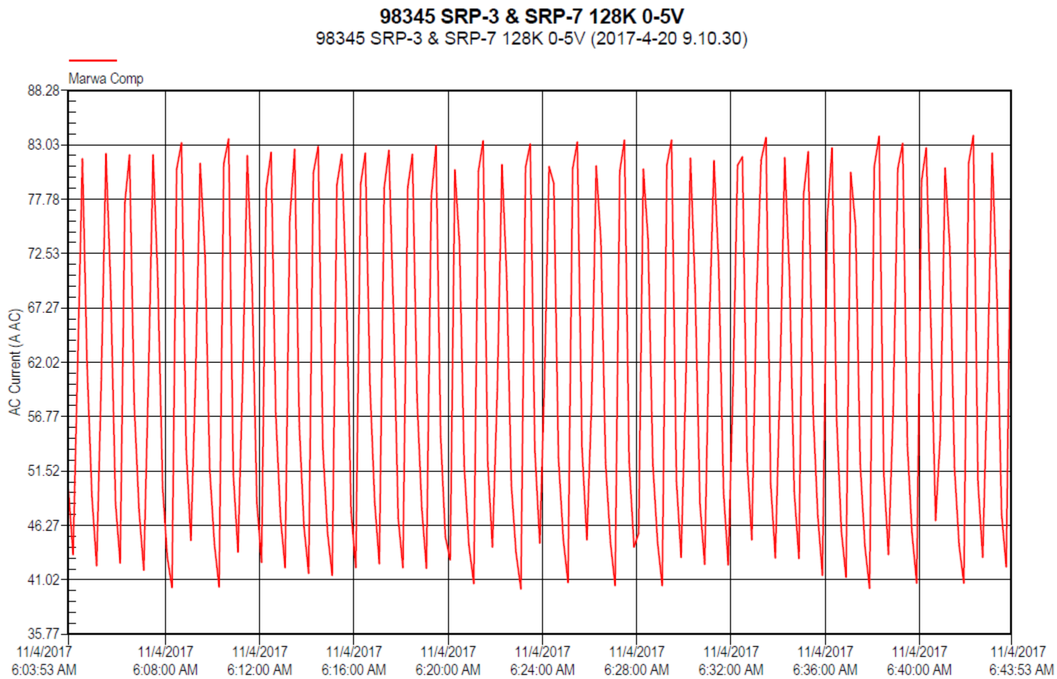



**Figure 14-37: Compressor's rapid loading/unloading cycle during a 40 minute window**

**Supply Piping**

The main header pipe, connecting the compressor with the storage tank is a 1.5" inner diameter pipe. This pipe, as indicated by the company team is in accordance to the fitting size of the compressor. Calculating the air velocity in that pipe for a 7.28 m$^3$/min air flow rate, at a pressure of 7.3 bar results in an air velocity of 12.8 m/s. This velocity is much higher than the recommended air velocity of not more than 5 m/s in the compressor room. Thus currently, the pressure drop in the main header pipe exceeds the recommended levels.

This option, however, in discussion with the company team, was not accepted due to the connection to the compressor which was built in from the manufacturer for the 1.5" pipe. Moreover, as the pressure measurements were not conducted, the actual pressure drop is not known for the current condition, and the savings cannot be estimated. The recommendation is to attempt to measure the pressure drop through the existing pipe before further pursuing this option.

**Leaks**

Air leak test detection was conducted through a walkthrough in the plant with the ultrasonic leak detector to identify leak points. A receiver pump-up leakage quantification test was also done during a period of no production. The calculations are as follow:

$$V_L = \frac{V_R(P_i - P_f)}{T} = \frac{2000(L)*(7(bar) - 5(bar))}{3\ (min)} = 1333\,\frac{L}{min} = 1.33\ m3/min$$

This leak is approximately equivalent to

$$\frac{1.33}{7.29(compressor\ nominal\ discharge) * 40\%(load\ percent)}$$

$$= 45\%\ of\ the\ compressor\ load$$

**Connecting the compressed air network between Marwa and Modern**

Consideration was given to connecting the compressed air systems between the two plants, however, through discussion with the company team, this option appeared no to be practical for the following reasons:

- The pipe line connecting the two plants should go through an existing utilities tunnel under the street between the two plants. This route would have lots of bends and connections resulting in an increase in pressure drop. Moreover, the pipe length would be more than 400 meters (including the bends and elbows connections).
- The two plants don't operate simultaneously except for approximately one month per year.
- Regarding the breakdown of a compressor, currently policy is to either relocate the other plants' compressor to keep the plant running, or to contact the supplier to get another compressor to sustain the production until they fix the out of service compressor.

## e. Recommended Energy Efficiency Measures

**EEM#1: Eliminate/minimize the air leaks**

The El Marwa maintenance team, after attending the expert training decided on going for an aggressive leak detection and repair program. They determined the leak locations, and most of the leaks were repaired.

That action, and based on the consumption figures from the submeter data, resulted in a reduction in the compressor consumption from the baseline consumption by about 25% (the average daily consumption in March reduced to 750kWh/day, from 1024kWh/day baseload in December as indicated below:

**Table 14-12: Compressor average daily consumption after leakage program was initiated**

| Month | Average daily consumption (kWh/day) |
|---|---|
| **Dec.** | 1,024 |
| **Jan** | 1,000 |
| **Feb** | 913 |
| **Mar** | 748 |

| | |
|---|---|
| Investment required (EGP) | Zero |
| Expected Savings (kWh/year) | 250kWh/day*7month*30day/month= 52.8 MWh/year |
| Expected financial savings (EGP/year) | 34000 |
| Payback period (Years) | Zero |

**EEM#2: Reduce the pressure setting for the compressor**

The compressor setting is to load at 6.4 bar, and unload at 7.3 bar. The expected pressure drop through the system is postulated to be around 0.15 bar at the compressor room, and 0.05 bar within the pipe network. While the users' specifications indicate a pressure requirement of 6 bar, reducing the set points to 6.2 bar - 7.1 bar should be safe for operation.

Further reduction in the pressure setting should be investigated and the users' specifications shall be challenged. The main use of compressed air is mainly for valves, which do not necessarily need 6.0 bar pressure. The recommendation for this opportunity is to implement it in small steps, via decreasing the pressure by 0.1 bar in each step and assure that no one from the operation and maintenance team complains from low pressure, then decrease another 0.1 bar.

This opportunity is categorized as a no cost opportunity, since it only needs setting the pressure for the compressor. The savings for this intervention is quantified at 1885 kWh/year.

| | |
|---|---|
| Investment required (EGP) | Zero |
| Expected Savings (kWh/year) | 1885 |
| Expected financial savings (EGP/year) | 1214 |
| Payback period (Years) | Zero |

**EEM#3: Increase the storage volume**

The air receiver volume is very small as stated earlier. Through discussion with the company team, they have indicated that they already have an unused storage tank of $1m^3$ of volume. The consideration is to install the $1m^3$ tank after the compressor and before the drier, and switch the existing $2m^3$ tank to be after the drier. This arrangement would allow for additional storage volume, and utilizing both the dry and wet storage approaches.

Increasing the storage volume would result in longer cycling, possibly leading to complete oil separation during the unload time. Approximate estimates indicate that the compressor power would decrease to around 70% rather than 75% of full load capacity. The expected saving would be equivalent to 24,528 kWh/y.

| | |
|---|---|
| Investment required (EGP) | 4,000 |
| Expected Savings (kWh/year) | 24,528 |
| Expected financial savings (EGP/year) | 15,796 |
| Payback period (Years) | 0.4 |

**Summary of Recommended Opportunities**

The compressed air system at ElMarwa company could be improved with several low cost actions. The saving achieved from implementing the low cost actions reached over 25% of the compressor energy consumption without any impact to the productivity or quality of the production. More savings are expected upon finalizing the implementation of actions.

**Table 14-13: Energy Savings Impact**

| Energy Savings Impact (annual) | |
|---|---|
| Total Electricity Savings [kWh] | 79,213 |
| Carbon Dioxide Reduction [Tons] (0.0048Tons CO2/kWh) | 38 |
| Total Cost Savings [EGP] | 51,000 |
| Total Investment Cost [EGP] | 4,000 |

**Table 14-14: Energy Savings Summary**

| Energy Savings Summary | | | | |
|---|---|---|---|---|
| | kWh Saving | Cost Saving | Investment | Payback |
| | [kWh] | [EGP] | [EGP] | [years] |
| Saving Opportunity 1 - Eliminate/minimize the air leaks | 52800 | 34000 | - | Immediate |
| Savings Opportunity 2 - Reduce the pressure setting for the compressor | 1885 | 1214 | - | Immediate |
| Savings Opportunity 3 - Increase the storage volume | 24528 | 15796 | 4000 | 0.4 |

**Next Action Steps**

In addition to the aforementioned options, more saving could be achieved through:

- Further reduction to air leaks (currently the company still experience a leak level of 45% of the compressor load).
- Investigate further increase of the storage volume, which requires the purchase of a new storage tank
- Increase the main pipe diameter; this would assist in the reduction of pressure loss in the main pipe, and thus facilitate further reduction to the set points
- Challenge the demand side pressure setting
- Install regulators for the diaphragm pumps as well as any other uncontrolled users (floor washing pumps)
- Replicate the same study to the sister company; Modern for Tropical fruit.

## 14.6 Juhayna Egyptian dairy products and juices

### a. Company Background

Juhayna Food Industries was established in 1983, an Egyptian company specializing in the production, processing and packaging of a variety of dairy products and juices as well as culinary products. The company has succeeded in taking the lead in dairy and juice markets in Egypt and expanding its markets in the Middle East. As part of Juhayna's endeavour to increase production capacity in line with growing demand for Juhayna products, the company acquired Egyptian dairy products and juices factory in 2005. With a total of 300 workers, including technicians, engineers, and administrators, the factory utilizes the most advanced technologies to produce milk and white cheese. Juhayna, through its production at Egyptian dairy products and juices factory, was the third globally ranked company to use the new TBA-Edge packaging technology, which guarantees the highest quality of milk products.

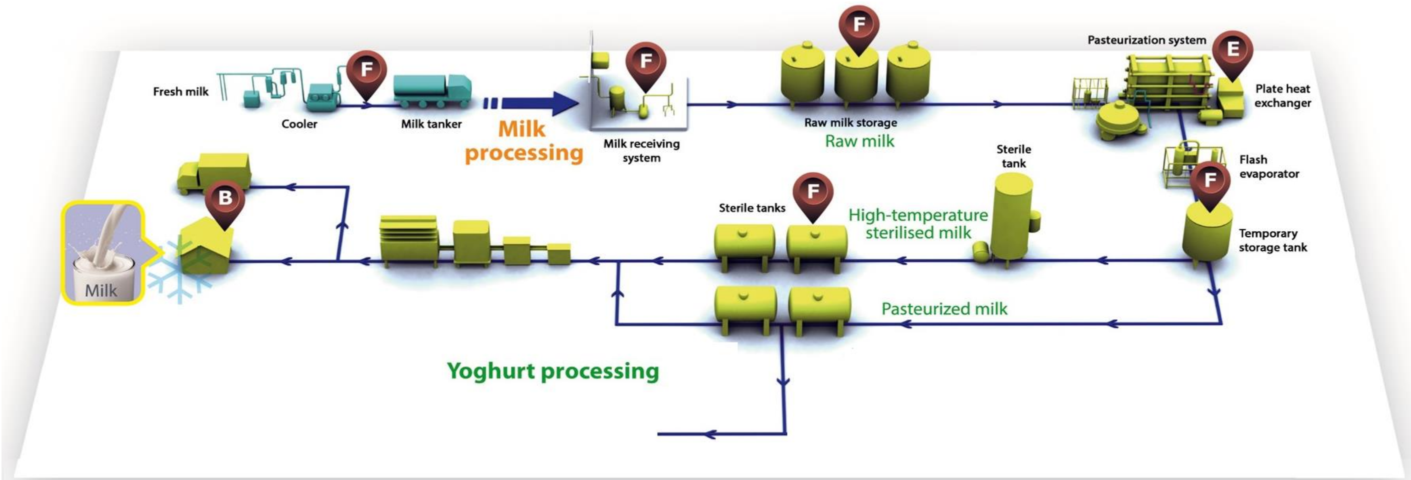



**Figure 14-38: Milk production Process Diagram**

## b. Compressed Air System Overview

The compressed air plant comprises of two Atlas Copco (GA55), single-stage, air-cooled, lubricant injected rotary screw compressors, one Atlas Copco (ZT250 VSD-FF), two stage, oil free compressor, and another Atlas Copco (ZT145), two stage, oil free compressor. The ZT250 VSD-FF compressor is controlled by means of a Variable Speed Drive (VSD), while the other ZT145 compressor has load / unload control. The two GA55 compressors are not operational.

**Table 14-15: Existing plant air compressors**

| Compressor Manufacturer | Model | Capacity Control Method | Power kW (pkg. = package) | Rated Capacity | Operating Pressure |
|---|---|---|---|---|---|

| | | | | | |
|---|---|---|---|---|---|
| **Atlas Copco** | ZT250 VSD-FF (oil-free)<br>Mfg. year. 2012 | VSD | 250 kW nom. shaft<br>273 kW total pkg. | 37.32 $m^3$/min | 10.4 bar(g) rated<br>8.3 bar(g) set point |
| **Atlas Copco** | ZT145 (oil-free)<br>Mfg. year. 2008 | Load/Unload | 145 kW nom. shaft<br>157 kW total pkg. | 19.88 $m^3$/min | 10 bar(g) rated<br>8.4 bar(g) operating |
| **Atlas Copco** | GA55 (oil-injected) | | 55 kW nom. shaft<br>63.5 kW total pkg. | 9.26 $m^3$/min | 9.1 bar(g) rated |
| **Atlas Copco** | GA55 (oil-injected) | | 55 kW nom. shaft<br>63.5 kW total pkg. | 9.26 $m^3$/min | 9.1 bar(g) rated |

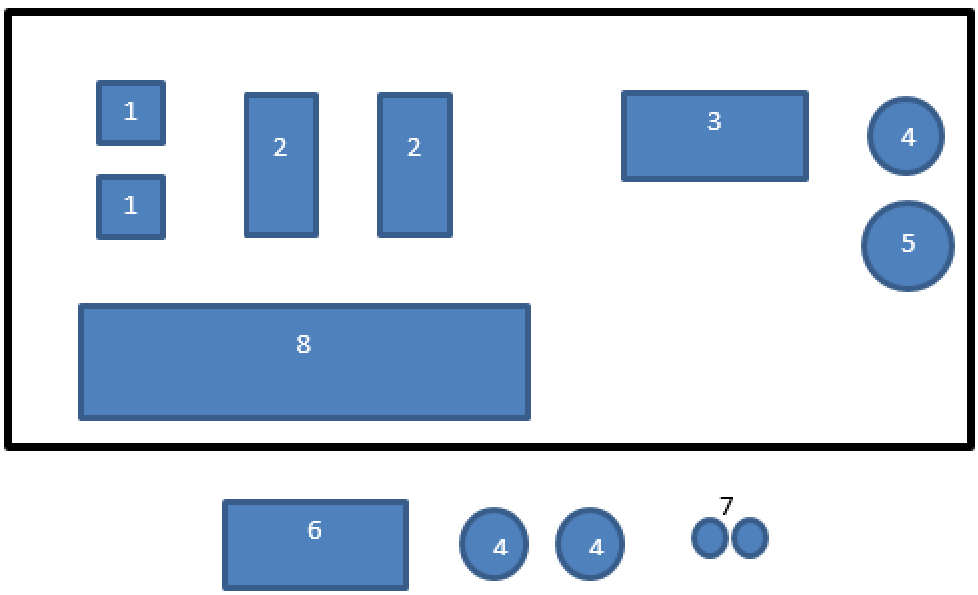



**Figure 14-39: Plant compressed air system block diagram**

1. Refrigerant dryer
2. GA55
3. ZT250VSD-FF
4. 2$m^3$ Receiver
5. 3$m^3$ Receiver
6. ZT145
7. Desiccant dryer
8. Refrigeration cycle screw compressor (not related to the compressed air system)

## c. Baseline

The total Egyptian dairy products and juices plants' electrical energy consumption is 11,879,399 kWh per annum, according to the 2016/2017 electricity bill data. The total Egyptian dairy products and juices plant electrical energy costs are EGP 7,958,639 per annum (at the 2016/2017 tariffs). Thus the total electrical energy rate during the previous year was 0.52 EGP/kWh. When adding the 47.5% tariff increase for 2017/18, the electrical energy rate for 2017/2018 is 0.767 EGP/kWh, which is the value used for calculations in this report.

The compressed air electric energy (including the dryer) annual usage is 1,398,144 kWh, costing Egyptian dairy products and juices plant EGP 986,220 per year, generating a total of around 21 million $m^3$ / year of compressed air. The system specific power is 0.088 kW/$m^3$. Each 11 $m^3$ of compressed air is currently consuming 1 kWh. (According to the measurements and the data provided by the company). Compressed air is utilising approximately 13% of Egyptian dairy products and juices plant's annual electric energy.

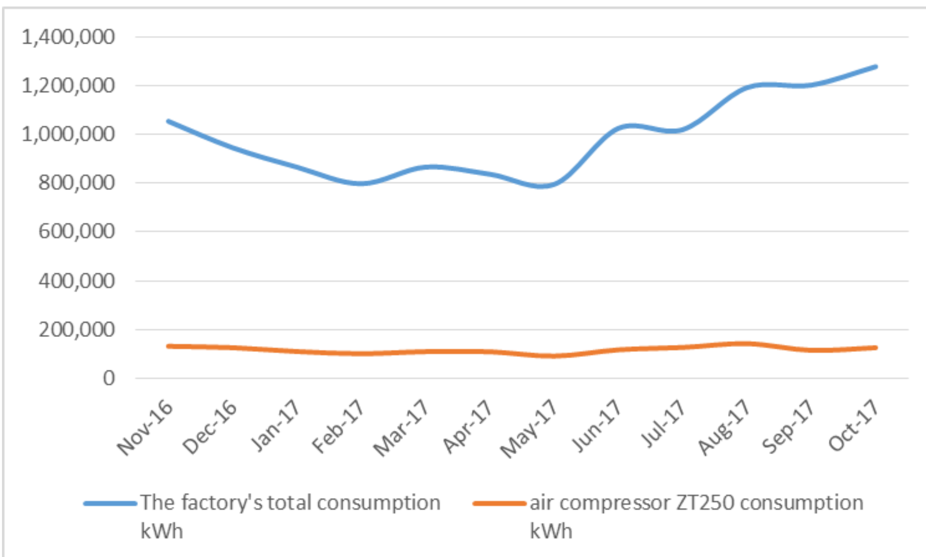



**Figure 14-40: Compressed air consumption vs. Total Electricity Consumption**

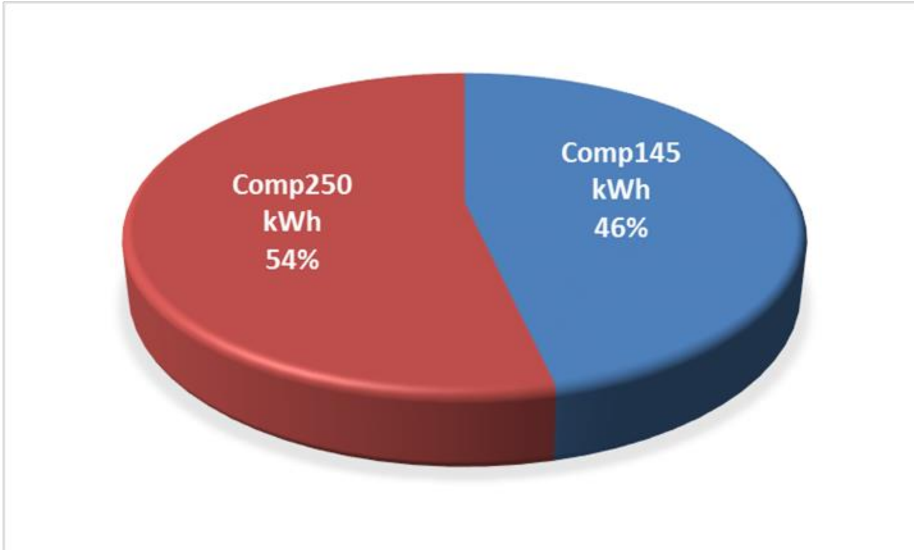



**Figure 14-41: Projected compressor annual electricity consumption breakdown during the assessment**

During the assessment both compressors were operational. According to plant personnel, the ZT250 is usually the only operational compressor, while the ZT145 is in standby mode (unload), which was not the case during the assessment due to ZT250 failures.

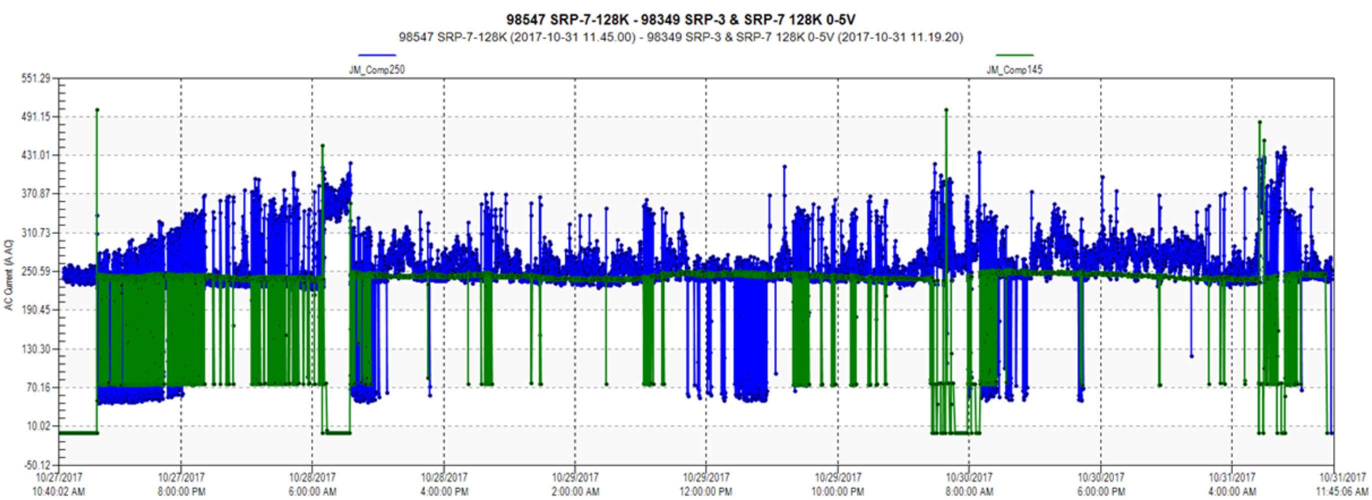



**Figure 14-42: The two operating compressors data loggers' measurements**

## d. Key Findings during the Assessment

**System Pressure Profile**

During the assessment, the compressed air system was operating above the minimum necessary pressure. According to information from the company, the minimum pressure requirement is 6.5 bar which, in discussions with plant personnel, may realise a potential pressure reduction of 0.5-0.8 bar.

**Artificial Demand**

In the milk processing area, the pressure demand is 6 bar and in the packaging area it is 6.5 bar. The majority of the regulators checked was adjusted to a pressure higher than the demand (refer to images below), which adds to artificial demand.

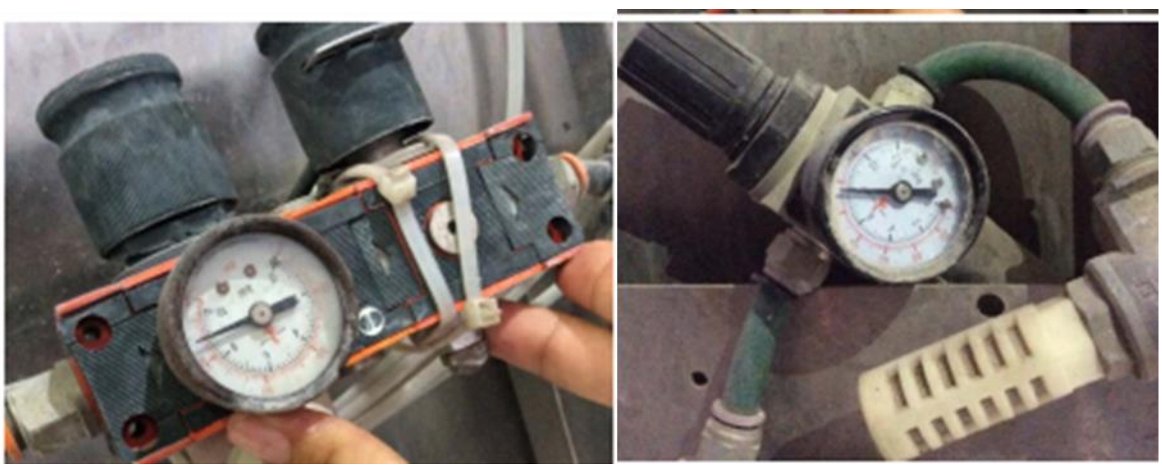


**Figure 14-43: Pressure regulators**

**Storage Volume**

The current situation as depicted below, is the compressed air output from the compressor ZT145 accessing the first storage tank from the lower port as shown by the input red arrow. The two tanks are connected through their higher port and the output of the ZT145 storage from the lower port of the second tank. This is not recommended as the lower part of the tank may result in condensate carry-over, especially due to the desiccant dryer of this compressor not being operational. As shown, the recommended storage connection is to keep the input of the storage from the lower port of the first tank and changing the output of the second tank from the higher port.

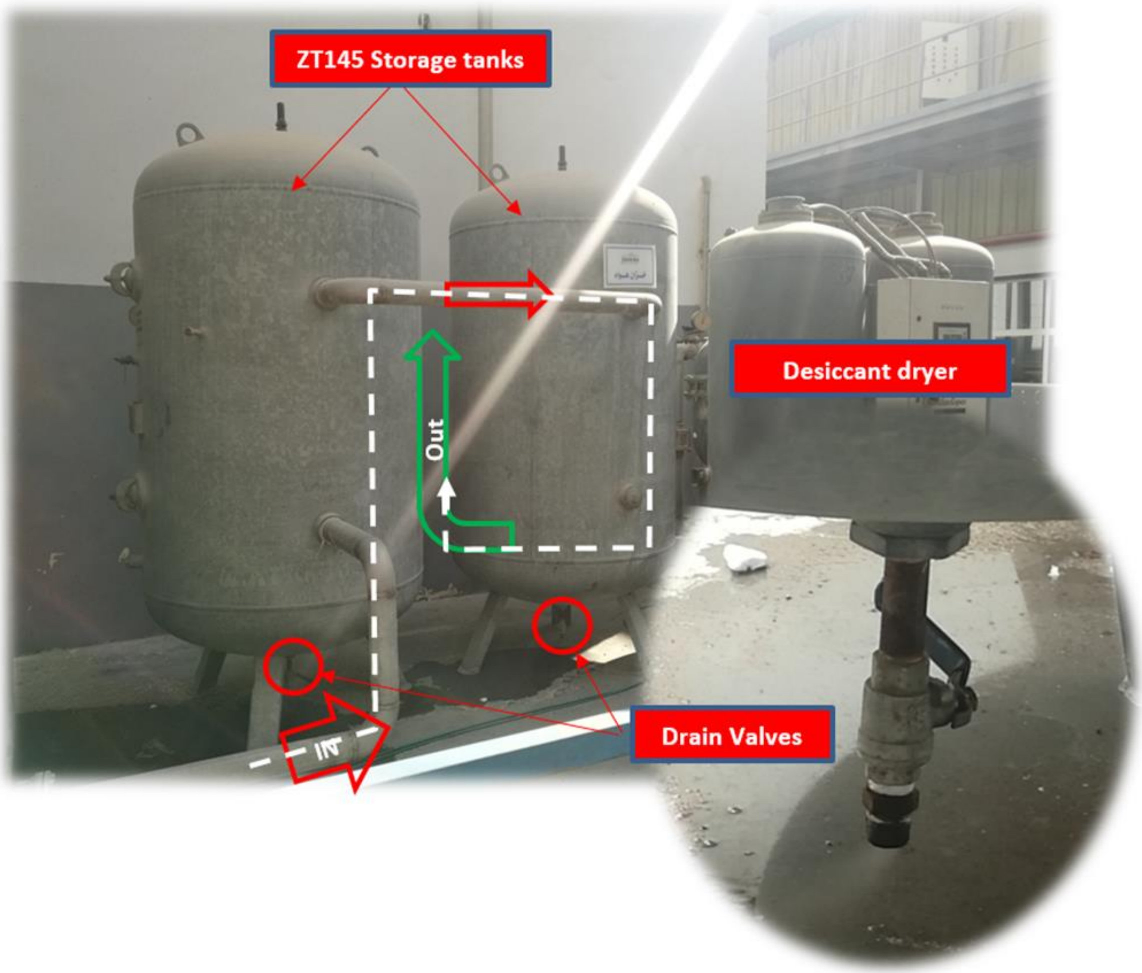



**Figure 14-44: ZT145 current storage and treatment system**

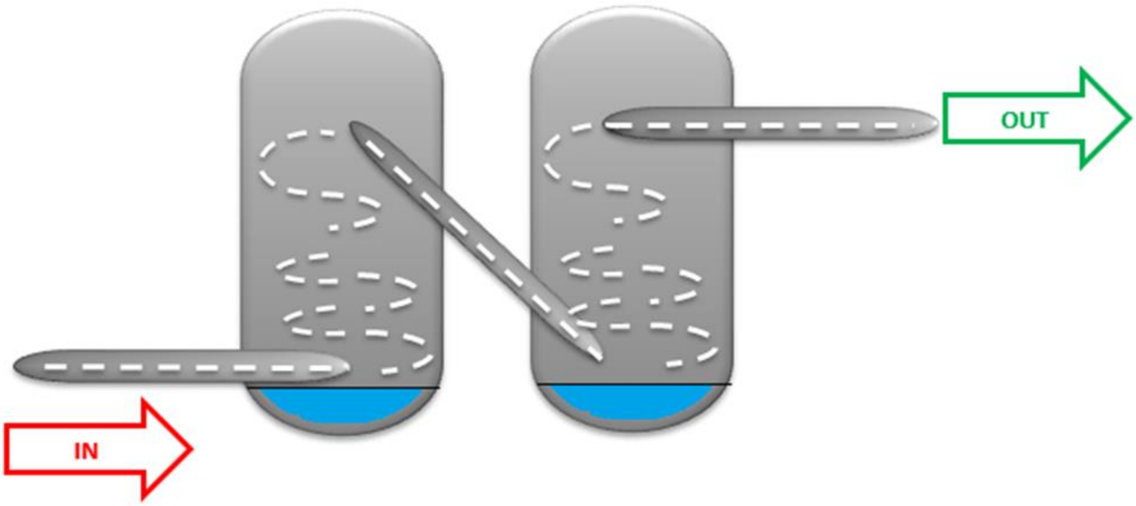



**Figure 14-45: ZT145 storage tanks recommended interconnection**

From the storage images above, the drains of the storage receivers were open and continuously leaking significant amounts of compressed air. No-loss automatic drain valves are recommended, and can often payback in a matter of weeks if they replace malfunctioning drains that leak. They are controlled either by pneumatic/float or electronic/sensor.

**Pressure Flow Control**

Due to the rapid cycling of the ZT145, it's recommended to install a pressure flow controller in order to control the release of air, to stabilize the air pressure delivered into the main header leaving the compressor room. Flow supply from the pressure flow control calve is continuously adjusted to correct the deviations from set point.

**Heat Recovery**

Recovering heat for boiler combustion air pre-heating using the cooling air of the should be considered. The company should check with the burner manufacture that the present fans can tolerate the raised air temperature.

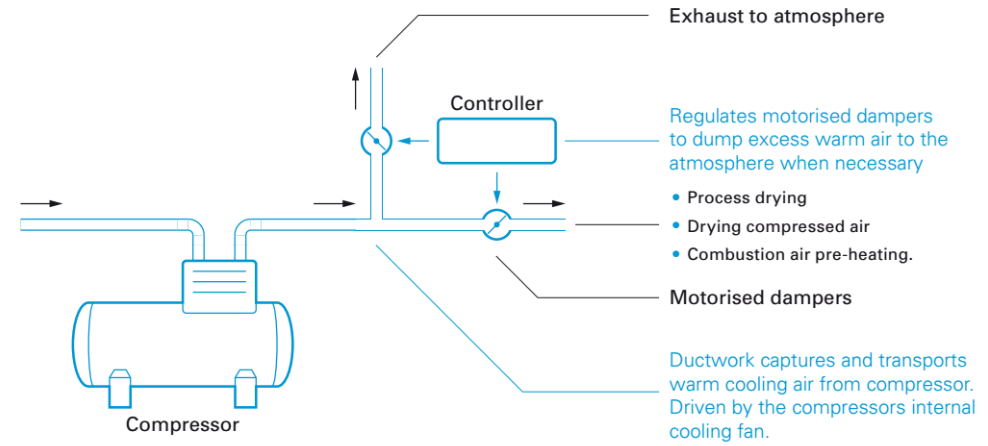



**Figure 14-46: Heat recovery diagram**

Other indirect heat recovery considerations are for domestic hot water using heat exchanger for heating water for workers' bathrooms, or for the pre-heating of boiler make-up water.

## e. Recommended Energy Efficiency Measures

***EEM#1: Adjust the regulators for the minimum required pressure.***

In the milk processing area, the pressure demand is 6 bar and in the packaging area it is 6.5 bar. The majority of the regulators checked was adjusted to a pressure higher than the demand, which will add to artificial demand. Expected energy savings from adjusting the regulators (pressure reduction of 0.5 bar) is 40,131 kWh/year and an annual cost savings of EGP 30,780/year.

***EEM#*2: Modify the interconnection piping of the storage.**

The recommended storage connection is to keep the input of the storage from the lower port of the first tank and changing the output of the second tank from the higher port of the second tank. The interconnection pipe has to be changed to connect the higher port of the first tank to the lower port of the second tank.

***EEM#*3: Replace the drain valves with no-loss drains.**

No-loss automatic drain valves are recommended. The estimated losses 0.3455 kW. This may realise an expected annual energy saving of 9,675 kWh/year across the four tanks. The expected annual cost saving is 7,420 EGP/year.

***EEM#*4: Install Pressure-Flow Control (Intermediate Control) and run compressors in cascade control.**

it's recommended to install flow controller to use the controlled release of air already in storage to stabilize the air pressure delivered into the main piping header leaving the compressor room, and pressure at control valve outlet is sensed and air flow is continuously adjusted to correct the deviations from set point, and the storage should cover the starting time of ZT145 if needed.

According to the logger of the ZT145 compressor, the unload percentage is 49.4 %. Saved energy from changing the ZT145 to start-stop control instead of load-unload is 154,767 kWh/year at a cost saving of 118,706 EGP/year.

***EEM#*5: Boilers' combustion air pre-heating by heat recovery of compressed air cooling.**

Combustion air pre-heating using the cooling air of ZT250 for pre-heating boiler combustion air, but the company should check with the burner manufacture that the present fans can tolerate the raised air temperature.

The annual recovered available energy is approximately 492,359 kWh/year, which is equivalent to 142,800 EGP/year, based on 4.75 USD/GJ of natural gas.

**Summary table of opportunities**

A summary of the quantified opportunities are given in **Error! Reference source not found.**. The compressed air recommendations need to be implemented as a group of initiatives to realise the stated savings, and not be implemented as individual measures due to the interactive effects of a compressed air system.

**Table 14-16: Recommended Potential CASO Energy Efficiency Measures Summary for Implementation**

| *EEM* | *Energy Saved (kWh/year)* | *$CO_2$ emissions reductions (kg $CO_2$/Year) (0.579 kg $CO_2$/kWh)* | *Cost Savings (EGP/Year)* | *Investment Cost (EGP)* | *Payback Period (years)* | *Return on Investment (%)* |
|---|---|---|---|---|---|---|
| Adjust the regulators for the minimum required pressure. | 40,131 | 23,236 | 30,780 | No Investment | Immediate | Immediate |
| Modify the interconnection piping of the storage. | Reduce dew point temperature, improve air quality, and slightly reduce energy consumption of the dryer. | | | | | |
| Replace the drain valves with no-loss drains. | 9,675 | 5,601 | 7,420 | 36,000 | 4.85 | 20.6 |
| Install Pressure-Flow Control (Intermediate Control) and run compressors in cascade control. | 154,767 | 89,610 | 118,706 | 110,000 | 0.92 | 108 |
| Boilers' combustion air pre-heating by heat recovery of compressed air cooling. | 492,359 | 82,163 | 142,800 | Need survey and depends on the distance between ZT250 and boilers | | |

## 14.7 Evergrow

### a. Company Background

Evergrow was established in 2006. It is engaged in manufacturing & marketing of specialty fertilizers. The first industrial complex was established in Abo Rawash Industrial Zone, 6th October City, Giza, Egypt on 40,000 $m^2$ as the first Egyptian factory specializing in the manufacturing of all kinds of fully soluble and granular Potassium sulphate, Mono-Ammonium Phosphate (MAP), Mono Potassium Phosphate (MKP), Urea Phosphate (crystal and liquid-UP), different formulas of solid, liquid and suspension NPK, Calcium nitrate (crystal and liquid), Copper sulphate, Calcium chloride, and hydrochloric acid.

### b. Compressed Air System Overview

The compressed air system consists of two separate compressed air rooms that feed the factory in parallel from both sides of the factory. The two rooms are connected together with a valve, which is normally closed, and opened manually if needed.

**Furnace zone compressed air system (room 1)**

The room has two similar fixed speed Ingersoll-Rand MH75 screw air compressors. The rated power for each compressor is 75 kW. The two compressors supply the air to the demand side without passing through the drier. There is poor ventilation in the room, which lead to the average ambient room temperature inside the room being 39.5$^{o}$C. The 2” supply pipes go underground for 130 meters to the storage tank, which is at approximately 5 barg.

In the two compressors, the water separator filter valves are always partially open to drain the water. The receiver tank blowdown valve is partially open to remove the water and now it is immovable.

**Figure 14-47: Furnace zone compressed air system**

**Liquid zone compressed air system (room 2)**

The room has one fixed speed Ingersoll-Rand UP5-30-10 screw air compressor and one dryer. The rated air compressor power is 30 kW. The receiver tank blowdown valve is partially open to remove the water. The liquid zone compressed air system used to feed factory(3), liquid factory and solution factory.

**Figure 14-48: Liquid zone compressed air system room 2**

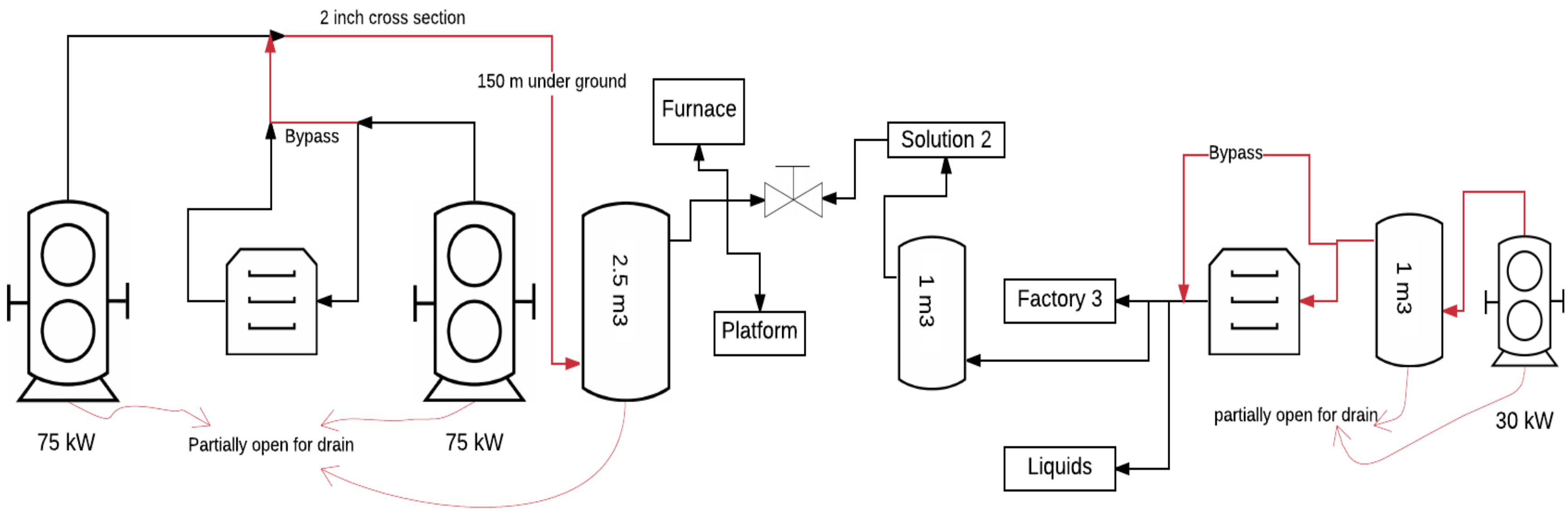



**Figure 14-49: Compressed air system block diagram**

## c. Baseline

Based on the measurements, the specific energy consumption is 0.115 kWh/m$^3$ at the operating pressure of between 6 and 7 barg.

| | AC1 | AC2 |
|---|---|---|
| Working hours | 5,596 hrs. | 5,636 hrs. |
| Production | 3,941,630 m$^3$/year | 2,382,170 m$^3$/year |
| Production | 6,323,800 m$^3$/year | |
| Average consumption | 416,324 kWh | 309,777 kWh |
| Total energy consumption | 726,101 kWh | |
| SEC | 0.115 kWh/m$^3$ | |
| Compressed air cost | 556,193 EGP | |

## d. Key Findings during the Assessment

**Furnace zone compressed air system (room 1)**

From the measured data during the assessment, the annual projected annual working hours for compressor 1 and 2 are 5,596 and 5,636 hours respectively.

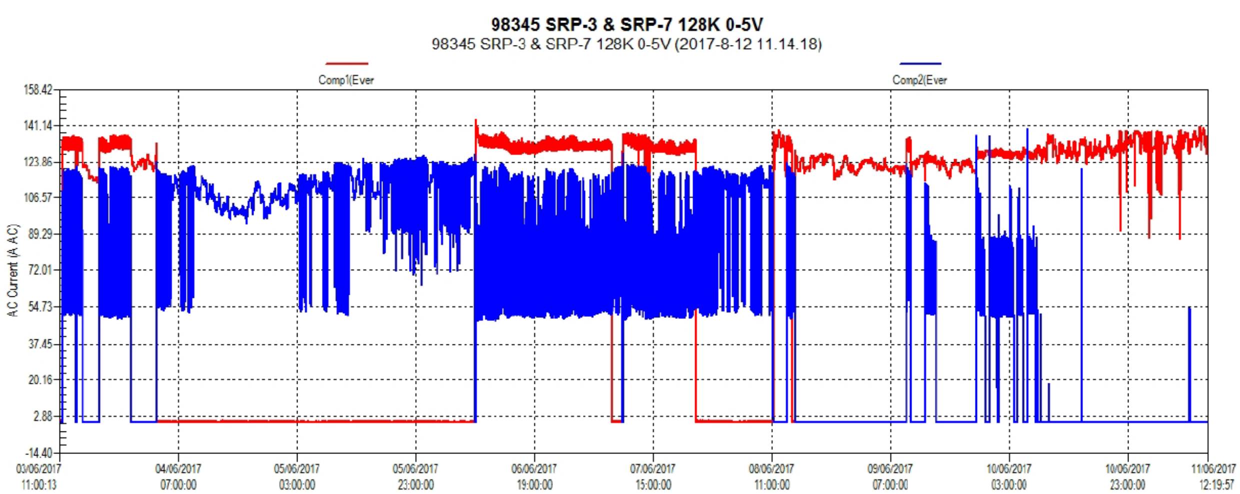



**Figure 14-50: Compressor 1 and 2 measured ampere profiles**

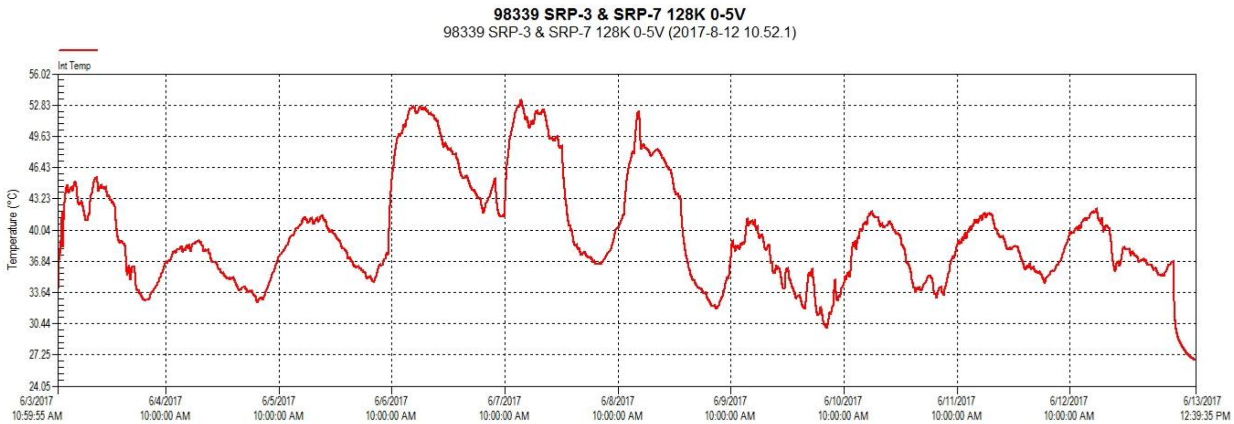



**Figure 14-51: The average compressor room temperature measured was 39.5°C**

**Liquid zone compressed air system (room 2)**

The compressor in room 2 did not operate at full load during the assessment, with the average current at 45 A as shown below, which represents 70% of compressor`s full load.

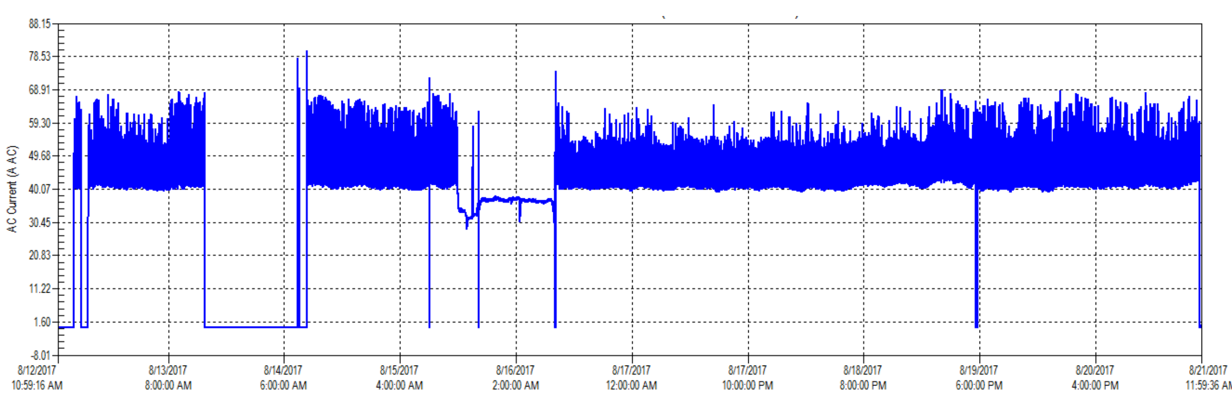



**Figure 14-52: Compressor 3 load profile**

**Drain Valves**

The drain valves of the air compressors and the tank are always partially opened to get rid of the condensed water. The estimated compressed air losses through the three open drain valves is approximately 252,288 $m^3$/year, which represents 3.7% of the total compressed air production. The drain valves are therefore costing 20,542 EGP/year in terms of compressed air loss.

**Figure 14-53: Tank Drain Valve**

**Table 14-17: Estimated quantified open drain valve loss**

| | | |
|---|---|---|
| operating pressure | 6 | bar |
| orifice | 2 | mm |
| leak amount per point | 0.16 | m3/min |
| number of points in the facility | 3 | points |
| total cfm leaks | 252,288.00 | m3/year |
| energy losses | 28,968 | kWh |
| energy losses cost | 22,218 | EGP/year |

**Leaks**

Some leaks were identified and listed during the assessment. This was brought under the attention of the plant personnel, who agreed to schedule and fix the points during maintenance downtime, and to do a regular periodical check of the remaining leak points.

**Figure 14-54: Examples of identified leak points on the demand side**

**Inappropriate use**

An identified inappropriate use application was the using of compressed air to cool the bearing of Dryer 5, as shown below. Regular bearing failure occurred due to overheating, thus compressed air was uses to cool the bearing to avoid failure. The valve is 10 mm diameter and fully opened at an operating pressure as 4.5 barg.

**Figure 14-55: Inappropriate use of compressed air (Bearing cooling)**

**Table 14-18: Estimated quantified bearing cooling loss**

| | |
|---|---|
| operating pressure | 4.5 bar |
| orifice | 10 mm |
| leak amount per point | 3.1 m3/min |
| number of points in the facility | 1 point |
| total cfm leaks | 1,629,360.00 m3/year |
| energy losses | 187,084 kWh |
| energy losses cost | 143,493 EGP/year |

**Pressure drop**

The main header from compressor room to air storage has a cross-section of 2 inches with a total length 130 m. Spot measures indicated that there is a pressure drop of 0.4 bar when one compressor is operational, and 0.89 bar when two compressors are both operating. The following figures show the pressure drop for each case.

**Compressor operating conditions**

During the assessment in accordance with a recommendation to improve compressor room operating conditions, plant personnel agreed to install a ventilation system for the air compressor room. A measurement was taken before and after the ventilation system and found that the air compressors' energy consumption reduced by 3% due to the room temperature reduction.

**Figure 14-56: Newly installed ductwork to improve operating conditions in compressor room**

**Dryer**

During the assessment it was found that the dryers were malfunctioning as the dryers were in a bypass mode, which resulted in no drying of the compressed air occurring before delivering to the process. There were no complaints from production regarding wet air, so it is recommended to shut down the dryers until the appropriate quality requirements for the end use equipment is established.

**Replacement with VSD compressor**

A longer term, lower priority consideration, which is not recommended at this stage due to the high capital requirement, is to match the demand with a variable speed compressor at a point in future when a replacement compressor is considered. Based on the calculated baseline and the technical data sheet for several VSD air compressors, the calculated potential energy savings from installing a VSD air compressor is tabled below:

| | |
|---|---|
| Calculated Annual flow rate | 6,323,800 m3/year |
| current SEC | 0.115 kWh/m3 |
| VSD proposed SEC | 0.102 kWh/m3 |
| **Annual Energy Savings** | **82,339 kWh** |
| **Energy losses cost** | **63,154 EGP/year** |

## e. Recommended Energy Efficiency Measures

**#EEM1: Drain Valves**

It is recommended that the factory install electronic level sensing drain traps on the drain valves to eliminate compressed air losses through the drain valves.

| **Timed Electric trap for the drain valves** | |
|---|---|
| **Investment** | 45,000 |
| **Savings ($m^3$/year)** | 231,264 |
| **Savings (EGP/year)** | 18,830 |
| **Payback time** | 2.4 years |

**EEM#2: Inappropriate Use**

The loss of compressed air for cooling the bearing is quantified as 1,629,360 $m^3$/year. The cost of this is calculated as 132,665 EGP/year. It is recommended to either use a fan or blower or air-knife application, or that these bearings be replaced with bearings that can work under a high-temperature condition.

**EEM#3: Pressure Drop**

It is recommended to move the two air compressors to the storage tank room to eliminate the pressure drop occurring through the underground pipe. This will also reduce the cycling on the AC2.

| | |
|---|---|
| Supply pressure | 6 bar |
| Tank pressure | 5.1 bar |
| **pressure drop** | **0.9 bar** |
| Investment | 0 EGP |
| **energy consumption increase for each ΔP 1 bar** | **7.00 %** |
| **energy losses** | **45,744 kWh** |
| **energy losses cost** | **35,086 EGP/year** |

**EEM#4: Remove or shut down non-operational dryers**

The table below shows the estimated annual energy consumption through each dryer, which is currently not performing any drying functionality.

| | | |
|---|---|---|
| Two Air dryers current | 6 | A |
| Working hours | 8760 | hrs |
| Investment | 0 | EGP |
| **Annual Energy consumpion** | **29,405** | **kWh** |
| **Energy losses cost** | **22,553** | **EGP/year** |

**Summary table of recommended energy efficiency measures**

The table below shows the summary of the recommendation measures to reduce the energy consumption in the compressed air system.

| Recommendation | Baseline before EMO, kWh | Energy Savings, kWh | Energy Savings, EGP | Energy Savings ratio % | Implementation Cost, EGP | Simple Payback, years | Carbon Reduction (kg p.a.)[7] |
|---|---|---|---|---|---|---|---|
| Drain Valves | 726,101 | 28,968 | 22,218 | 3.7 | 45,000 | - | 15.75 |
| Inappropriate use (Bearing cooling) | | 187,084 | 143,493 | 25.7 | Bearing cost | Low cost | 101.72 |
| Pressure drop | | 45,744 | 35,086 | 6.3 | 0 | - | 24.87 |
| Dryer | 29,405 | 29,405 | 22,553 | 100 | 0 | - | 15.98 |
| **TOTALS** | | **292,201** | **223,350** | **≤ 40.2** | | | **158.3** |
| Variable speed Air Compressor | | 82,339 | 63,154 | 11.3 | 89,000 | 1.4 | 44.77 |

[7] 0.5437 kg $CO_2$/kWh

## 14.8 Beshay Steel - Sadat City

### a. Company Background

Beshay Steel group is one of the largest steel producers in Egypt and the Middle East. The group employs more than 4,000 handpicked personnel, qualified at the highest levels to continue to exceed the standards of the industry.

The company is capable of producing over 2 MTPY of steel long products, re-bars, wire rods and merchant bars. The majority of the production meets the demands of the local market and the balance is exported to the Middle East, Africa, Europe and Asia.

The group is divided to three parts:

1. International steel rolling Mill (ISRM)
2. **Egyptian American Steel Rolling Co. (EASROC)**
3. Egyptian Sponge Iron Steel Co. (ESISCO)

EASROC plant is located in Sadat City, 5$^{th}$ industrial zone, Menofia Governorate, Egypt. It has a production facility comprised of steel melt shop and two rolling mills plants. The annual production capacity of the rolling mills is 600,000 Tons /year for each plant. The products are re-bars, and wire rods.

EASROC is ISO: 9001, ISO: 14001, ISO: 18001, CARES, and ISO: 50001 certificated, which reflects top management commitment to energy saving issues. The energy committee was formed accordingly in 2014 to establish the energy management system in EASROC as indicated by company top management.

As a leading steel manufacturing company, Egyptian American Rolling Company is committed to implement and continually improve an effective and transparent energy management system in all its energy intense works for the good of the nation and the community.

Top management determined to sustainability provide all necessary information and resources to:

- Gradually reduce energy consumption.
- Comply with the national environmental and legal requirements.
- Consider energy performance improvements in design, modification and procurement.
- Focus on employees' awareness and capacity building as a key factor in the success of the policy.

- Assure that the policy is documented and communicated at all levels within the company and is regularly reviewed and updated as necessary in addition to its objectives and targets.

## b. Compressed Air System Overview

The compressed air system consists of eight air compressors of 56 kW each, and two air compressors at 75 kW per each. There are five air dryers with a rated capacity of 600 cfm each. The system is controlled by a master control, which set the load and unload settings for each compressor. The operating pressures are between 6.6 and 8 barg. There are four wet receiver tanks [3.3 $m^3$ each] and two dry receiver tanks [5.4$m^3$ each]. The main pipe in the compressed air room is 6-inch, which is then reduced to a 4-inch pipe after the dry tanks.

**Figure 14-57: Compressor room**

Figure 14-58: Air dryers

**Table 14-19: Compressors' settings from nameplates**

| | Load Pressure | Unload pressure | Working hours | Load hours | Load to working hrs ratio |
|---|---|---|---|---|---|
| **Air Compressor 1** | Out of service | Out of service | Out of service | Out of service | N/A |
| **Air Compressor 2** | 6.9 | 7.3 | 56,410 | 51,104 | 91% |
| **Air Compressor 3** | 7 | 7.9 | 10,290 | 1,588 | 15% |
| **Air Compressor 4** | 6.7 | 7.4 | 45,001 | 40,207 | 89% |

| | | | | | |
|---|---|---|---|---|---|
| **Air Compressor 5** | 7.3 | 8 | 1,366 | N/A | N/A |
| **Air Compressor 6** | 7.6 | 8.3 | 2,958 | 2,312 | 78% |
| **Air Compressor 7** | 7 | 7.7 | 59,558 | 52,700 | 88% |
| **Air Compressor 8** | 6.6 | 7.5 | 11,792 | 2,046 | 17% |
| **Air Compressor 9** | 6.9 | 7.6 | 8,851 | 1,884 | 21% |
| **Air Compressor 10** | 6.6 | 7.3 | 41,802 | 34,459 | 82% |

The single line diagram below indicates the process flow, a well as the compressors` information and the size of the storage tanks.

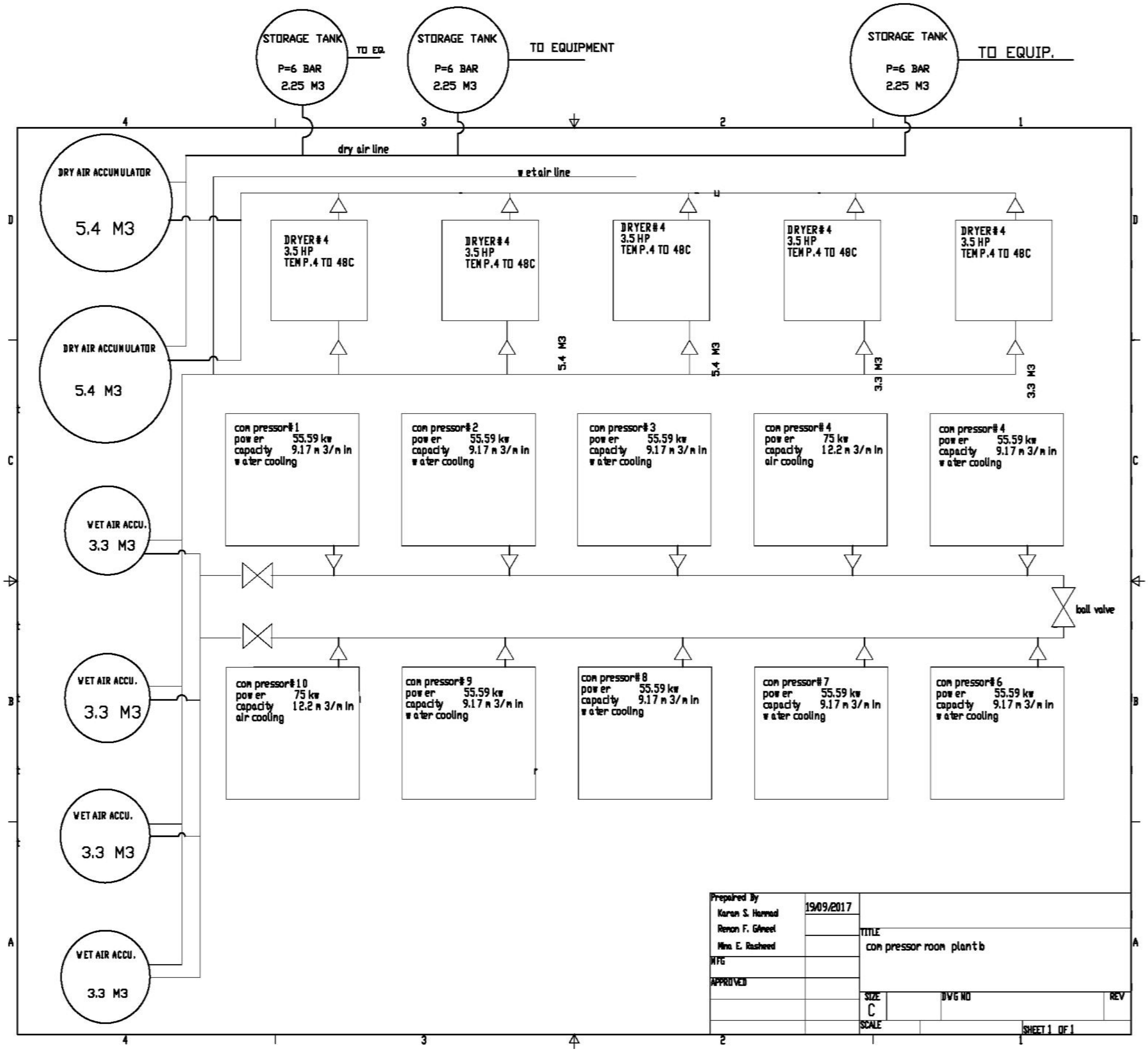



**Figure 14-59: Compressed air system single line diagram**

## c. Baseline

Based on the measured data in determining the baseline, the specific energy consumption was 0.15 kWh/m$^3$ with total compressed air consumption at 21,514,218 m$^3$/year (2,205,144 EGP/year). These projections were based on the measured data during the assessment over a measurement period of one week. The electricity tariff is 0.676 EGP/kWh. The compressed air system specific cost is quantified at 0.116 EGP/m$^3$.

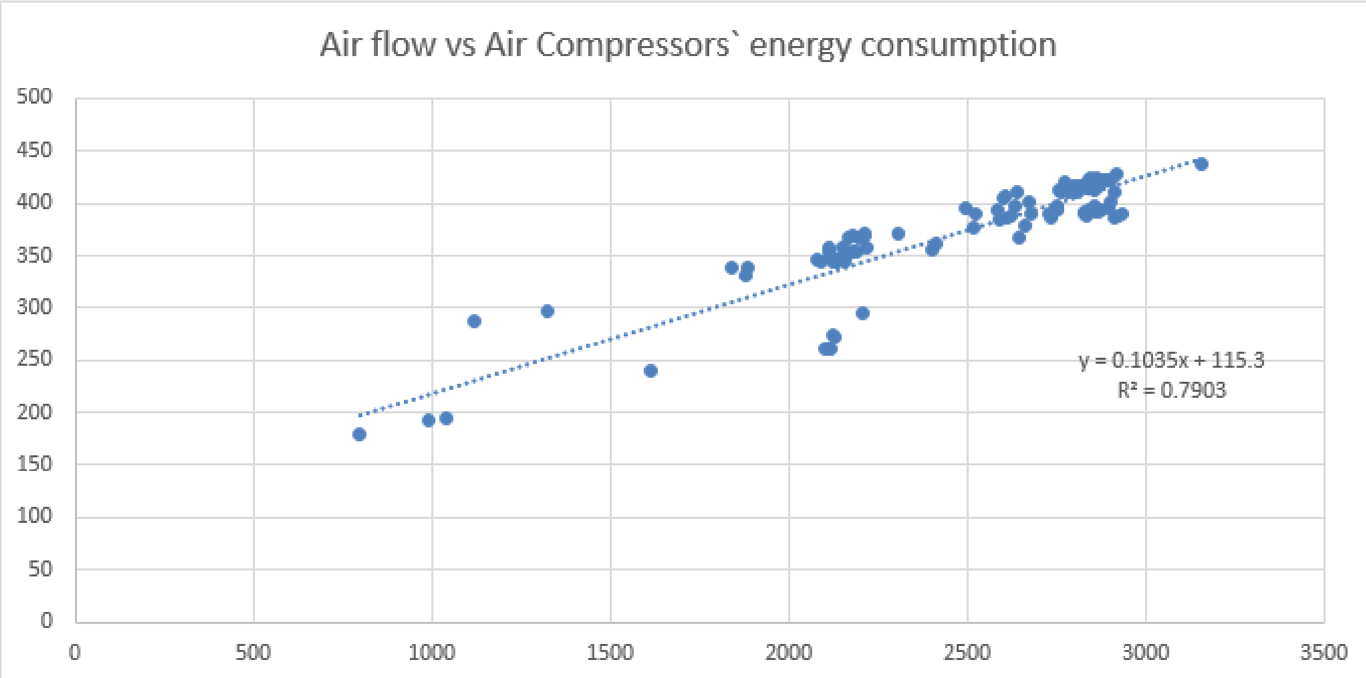
| Min Value (AF) | Max Value (AF) | Average Value (AF) | Total Flow | Measure-ment Period | total kWh | SEC | CA cost for the measured period | EGP/year | Cost of m$^3$ |
|---|---|---|---|---|---|---|---|---|---|
| 10.8 m$^3$/min | 72.22 m$^3$/min | 39.77 m$^3$/min | 280, 446 m$^3$ | 114.19 hrs. | 42,522 kWh | 0.15 kWh/m$^3$ | EGP 28,744.91 | EGP[8] 2,205,144 | 0.116 EGP/m$^3$ |

**Regression analysis between measured air flow and compressed air electricity consumption**

The chart below shows the relationship between the measured air flow and the air compressors' energy consumption. There is a relatively strong relationship as the $R^2$ is around 79%. This regression analysis is based on the measured flow and the air compressors` energy consumption during the assessment of 115 hrs. The y-intercept or baseload represents the air compressors energy consumption at no air flow production, which is 115.3 kWh.

[8] Proposed Energy consumption is **3,227,133** kWh/yr.




## d. Key Findings during the Assessment- Compressor power (kW) measurements

During the assessment, two clamp ampere meters with one data logger for the two air compressors, and one clamp ampere with one data logger for the dryer and the ambient room temperature, were installed. From the measured data it was evident that AC 9 & 10 were running unloaded for most of the time. Flow was also measured during the assessment, with the flow profile depicted below:

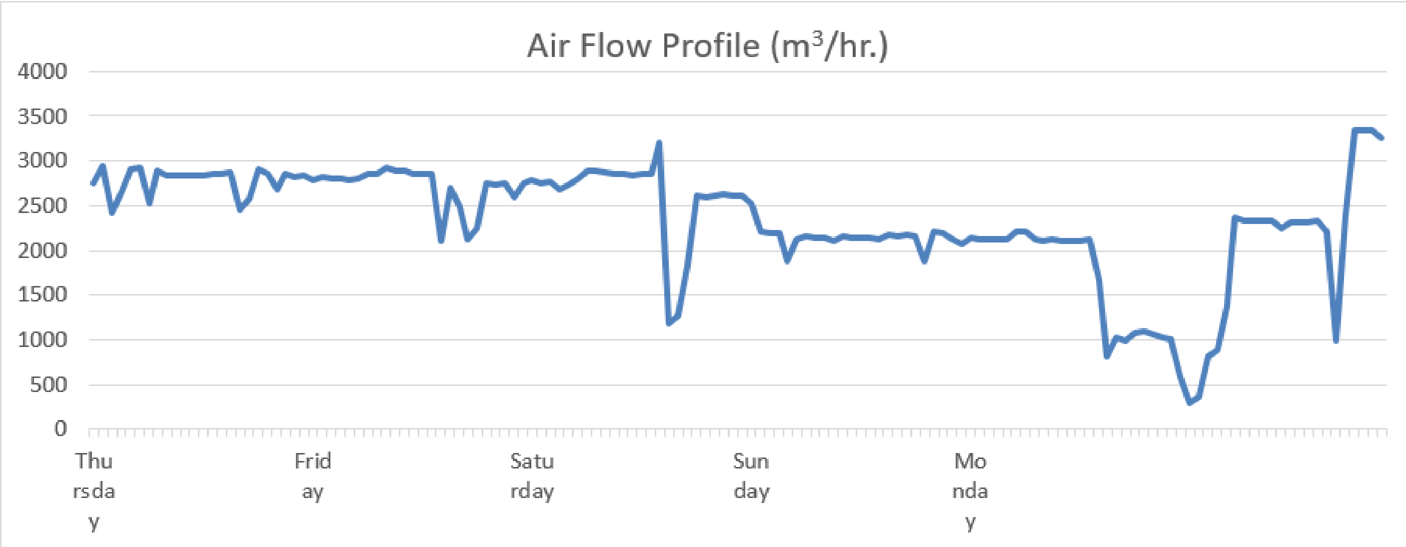



**Figure 14-60: Air Flow profile**

**Condensate drainage**

The receiver tanks` blowdown is manually open at certain periods to remove the condensed water.

**Figure 14-61: Manual drain valve for wet tanks**

**Filtration**

An air filter bypass valve was found partially open as depicted below.

**Figure 14-62: Partially open filter bypass**

**Leakages**

Some leakages were found during the assessment, which the factory team committed to in the preparation of an identified leakages list, which will be periodically fixed in downtime, and then managed appropriately.

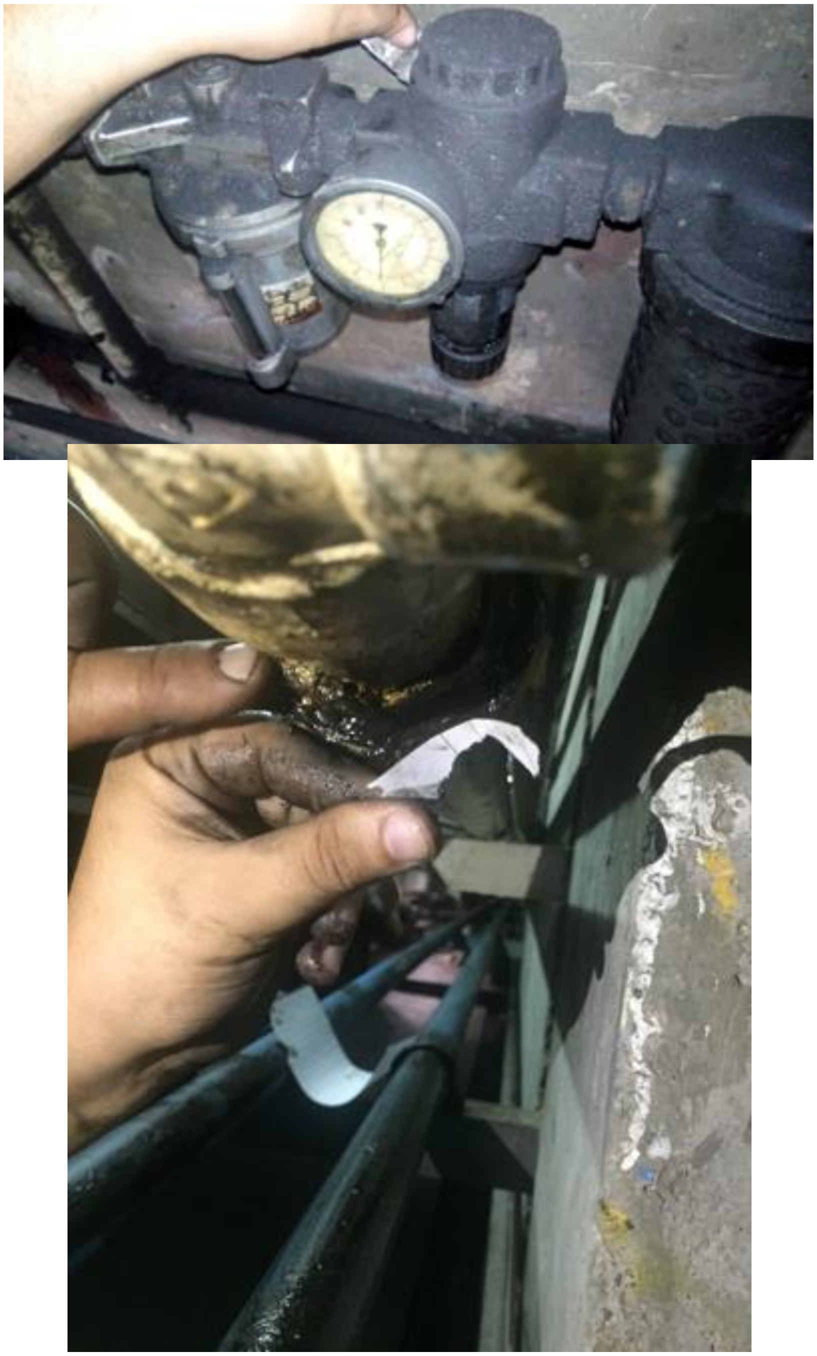


**Figure 14-63: Examples of some identified leakages**

**Inappropriate Uses – Bearing Cooling**

The production team indicated they had problems with bearing failures due to its overheating. Concurrently, they decided to use compressed air to cool the bearings to reduce the bearing failure rate.

**Figure 14-64: Bearing Cooling**

**Inappropriate Use – Air Curtain**

Compressed air is used as an air curtain in pinch roll to prevent more Oxygen from entering the furnace, and to assist in keeping the furnace pressure higher than atmosphere pressure, which causes more oxidation of the melting iron.

**Inappropriate Use – Stand of Rod Milling**

After using water to remove impurities from the hot rods, compressed air is used to remove the water vapour away of the product. There are 14 stands, with 6 hoses fully opened to remove the water vapour.

**Compressor Control**

The master controller is not functioning correctly. The compressors are only controlled in a cascade fashion with its operation based on the set pressure without any interfere from the controller. Compressors 9 & 10 are operating on standby mode most of the time, as reserve for when sudden increase in demand occurs. Compressors 2 & 8 is both frequently cycling, compressor 5 is frequently stopping and starting, and compressor 7 is operating as both load-unload, and start-stop.

24 $m^3$ of useable storage is available, which gives a 75 kW compressor 2 minutes in the case of a supply event, to start an additional compressor and supply air into the system without affecting the network pressure, instead of keeping a compressor continuously in standby mode.

It is imperative that the master control setting be readjusted to eliminate excessive start-stop operation and to not run compressors continuously in standby mode.

**Variable Speed Drive Compressor**

In the longer term, once the state energy efficiency measures have been successfully implemented, a VSD compressor could be considered when an existing operating compressor is reaching its end of life. This will enable the system to more closely match the required fluctuating demand with demand at a lower specific energy use level, which is currently not the case due to compressors being of similar sizes giving less options when demand has to be matched.

The simulation profile below shows the current situation with load-unload mode versus the proposed system which includes one VSD compressor.

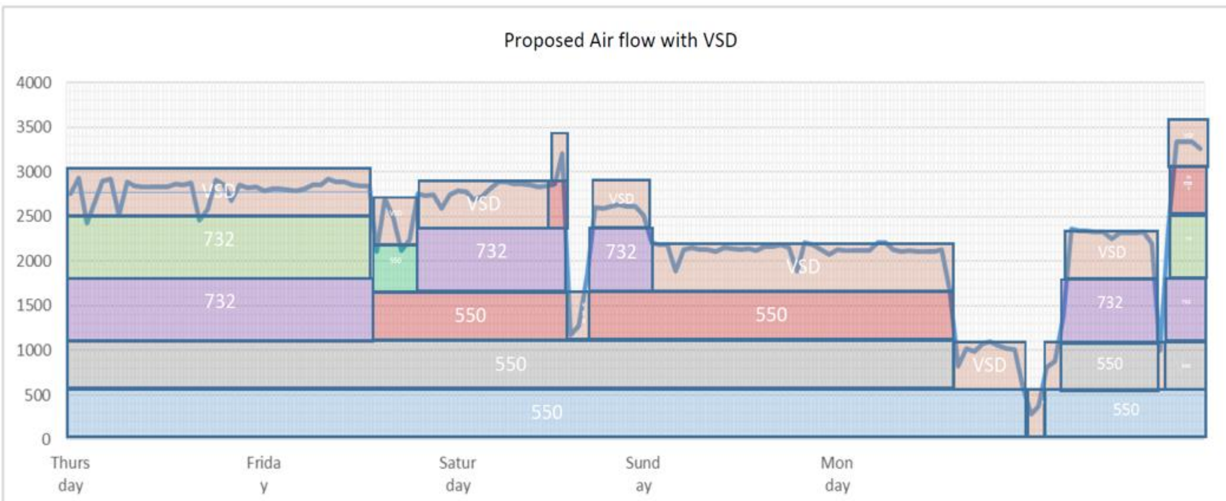



**Figure 14-65: Current compressors` control versus a proposed system with a VSD trim compressor**

The VSD option is not recommended for implementation at this stage, since the company has to finish the rest of the energy efficiency measures first before commencing with the study of the VSD option.

## e. Recommended Energy Efficiency Measures

**EEM#1: Drain Valves**

The manual valves that were found open should be replaced with electronic level sensing drain valves, which is estimated to reduce compressed air losses by 752,997 m$^3$/year – a saving of 77,180 EGP/year.

**Table 14-20: Business case for drain valves replacements**

| | Installing Automatic Drain Valves |
|---|---|
| **Investment** | 120$/unit |
| **Number of Units** | 5 |
| **Annual Savings ($m^3$/yr.)** | 752,997 |
| **Annual Savings (kWh/yr.)** | 112,949 |
| **Annual Savings (EGP/yr.)** | 77,180 |
| **Payback Period (yr.)** | 0.14 |

### EEM#2 – EEM#4: Inappropriate Uses

The estimated compressed air used for cooling bearings is 4,782,960 $m^3$/year. The estimated compressed air used for the air curtain is 3,048,480 $m^3$/year. The estimated compressed air used to remove the water vapour from the rods is 4,635,792 $m^3$/year.

**Table 14-21: Financial details pertaining to the identified inappropriate uses**

| | Bearing Cooling | Air Curtain | Compressed air used in the stands of rod milling |
|---|---|---|---|
| **Operating Pressure** | Ambient pressure | Considered as leak points | Ambient Pressure |
| **Orifice** | 0.5 inch (12.7 mm) | 3 mm | 0.25 inch (6.35 mm) |
| **leak amount per point** | 9.1 $m^3$/min (Hose length 2 m) | 0.29 $m^3$/min | 1.47 $m^3$/min (Hose length 2 m) |
| **number of points** | 1 | 20 | 6 |
| **Compressed air ($m^3$/yr.)** | 4,782,960 $m^3$/yr. | 3,048,480 $m^3$/yr. | 4,635,792 $m^3$/yr. |
| **Energy Consumption** | 717,444 kWh/yr. | 457,272 kWh/yr. | 695,369 kWh/yr. |
| **Energy Consumption Cost** | 484,992 EGP/yr. | 309,116 EGP/yr. | 470,069 EGP/yr. |
| **Recommendation** | Replace the bearing with a suitable bearing or use a Blower instead of CA | use a Blower instead of CA | Installing three Air blowers with rated power 1.5 kW/each |
| **Investment** | 10,000 EGP | 6,000 EGP | 9,000 EGP |
| **Proposed Energy Consumption** | = 5 kW*8760hr./yr. = 43,800 kWh/yr. | = 2*1.5 kW*8760hr./yr. = 26,280 kWh/yr. | =3*1.5 kW*8760 hrs./yr.= 39,420 kWh/yr. |
| **Proposed Energy Savings** | 717,444-43,800 = 673,644 kWh/yr. | 457,272-26,280 = 430,992 kWh/yr. | 695,369 - 39,420 = 655,949 kWh/yr. |
| **Savings (EGP)** | 455,383 EGP/yr. | 291,350 EGP/yr. | 443,421 EGP/yr. |
| **Payback period** | Less than a month | Less than a month | Less than a month |

**EEM#5: System Pressure Drop due to High Volume Intermittent Demand Event**

A high volume intermittent demand event was identified that caused the pressure to drop from 6.7 barg to 5.5 barg within 17 seconds. It is recommended to dedicate one of the existing receiver tanks in the factory to this demand events to avoid a pressure drop throughout the whole pipe network. The estimated realised saving is in the order of 5,709 kWh/year.

**EEM#6: Treatment Section Pressure Drop**

The pressure drop across the air dyers and filters is 1.3 bar. When considering average measured flow rate of 39.77 $m^3$/min, even when considering the air dryers' manufacturer correction factors, the dryers' rated capacity of 17 $m^3$/min each does not justify that five air dryers need to be operating simultaneously. The excessive pressure drop might be caused by some blocked dryers or filters and needs to be further investigated or appropriately maintained.

**Table 14-22: Business case for reducing pressure drop across treatment**

| | **Pressure drop between the wet and the dry tanks** |
|---|---|
| **Investment** | 5 x 624$/filter x 17.8 EGP/USD = 55,536 EGP |
| **pressure drop** | From 8 to 6.7 bar<br>Pressure difference 1.3 |
| **Annual Savings (kWh/yr.)** | 296,846 |
| **Annual Savings (EGP/yr.)** | 200,668 |
| **Payback Period (yr.)** | 0.276 |

**Summary Table of Energy Efficiency Measures**

In conclusion, the below table summarised the recommended energy efficiency measures, with the savings and business case parameters indicated.

**Table 14-23: Summary Table of Energy Efficiency Measures**

| Recommendation | Baseline before EMO, kWh | Energy Savings, kWh | Energy Savings, EGP | Energy Savings ratio % | Implementation Cost, EGP | Simple Payback, years | Carbon Reduction (kg p.a)[9] |
|---|---|---|---|---|---|---|---|
| **Drain Valves** | 3,227,133 | 112,949 | 77,180 | 3.5% | 10,680 | 0.14 | 61,410 |
| **Inappropriate use (Bearing cooling)** | | 673,644 | 455,383 | 22% | 10,000 | < month | 366,260 |
| **Inappropriate use (Air Curtain)** | | 430,992 | 291,350 | 14% | 6,000 | < month | 234,330 |
| **Inappropriate use (Stands of rod milling)** | | 655,949 | 443,421 | 21.3% | 9,000 | < month | 356,639 |
| **Pressure drop at peak demand** | | 5,709 | 3,859 | 0.18% | 0 | - | 3,104 |
| **Pressure across treatment** | | 296,846 | 200,668 | 9.2% | 55,536 | 0.276 | 161,395 |
| **TOTALS** | | **2,176,089** | **1,471,861** | **67.4%** | | | **1,183,138** |

[9] 0.5437 kg $CO_2$/kWh

# 14.9 Miser Fertilizers Company (MOPCO)

## a. Company Background

MOPCO, one of the petroleum sector companies, was established in 1998 inside the free zone in Damietta on a space of 400,000 $m^2$. This project produces Urea as a main product and liquid ammonia. MOPCO Train 3 began operating in 2008 and MOPCO Train 1 & 2 began operating in 2016. Each plant producing approximately 1200 metric tons per day of Ammonia (UHDE technology), 1925 metric tons per day urea Granulation (Stamicarbon technology).

## b. Compressed Air System Overview

The complete air compressor/Nitrogen generation plant consists of the air compressor, the Freon refrigerator unit, a molecular sieve station and the cryogenic unit. The liquid nitrogen storage receiver is downstream the cryogenic unit. The liquid nitrogen will be vaporized in the ambient air heated liquid nitrogen evaporator. The compressor produces compressed air at the pressure required for the expansion in the cryogenic unit.

The air is drier in a refrigerated dryer whereafter it’s sent to the molecular sieve absorber dryer. Change over between the drying and regenerating vessels is done automatically.

The dried air which is free of carbon dioxide enters the cold box where the air will be cooled down in the main heat exchanger before it enters the column. The cold air will be liquefied by the expansion of the nitrogen above the liquid surface. The liquid part contains all the oxygen and a part of the nitrogen, and found above the liquid air is the pressurized gaseous nitrogen which is free of oxygen. The produced gaseous nitrogen is fed to the nitrogen grid at 8 bar which is the battery limit.

A partial stream of the cold gaseous nitrogen from the top of the rectifier column enters the nitrogen condenser and will be liquefied by evaporating liquid air. Therefore, liquid air from the bottom of the rectifier column is fed as cooling medium to the nitrogen condenser. The liquid nitrogen is sent to the liquid nitrogen receiver via the liquid nitrogen pump.

The evaporator air will be fed from the top of the rectifier column to the main heat exchanger as cooling medium. Downstream of the main heat exchanger the gaseous air will be used as diving medium from the expansion turbine, and after expansion as the cooling medium in the main heat exchanger. Part of this air will be used for the regeneration of the molecular sieves, and the rest will be sent directly to the atmosphere via the silencer.

The operating compressor is an oil-free, rotary screw 200kW rated shaft power compressor. Its rated for 11 bar(a) pressure and its discharging 8.8 bar(a). Its design flow supply is 1550 $Nm^3$/hour. Only this one compressor is in continuous loading operation, and at the time of the assessment, had 86,270 running hours. It was installed in 2008.


**Figure 14-66: ZR200 Compressor**

## c. Baseline

At the time of the compressors' commissioning, an energy and flow meter was also installed, from which the baseline data was derived. Based on two years of data, the average monthly energy consumption and flow supply was 142,031 kWh/month and 1,052,833 $Nm^3$/month, respectively.

The specific energy consumption is 0.1349 kWh / $Nm^3$ at a cost of 0.0793 EGP / $Nm^3$.

## d. Key Findings during the Assessment

**Condensate drains**

During the assessment permanently open drain valves on the aftercooler (2 drains), refrigerated dryer, and on the desiccant dryer was identified. These valves are continuously open at 25 % of their diameters. These valves represent continues source of air losses.

## e. Recommended Energy Efficiency Measures

**EEM#1: Drain valves**

The quantified compressed air losses due to the partially open condensate drains is equivalent to 2.9 $m^3$/min each, or 11.8 $m^3$/min in total from these 4 drain valves. This represents an annual loss of 5,852,304 $m^3$/year, or a loss of 789,475 kWh/year.

At 0.5882 EGP/kWh, these losses are equivalent to 464,369 EGP/year. The main idea here is to replace all these valves with electronic level sensing drain traps. The price of these traps is approximately 25000 EGP each.

Opportunity summary:

- Investment required: 100,000 EGP
- Energy saving: 789,475 kWh /year
- Cost saving: 464,369 EGP/ year
- Simple payback: 3 months
- $CO_2$ reduction: 315 ton/ year

**EEM#2: Heat Recovery**

A study was done based on using demineralized water instead of cooling water for the existing heat exchanger. 206 kW of available energy heat recovery was calculated based on the operating condition and by using a simulation program. Due to the amount of heat exchangers and ectothermic reactions in the plant, it is not possible to utilize additional heat, considering that it is a small portion when comparing with other heat exchangers.

Opportunity summary

- Inlet flow water: 16.6 m3/h
- Inlet temperature: 30 C
- Outlet temperature: 40 C
- Energy heat recovery: 206 kWh x 24 h x 345 days = 1,705,680 kWh/year
- Cost saving: 86,000 EGP/ year

## 14.10 Sidi Kerir Petrochemicals Company (SIDPEC)

### a. Company Background

SIDPEC is an Egyptian joint stock company established on 16 November 1997 Under Egyptian investment law. The SIDPEC area is 180.3 acres located in the El-Nahda territory – El Amreya in an industrial area including several other companies such as Egyptian Petrochemicals Co., Gasco, ETHYDCO, Alex Fiber Co., Pirelli and Alex Carbon Black. SIDPEC is considered the first integral step towards the petrochemicals future in Egypt.

The trade name of SIDPEC polymers is named "EGYPTENE®" polymer portfolio includes linear low density polyethylene (LLDPE) and high density polyethylene (HDPE).

SIDPEC's objectives are:

- Maintain Polyethylene local market leadership through strict compliance with product quality and marketing services to attain Customer's satisfaction.
- Maintain and develop global market share with aid of its technology edge in the field of polyethylene production.
- Fulfil the requirements of the Egyptian Petrochemical Co. of Ethylene instead of importing it to produce PVC.
- Maximize the value of natural gas resources in Egypt.
- Maximize the profit through production and marketing of intermediate products as LPG, Butene-1.
- Comply with Egyptian Environmental Regulations through adopting a state of the art technology that consumes less energy and has the least impact on the environment.

SIDPEC has its continuous operation through two shifts per day in seven days per week. SIDPEC production portfolio includes Ethylene, polyethylene, LPG, and Butene-1.

### b. Compressed Air System Overview

There are three compressors rated for 4,200 $Nm^3$/hr which are three-stage, water-cooled, centrifugal, electric motor driven compressors which deliver oil-free air to the plant air receiver at 9 barg. Two of these compressors are in duty and the third is standing by. About 3,000 $Nm^3$/hr is distributed as utility air into the network at 7.5 barg.

The PLC continuously regulates the capacity of the compressor from maximum air capacity to a minimum capacity by opening and closing the Inlet Guide Vanes (IGV).

Three moisture traps are installed per compressor; one downstream of the first intercooler to prevent condensate from entering stage 2, one downstream of the second intercooler to prevent condensate from entering stage 3 and one downstream of the aftercooler to prevent condensate from entering the air outlet pipe. Each trap is provided with a float valve to automatically drain condensate, as well as a manual drain valve. If condensate is formed in the air inlet silencer, this will also be drained automatically.

For the instrument air, the compressed air is routed through refrigeration package dryer that provides output dry air at +5 ºC dew point with a capacity 4,000 $Nm^3$/hr. Air from the refrigeration dryer is routed through a three bed 24-hour cycle activated alumina desiccant type dryer which produces minus 42 ºC dew point with a capacity 4,000 $Nm^3$/hr.

When dryer inlet conditions produce lower moisture loads, the dew point demand control can extend the drying cycle without purging. The moisture analyzer senses the dew point at the outlet of the dryer and extends the cycle until the preset adjustable dew point is reached, thereby reducing purge loss and extending the valve life.

The desiccant dryer is of an "externally heated" (thermal swing) adsorption type consisting of three identical drying towers (adsorbers). While one adsorber is drying the wet inlet air (downward flow), the other two adsorbers are either in "standby" mode or are being "regenerated".

Adsorber regeneration is achieved by purging a portion of the compressed air in a counter-current flow manner (upward flow). Regenerated hot air is taken from the 1st stage of compression at approximately 2 barg at 130°C to 190°C.

The regenerated hot air is then filtered through dual regeneration air filters. These filters are capable of removing 99.5% of 0.3 to 0.6 micron particles. The regeneration hot air then flows into dual regeneration heaters. In the event that the regeneration air temperature from the 1st stage of compression is low, control logic initiates the regeneration heater to boost the temperature to the desired regeneration temperature. Upon completion of the "Regeneration" cycle, the regeneration hot air is isolated, and the cool dry air is opened to start cooling of the regenerating drying tower.

Instrument air is stored in dry instrument air receiver and thus distributed to the network at 7.5 barg.

To produce breathing air, a part of the moisture air and dried air is withdrawn, routed through a breathing air package that provides output air at -56 ºC dew point with a capacity 800 $Nm^3$/hr. The air then proceeds to a coalescent pre-filter to removes oil mists, to separate liquid oil and condensate, and to automatically discharges these contaminants from the package. After the pre-filter, the air travels to the activated carbon bed to removes hydrocarbon vapors which eliminates objectionable tastes and odors.

Hereafter the air moves through the catalyst bed where the catalyst converts carbon monoxide to carbon dioxide at an efficiency of 95 percent or higher. Following the catalyst bed the air goes on to the particulate after-filter to removes fine particles to protect sensitive respirators and related equipment. Breathing air is stored in a breathing air receiver and then distributed to the network at 7 barg.

Cooling water is provided to supply the intercoolers, aftercoolers, and oil coolers. This closed system comprises a surge tank filled with demineralized hot water in which it has been cooled by supply cooling water through a heat exchanger. Demineralized cold water is supplied to the above mentioned coolers and recycled again into the surge tank.

## c. Baseline

The collected data regarding the compressed air system baseline is shown below:

**Table 14-24: Summarised calculated compressed air system baseline**

| | | |
|---|---|---|
| Annual Energy Consumption | 10,272,000 | kWh |
| Average Power | 1,254 | kW |
| Average Total Flow Rate | 6,200 | $Nm^3/h$ |
| Average Discharge Pressure | 8.8 – 9.2 (SP=9.0) | $kg/cm^2$ |
| Average Distribution Pressure | 7.5 | $kg/cm^2$ |
| Dew Point Level (instrument Air) | -74 | Degree C |
| Leakage Level | Approximately 800-1,000 | $Nm^3$ |

- Average Specific Energy Consumption (SEC) = 0.171 $kW/Nm^3$ (Theoretical Specific Energy 0.126 $kW/Nm^3$)
- 2 x 608 $Nm^3$ storage capacity ensures continuous supply of instrument air for 30 min with zero air feed (e.g. power failure case).
- Maximum instrument air production 4,000 $Nm^3/h$ (2,531 SCFM) with a present average consumption of 3,050 $Nm^3/h$ (1,930 SCFM)
- Maximum breathing air production 400 $Nm^3/h$ (253 SCFM)

## d. Key Findings during the Assessment

**Problems Identified & Observed Deficiencies**

- Continuous opening of manual drains of air compressors and air refrigeration.
- Fluctuating consumption of plant air.
- Artificial demand.
- Leakage from unknown areas.

**Potential for Improvements**

- Utilize the surplus of compressed air.
- Optimize system pressure.
- Detect areas of leaks.
- Integrate with other air systems.
- Exclude users that don't need special specifications of air.

**Compressor Control**

Two out of the three compressors are in duty; one is fully loaded (Inlet Guide Vane (IGV): 100% & Blow Off Valve (BOV): 100%) and the other is partially loaded (IGV: 50% & BOV: 100%). Both compressors are operating continuously to supply all users with the required quantity of plant air, instrument air and breathing air.

The use of excess plant air produced from the utilities air compressors, which performs decoking for the cracking heater instead of the decoking compressor. Trials were performed 2014; actual decoking time was 35 hours and for about 20-22 hours of this 35 hours, the decoking process used plant air. The results of these experiments show that saving shall be about 55 % of decoking compressor electrical consumption. This project also affected the BOV of the swing compressor, as it consumed the excess air that led to the lowering the fluctuation of the valve during decoking process.

Another project was identified; to decrease the IGV minimum opening position from 50 % to 35 %. The objective of this project is to save about 3.8 % of total electrical consumption of air compressors package. A trial was performed in 2017 where energy meter readings were recorded for the swing compressor (C) at a IGV opening of 50%. The actual reading here was 509 kWh during a measurement period of four days. Hereafter the IGV opening was reduced to 35% and the actual reading was 470 kWh during a four-day period. The difference between the two cases is 38 kWh, which is equivalent to 337 MWh per year savings in electrical consumption (≈ 190,000 EGP).

Another suggested opportunity similar to the decoking compressor, are to use the excess plant air produced from the utilities' compressed air to perform aeration for the bioreactor, instead of running the bioreactor air compressors in the ethylene plant.

**Pressure Reduction**

The compressor discharge pressure set point is 9.0 bar in the plant air receiver, while the network pressure is 7.5 bar. So there is about 1.5 bar difference between system pressure and the network pressure. It is recommended that the discharge pressure set point is reduced by at least 0.5 bar. The objective of this project is to save about 3 % of the compressed air system's electrical consumption. This equivalent to about 300 MWh per year (≈ 170,000 EGP).

Supplying 1.5 bar more pressure to the system will force unregulated uses in the system to consume about 20% more air flow. The reduction in system pressure will have an influence on the useable storage volume, therefore a test must be performed to notice the effect of this change on the compressor control mode.

**Storage Volume**

The surge volume of instrument air system (instrument air receiver plus plant air receiver) at power failure is designed to provide instrument air consumption for 30 minutes. According to this design, surge capacity of both plant air and instrument air receivers is about 1,216 $Nm^3$. Actual total consumption of instrument air in 2016 was about 3,300 $Nm^3$/hr. This means that surge volume requirements from both receivers could provide instrument air for about 22 minutes. There is also an instrument air receiver in the polyethylene plant with a capacity of 562 $Nm^3$ which can provide instrument air for the polyethylene plant for about 20 minutes.

**Inappropriate Use**

After the beginning of start-up in 2000, there was only one compressor in duty and two in stand-by. Since then a problem occurred regarding bagging, which was solved by the use of about 1,500 $Nm^3$/hr of plant compressed air, which led to the additional operation of the second air compressor.

Exclude bagging unit which from the compressed air distribution system. Approximately 23% of the compressed air total air consumption could be saved, which is equivalent to about 300 MWh per year (≈ 170,000 EGP). It is further recommended to install a new air compressor in the bagging unit (22kW) and an air tank to isolate the bagging unit air consumption from the plant air network.

**System Integration**

The Nitrogen unit in the polyethylene plant uses air to produce nitrogen through two nitrogen compressors. There is a surplus of air that can be used for process air applications.

Connect the Nitrogen unit lines to air network. This idea focusses on the availability of tying the plant air and instrument air outlet from either the ethylene or/and polyethylene nitrogen plants to the plant air and instrument air network, in order to save the operation of one compressor in the utilities department. This idea shall be further discussed and investigated in detail in SIDPEC's internal polyethylene nitrogen compressors assessment report.

**Leakages**

One year after the initial plant start-up, there was about 5,100 $Nm^3$/hr total average flow rate, (2,700 PA + 2,400 IA). In 2016 the average flow rate was about 6,400 $Nm^3$/hr (3,300 PA + 3,100 IA). The increase in flow demand can also indicate to an increase in the leakage rate, especially since no leakage management or quantification has been done as of yet. It is recommended that plant personnel check the distribution networks of both plant air and instrument air using an ultrasonic acoustic detector to detect air leakage. Air leaks can occur in any part of the system, but the most common problem areas are: couplings, hoses, tubes, fittings, pipe joints, quick disconnects, FRLs (filter, regulator, and lubricator), condensate traps, valves, flanges, packings, thread seal-ants, and point-of-use devices.

**Condensate Drains**

Air compressors 70-K-9251 A, B & C are equipped with moisture traps, where every moisture trap is installed downstream of each intercooler. Each trap is provided with a float valve to automatically drain condensate as well as a manual drain valve.

Refrigeration dryers 70-X-9252 A&B are equipped with condensate traps that are mounted horizontally after and before the air-to-refrigerant heat exchanger. Each trap is provided with a solenoid valve to automatically drain condensate.

Automatic moisture level control valves are installed on both plant air receivers (wet receiver) and breathing air receiver to drain condensate. Common mechanical trap types are installed in the desiccant dryers.

It is recommended that scheduled check and maintenance are performed on the installed level-operated mechanical float traps to prevent blockage from sediment getting the trap stuck in an open position. Appropriate intervals should be used for manual moisture drains. Scheduled check and maintenance for the setting of the valves for solenoid-operated drain valves should also be carried out. Zero-loss traps are recommended (higher price but lower maintenance cost).

## e. Recommended Energy Efficiency Measures

The summarized Energy Conservation Opportunities (ECO) list according to the above measures explored is shown below:

| Ser. | Description | Measure Explored |
|---|---|---|
| ECO-1 | Load Sharing: one of provided external offers was "Equidistant Load sharing" technique. (Rejected) | Compressor Operating Mode |
| ECO-2 | Use of excess plant air produced from utilities air compressors unit in performing decoking for cracking heater instead of decoking compressor. (Achieved in Sep. 2014) | Compressor control |
| ECO-3 | Decrease IGV min. opening of air compressors from 50 % to 35 %. (Achieved in Feb. 2017) | Compressor control |
| ECO-4 | Use of excess plant air produced from utilities air compressors unit in performing aeration for in bioreactor instead of Bioreactor air compressors in ethylene plant. | Compressor control |
| ECO-5 | Reduce set point discharge pressure of air compressor by at least 0.5 bar | Pressure Reduction |
| ECO-6 | Exclude bagging unit which from the compressed air distribution system. | Flow Reduction |
| ECO-7 | Connect Nitrogen unit lines to air network | Flow Reduction |
| ECO-8 | Check the distribution networks of both Plant air and Instrument air using an ultrasonic acoustic detector to detect air leakage. | Air Leakage |
| ECO-9 | Scheduled check and maintenance on the installed level-operated mechanical float traps, Scheduled check and maintenance for the setting of valve opening duration and intervals, and use Zero-loss traps. | Condensate Drains |
| ECO-10 | Change the mode of operation from cycle time to dew point demand. (Achieved in Jan. 2004) | Air Dryers |

The summarized **ECO business case** according to the above measures explored is shown below:

| ECO No. | Expected Benefits | Estimated Saving |
|---|---|---|
| ECO-1 | Most energy efficient (100 MWh per year) and the safest way of operating compressors as it minimizes the risk of surge | 500,000 EGP |

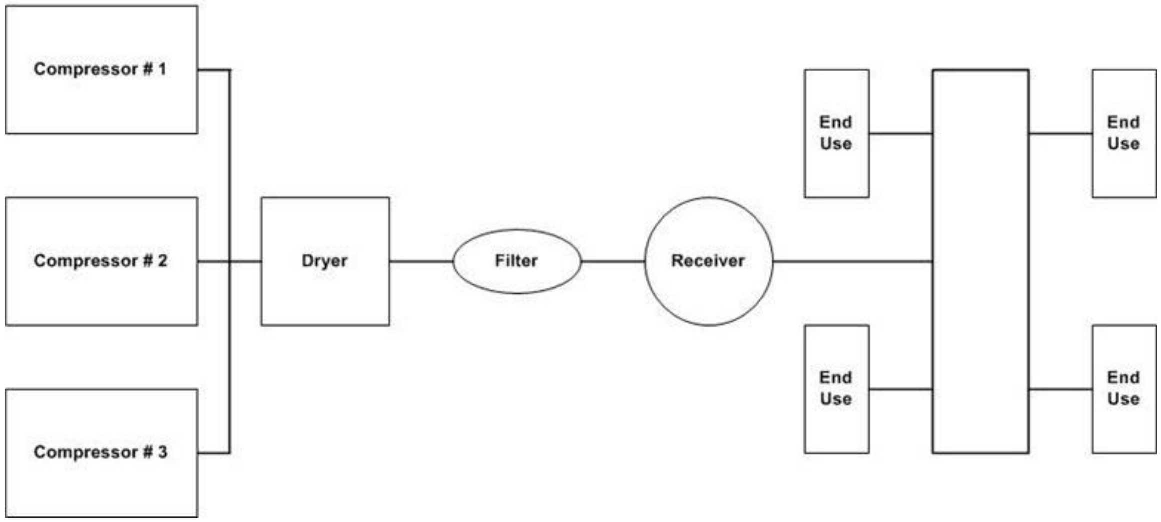
| | | |
|---|---|---|
| ECO-2 | This project affecting on BOV regarding swing compressor, as it consumed the excess air led to lowering the fluctuation of the valve during decoking process. | - |
| ECO-3 | The objective of this project is to save about 3.8 % of total electrical consumption of air compressors package (337 MWh per year) | 190,000 EGP |
| ECO-4 | As in ECO-2 | - |
| ECO-5 | The objective of this project is to save about 3 % of total electrical consumption of air compressors package (300 MWh per year). Eliminate unregulated uses in the system (20% more air flow, or 20% waste more air). | 170,000 EGP |
| ECO-6 | The objective of this project is to save about 23 % of total air consumption (300 MWh per year).<br>Install a new air compressor in the bagging unit (22KW) and air tank to isolate the bagging unit air consumption from the plant air network. | 170,000 EGP |
| ECO-7 | In addition to other ideas may be lead to saving the operation of one compressor. | HOLD |
| ECO-8 | Save uncontrolled air consumption | HOLD |
| ECO-9 | Save uncontrolled air consumption | HOLD |
| ECO-10 | Reduce air consumed for regeneration, air heater operating hours and save its life time. Increase life time of valves and dew point sensors. Reduce the noise time duration. | |

# 15. Summary of Best Practices

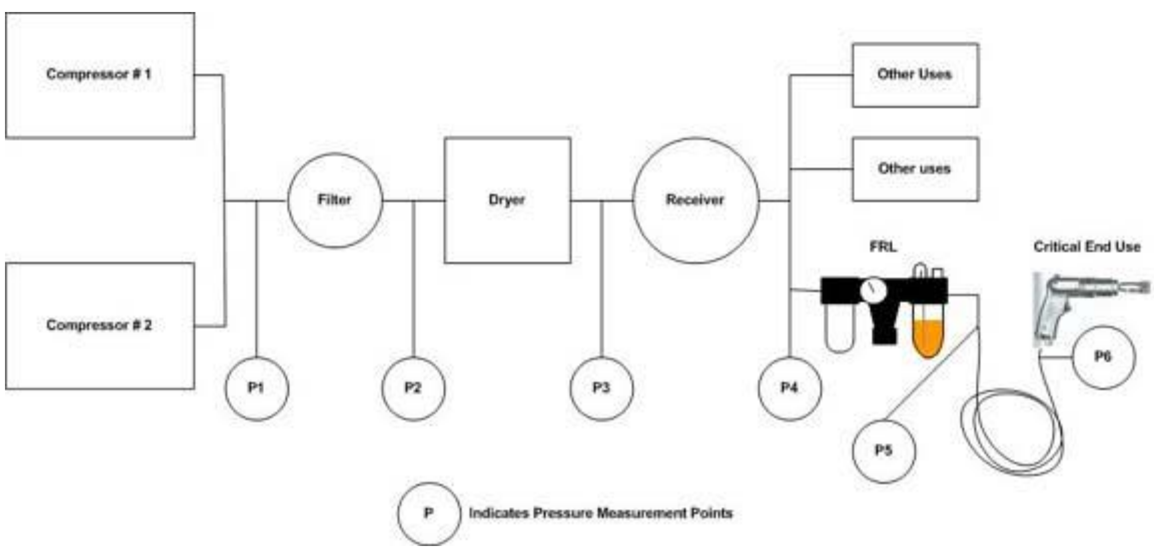
- Make sure that compressed air is the best alternative for the application.
- The cost of compressed air often is overlooked because of the convenience and ergonomic advantages it provides. Not all uses of compressed air are considered appropriate or economically feasible. Many of the productivity improvements in automated manufacturing processes have been achieved through the appropriate use of compressed air.
- Determine the minimum practical pressure required for the application and use a blower, rather than a compressor, if appropriate.
- Look for potentially inappropriate uses, which can be served better by a low-pressure blower, an electric motor drive, or vacuum pump.
- Applications that do not require compressed air 100% of the time should have the supply shut off when not needed.
- Make sure the application uses only the required amount of compressed air.
- All parts of a process may not need air simultaneously.
- The peak and average rates of flow should be analyzed to determine actual needs and whether local secondary storage may be advantageous.
- The system should be delivering air at the lowest possible pressure.
- Operating at the minimum practical pressure at end uses, together with a corresponding reduction in compressor discharge pressure(s), will reduce both the consumption of compressed air and the energy required.
- Check the appropriateness of equipment used to control and deliver compressed air.
- Are compressor controls of the right type and with the right range of operating pressures?
- Are primary and secondary receiver sizes adequate and well located?
- Is the air supply from the compressor room controlled by a pressure/flow controller? If so, is the controller and the distribution piping properly sized?
- Is pressure to points of use further reduced by FRLs (Filter/Regulator/Lubricator) and are they needed?
- Use automatic system controls to anticipate peak demands.
- Only the number of compressors required, to meet the demand at any given time, should be in operation and only one in a “trim” control mode. Automatic sequencing of compressors can optimize the selection of compressors for changing demand cycles.
- Turn off compressed air supply at a process not running.
- Stopping the supply of compressed air to applications not in operation can reduce the consumption of compressed air. This can be accomplished very easily by means of a solenoid valve in the air supply to each application.
- Determine the cost of compressed air for each machine or process.

- Accurate measurements of air consumption and electrical power allow proper assessment and appreciation of the true cost of operation. This, in turn, can help in management and conservation of available resources.
- Review and understand the cost of leakage.
- Leakage rates of 20 – 30% in the compressed air system of an industrial plant are not uncommon. An aggressive and continuous program of leak detection and elimination can reduce consumption substantially.
- Large quantities of compressed air can be lost through means of condensate drainage:
- Float type and other mechanical traps may stick in the open position.
- Manual drains are left cracked open.
- Time cycle blow-down can cause drain valves to be open even when no condensate is present.
- "Demand Drains", which open only when condensate is present and close when the condensate is discharged, can eliminate waste.
- The above remedies must be followed by a review of the number of compressors in operation and their control settings, so that a corresponding reduction in energy is realized.
- Make sure that the compressed air supply side personnel are involved in process/end use related decisions.
- Changes in processes and end uses of compressed air can impact the entire system. Required flow rates and pressures can impact the number of compressors required, their control pressure ranges, compressed air treatment equipment and the distribution system. Coordination among departments is essential for an efficient operation.
- Know what equipment you have.
- Develop a basic block diagram of compressors, dryers, filters, receivers, etc., as shown below:

## Simplified Sample Block Diagram




- Know what is happening in your system.
- Create a system pressure profile, using pressure readings at key points throughout the system as shown below:




- Know your starting point
- Measure your baseline and calculate energy use and costs, with tools available.
- Know existing and potential problems and opportunities.

- Walk through to check for obvious preventive maintenance items and other opportunities to reduce costs and improve performance.
- Identify and fix leaks.
- Address point-of-use issues.
- Determine actual air quality requirements and treat air appropriately.
- Investigate and reduce highest point-of-use pressure requirements.
- Investigate and address high-volume, intermittent applications.
- Take stock of what you have and challenge point-of-use requirements and appropriateness (or inappropriateness) of applications.
- Determine the best control strategy.
- Analyze existing compressor(s) and system controls, and implement an effective control strategy.
- Ensure compatibility of the Supply Side and the Demand Side.
- Align the demand side with the supply side operation.
- Realize that improving system efficiency is an on-going effort.
- Implement strategies to maintain system alignment.
- Know what the system costs and resulting savings.
- Know costs, re-measure and adjust controls as above.
- Ensure that equipment is properly maintained to maintain efficiency.
- Develop and implement a compressed air system maintenance program. Keep adequate records of required and actual maintenance.
- Involve all stakeholders in the decision making process.
- Communicate to gain support of plant and production management.
- Target the decision makers.
- Develop a cost-benefit analysis that addresses life cycle cost savings, benefits to production (such as reliability and productivity) and return on investment.
- Report to management using an effective format.
- Keep good records of measurements, before and after any changes.
- Use pre-measurement and post-measurement of kWh and production output to document cost savings from actions taken with production's support, and report improvements to management.

Survey: Most common objectives of compressed air system management

- Maintain continuous operation
- Ensure adequate supply of air
- Maintain quality of air
- Preventive maintenance
- Control or reduce energy use

Survey: Problems most commonly reported by operating personnel:

- Excess moisture in compressed air.
- Inadequate pressure in whole system
- Contaminants in compressed air
- Frequent fouling of inlet and/or pipeline filters

Survey: Problems most commonly reported by compressed air system expert consultants:

- Controls not working properly and/or not understood
- Timer activated drains instead of level activated
- Improperly sized filters and dryers causing pressure loss and performance
- Regulators cranked open to full header pressure
- Pressure drop at the point of use because of “improper” piping or FRL size
- Open blowing
- Using excessively high pressure for a low pressure application
- Demand side waste; potentially inappropriate uses and leaks
- Lack of provision of measurements
- Poor sub-headers for high flow intermittent demands
- Undersized piping, high air velocities
- Operating end use equipment higher than the design pressure
- Unregulated uses
- Pressure supplied to main headers is too high